Chapter 10
Wave-to-Wire Modelling of WECs

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10.1 Introduction

Numerical modelling of wave energy converters (WECs) of the wave activated body type (WAB, see Chap. 2) is based on *Newton*'s second law, which states that the inertial force is balanced by all forces acting on the WEC's captor. These forces are usually split into hydrodynamic and external loads.

In general, the **hydrodynamic source** comprises the (more details in Chap. 6):

- **Hydrostatic force** caused by the variation of the captor submergence due to its oscillatory motion under a hydrostatic pressure distribution,
- **Excitation loads** due to the action of the incident waves on a motionless captor,
- **Radiation force** corresponding to the force experienced by the captor due to the change in the pressure field as result of the fluid displaced by its own oscillatory movement, in the absence of an incident wave field.

Depending on the type of WEC, the **external source** may include the loads induced by the

- **Power-take-off** (PTO) equipment, which converts mechanical energy (captor motions) into electricity (more details in Chap. 8),
- **Mooring system,** responsible for the WEC station-keeping (more details in Chap. 7),
- **End-stop** mechanism, used to decelerate the captor at the end of its stroke in order to dissipate the kinetic energy gently, and therefore avoid mechanical damage to the device.

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The hydrodynamic modelling of the interaction between ocean waves (see Chap. 3) and WECs is often split into three different phases according to the sea conditions:

(i) During small to moderate sea states linear wave approximations are valid, corresponding to the current state-of-the-art methods of hydrodynamic modelling.

(ii) Under moderate to extreme waves, in general, some sort of non-linear hydrodynamic modelling is required in order to more accurately model the wave/device interaction.

(iii) Ultimately, under stormy conditions, a fully non-linear approach is necessary to model the hydrodynamic interaction of the waves and the device.

With respect to the modelling of the external loads it is commonly accepted that the production of energy should be restricted to non-stormy conditions (WEC operating mode), comprising both small to moderate and moderate to extreme waves, which correspond to low and intermediate energetic sea states. Under stormy conditions, usually it is not necessary to model the dynamics of the PTO equipment as the WEC is interacting with extreme waves and so it must assume the survivability mode with no energy production. In the operating mode of the device the loads induced by the **PTO equipment,** the **mooring system** and the **end-stop mechanism** may be linearized under certain assumptions; however, typically they exhibit strongly nonlinear behaviour, which requires a time domain approach in order to be described properly.

Although the scope of this chapter is confined to wave-to-wire modelling it is important to emphasize that there are other modelling methods and that the most adequate one depends on several factors such as the required accuracy (which is typically inversely proportional to the computational time), the sea state (stormy or non-stormy conditions), the device regime (operational or survival mode) and its work principle (some concepts exhibit more non-linear behaviours). In view of that, the modelling tools are typically split into 3 different types:

1 **Frequency models**: The hydrodynamic interaction between WECs and ocean waves is a complex high-order non-linear process, which, under some particular conditions, might be simplified. This is the case for waves and device oscillatory motions of small-amplitude. In this case the hydrodynamic problem is well characterised by a linear approach. Therefore, in such a framework (which is normally fairly acceptable throughout the device's operational regime), and with linear forces imposed by both the PTO and the anchoring system, the first step to model the WEC dynamics is traditionally carried out in the frequency domain (where the excitation is of a simple harmonic form). Consequently, all the physical quantities vary sinusoidally with time, according to the frequency of the incident wave. Under these circumstances, the equations of motion become a linear system that may be solved in a straightforward manner.

Although frequency models have limited applicability, being restricted to linear problems where the superposition principle is valid, the frequency domain

approach is extremely useful as it allows for a relatively simple and fast assessment of the WEC performance, under the aforementioned conditions. Hence, this approach is generally used to optimise the geometry of WECs in order to maximize the energy capture [1–3].

2 **Wave-to-wire models** (time domain tools): Besides the interest of the frequency domain approach, in many practical cases the WEC dynamics has some parts that are strongly non-linear, and so the superposition principle is no longer applicable. These nonlinearities arise mostly from the dynamics of the mooring system, the PTO equipment and control strategy and, when present, the end-stop mechanism. Furthermore, under moderate to extreme waves, nonlinear effects in the wave/device hydrodynamic interaction are more relevant. This requires some sort of non-linear modelling that typically consists of treating the buoyancy and the excitation loads as non-linear terms. In addition, second-order slow drift forces may be also included in a time domain description of the WEC dynamics (this force must be undertaken by the station-keeping system). To properly account for these nonlinearities the WEC modelling has to be performed in time domain. Moreover, the motion of the free surface in a sea state rarely reaches steady-state conditions, and so must also be represented in the time domain.

The time domain approach is a reasonably detailed and accurate description of the WEC dynamics. Since this approach allows modelling of the entire chain of energy conversion from the wave/device hydrodynamic interaction to feeding into the electrical grid, time domain models are commonly named wave-to-wire codes. The most relevant outcomes of a wave-to-wire code includes, among others, estimates of the instantaneous power produced under irregular sea states, motions/velocities/accelerations of the WEC captor and loads on the WEC. Besides, wave-to-wire models are extremely useful tools to optimize the WEC control strategy in order to maximize the power captured. The Structural Design of Wave Energy Devices (SDWED) project, led by Aalborg University, has generated a comprehensive set of free software tools including advanced hydrodynamic models, spectral fatigue models and wave to wire models [4].

3 **Computational fluid dynamics—CFD**: Due to the large computational time the use of CFD codes is typically restricted to study the wave/device interaction under extreme waves, which is a strongly non-linear phenomena. Normally, the main objective in this case is to model the WEC dynamics in its survival mode with no energy production (in order to evaluate the suitability of the survival strategy). This type of wave-body interaction is usually computed solving the Reynolds Averaged Navier-Stokes Equations (RANSE[1]) with some

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[1]The decomposing of the Navier-Stokes equations into the Reynolds-averaged Navier–Stokes equations (RANSE) makes it possible to model complex flows, such as the flow around a wave power device. RANSE are based on the assumption that the time-dependent turbulent velocity fluctuations may be separated from the mean flow velocity. This assumption introduces a set of unknowns, named the Reynolds stresses (functions of the velocity fluctuations), which require a turbulence model to produce a closed system of solvable equations.

sort of numerical technique to model the free surface of the water. Among several different methods to model the free surface one of the most commonly used is the Volume of Fluid (VoF) [5]. At present there are some CFD codes capable of modelling this sort of wave-body interaction and flows with complex free-surface phenomena such as wave breaking and overtopping (see Sect. 10.3: Benchmark Analysis).

10.2 Wave-to-Wire Models

At present there are many designs being pursued by developers to harness wave power, which may be categorized according to the location and depth in which they are designed to operate, i.e. shoreline, near shore or offshore, or by the type of power capture mechanism. However, there is no common device categorization that has been widely accepted within the international research and technology development community, but the most popular distinguishing criteria is based on their operational principle. According to this criterion WECs are usually divided into six distinct classes: attenuators; point absorbers; oscillating-wave surge converters; oscillating water columns (OWC); overtopping devices; and submerged pressure-differential devices [6]. These categories may be regrouped into three fundamentally different classes, namely OWC, WECs with wave-induced relative motions and overtopping devices. For WECs within the two first fundamental classes the generic approach to develop wave-to-wire models presented herein is valid, however, for overtopping concepts the performance analysis requires the use different type of numerical tools based on empirical expressions (such as e.g. WOPSim: Wave Overtopping Power Simulation [7]) or CFD codes.

In the field of wave energy, the term wave-to-wire refers to numerical tools that are able to model the entire chain of energy conversion from the hydrodynamic interaction between the ocean waves and the WEC to the electricity feed into the grid. In terms of complexity, and consequently time expenditure, these types of numerical tools are in-between frequency domain codes, which are much faster but less accurate (because all the forces are linearized), and CFD codes, which are currently the most precise numerical tools available, but also extremely time demanding, which makes their use unviable to solve the majority of problems in this field.

This section presents a discussion on the assumptions, considerations and techniques commonly used in developing wave-to-wire models, highlighting the limitations and the range of validity of this type of modelling tool. A general discussion is presented aiming to embrace the majority of existing WECs, nevertheless when appropriate, an annotation regarding the fundamental differences in the working principle of some particular WECs and the subsequent adjustments in the wave-to-wire model will be made.

10.2.1   Equation of Motion

In essence, the algorithm to build a wave-to-wire model relies on Newton's second law of motion, which states that the inertial force is balanced by all of the forces acting on the WEC's captor. This statement is expressed by the equation

$$M\ddot{\xi}(t) = F_e(t) + F_r(t) + F_{hs}(t) + F_f(t) + F_{pto}(t) + F_m(t), \qquad (10.1)$$

where M represents the mass matrix and $\ddot{\xi}$ the acceleration vector of the WEC. The terms on the right hand side of Eq. 10.1 correspond to:

- The excitation loads—F_e
- The hydrostatic force—F_{hs}
- The friction force—F_f
- The radiation force—F_r
- The PTO loads—F_{pto}
- The mooring loads—F_m

   In the following section a discussion on the different sources of loads on the WEC captor is presented and their impact on the overall dynamics of the WEC is given in order to substantiate the assumptions and simplifications commonly considered in the development of wave-to-wire codes.

10.2.2   Excitation Force

The **excitation force** results from the pressure exerted on the body's wetted surface due to the action of the incoming waves. The most popular approach to compute this force is based on linear wave theory, in which the body is assumed to be stationary and the area of the wetted surface constant and equal to the value in undisturbed conditions. Obviously this assumption is only valid for small wave amplitudes, which is a fundamental assumption of linear theory. Therefore, under linear assumptions the excitation load on the WEC captor is given by

$$F_{exc}(t) = \int_{-\infty}^{\infty} f_{exc}(t - \tau)\eta(\tau)d\tau, \qquad (10.2)$$

where η is the free surface elevation due to the incident wave (undisturbed by the WEC) at the reference point where the WEC is located and f_{exc} is the so called excitation impulse response function derived from the frequency coefficients commonly obtained with a 3D radiation/diffraction code (see Sect. 10.3). Equation 10.2 shows that it is necessary to model the random sea state behaviour in order to estimate the excitation force. The most common approach consists of using Airy wave theory, a linear theory for the propagation of waves on the surface of a

potential flow and above a horizontal bottom. The free surface elevation, η, may be then reproduced for a wave record with duration T as the sum of a large (theoretically infinite) number, N, of harmonic wave components (a Fourier series), the so called wave superposition method, as

$$\eta(t) = \sum_{i=1}^{N} a_i \cos(2\pi f_i t + \alpha_i), \qquad (10.3)$$

where, t is the time, a_i and α_i the amplitudes and phases of each frequency, respectively, and $f_i = i/T$. The phases are randomly distributed between 0 and 2π, so the phase spectrum may be disregarded. Hence, to characterize the free surface elevation only the amplitudes of the sinusoidal components need to be identified, which are given by

$$a_i = \sqrt{2S_f(f_i)\Delta f}, \qquad (10.4)$$

where S_f is the variance density spectrum or simply energy spectrum (see Fig. 10.1) and Δf the frequency interval. As only the frequencies f_i are presented in the energy spectrum, while in reality all frequencies are present at sea, it is convenient to let the frequency interval $\Delta f \to 0$. The spectrum of energy is usually plotted as energy density, (unit of energy/unit frequency interval, Hz) given by the amount of energy in a particular frequency interval.

Fig. 10.1 Typical variance density spectrum
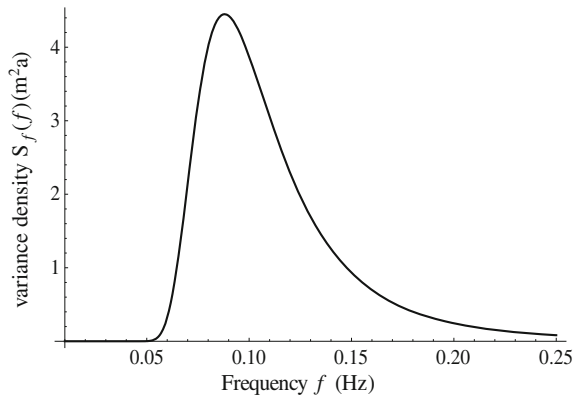
For more realistic descriptions of the wave surface elevation the wave's directionality must be considered. In this case the direction resolved spectrum $S(\beta, f)$, dependent on the frequency, f, and wave direction, β, is written as

$$S_f(f, \beta) = D(f, \beta)S_f(f), \qquad (10.5)$$

where the directional distribution $D(\beta,f)$ is normalized, satisfying the condition

$$\int_{-\pi}^{\pi} D(\beta,f)d\beta = 1. \tag{10.6}$$

The spectrum is defined with several parameters in which the most important ones are the significant wave height, denoted by H_s or $H_{1/3}$ (which corresponds to the average of the highest third of the waves), and the peak period, T_p corresponding to the period with the highest peak of the energy density spectrum (the spectrum may have more than one peak).

10.2.3 Hydrostatic Force

When a body is partially or completely immersed in a liquid it will experience an upward force (buoyancy) equal to the weight of the liquid displaced, which is known as Archimedes' principle. The **hydrostatic force** results from the difference between this upward force and the weight of the body. Accordingly, the variation of the captor submergence due to its oscillatory motion under a hydrostatic pressure distribution causes a change in the buoyancy (equal to the change of weight of displaced fluid) and hence a variation in the hydrostatic force.

A fundamental assumption of linear theory is that the resulting body motions are of small amplitude, which normally conforms with the behaviour of WECs during the operational regime. In fact, the motion of WECs tends to be of small amplitude because otherwise the dissipative viscous effects would be dominant in the device dynamics, which would ultimately limit the motion and reduce the device efficiency. Therefore, the hydrostatic force, F_{hs}, is commonly implemented in wave-to-wire models merely as a function proportional to the body displacement, where the proportionality coefficient is known as the hydrostatic coefficient, i.e.,

$$F_{hs}(t) = Gz(t), \tag{10.7}$$

where G is the hydrostatic coefficient and z the motion in the direction of the degree of freedom (DoF) being considered. In the case of several DoFs being analysed G and z represent the hydrostatic matrix and the displacement vector, respectively.

For example, in the case of a heaving body undergoing small-amplitude oscillations the variation of the buoyancy force may be simply given by

$$F_{hs}(t) = \rho g A z(t), \tag{10.8}$$

where ρ denotes the water density, g the gravitational acceleration, A the cross sectional area of the body in undisturbed conditions and z its vertical displacement. The variation of the volume of water displaced by the oscillating body is equal to the variation of its

submerged volume, given by Az. We should note that typically the assumption of constant cross sectional area along the vertical axis is only valid for motions of small-amplitude. Depending on the body geometry, typically this simplification (based on the linear wave theory) is not valid for large-amplitude motions where in general the variation of the cross-sectional area is more noticeable, and so a non-linear approach is required to accurately assess the hydrostatic force.

10.2.4 Mooring Loads

Wave drift forces,[2] along with currents and wind, have a tendency to push the WEC away from the deployment position. To prevent this drifting, the WEC should be maintained in position by a station-keeping system, also commonly called a "mooring system". The station-keeping system is usually designed to withstand survival conditions, e.g. 100 year storm conditions. The moorings designed for floating WECs are required to limit their excursions and, depending on the concept, aligning its position according to the angle of incidence of the incoming waves. Moreover, unlike typical offshore structures, the mooring design has an additional requirement of ensuring efficient energy conversion, since it may change the response of the WEC and so change its ability to capture wave energy.

Depending on the working principle of the device and ultimately the manner in which the mooring system provides the restoring force, mooring systems might be passive, active or reactive. Passive mooring systems are designed for the unique purpose of station-keeping. Conversely, active mooring systems have a stronger impact on the dynamic response of WECs since the system stiffness may be used to alter the resonant properties of WECs. Ultimately, reactive mooring systems are applied when the PTO exploits the relative movements between the body and the fixed ground, such that the mooring system provides the reaction force. In this mooring configuration the inboard end of the mooring line/s is connected to the PTO equipment which controls the tensions or loosens of the mooring line/s in order to adjust the WEC position according to the established control strategy. A review of design options for mooring systems for wave energy converters is presented in Refs. [8, 9].

Mooring systems are traditionally composed of several mooring lines (slack or taut), with one extremity attached to the device, at a point called the fairlead, and the other extremity attached to a point that must be able to handle the loads applied by the device through the line. This point can be fixed to an anchor on the seabed, or moving, e.g. the fairlead on another floating offshore structure. Mooring lines are

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[2]Drift forces are second-order low frequency wave force components. Under the influence of these forces, a floating body will carry out a steady slow drift motion in the general direction of wave propagation if it is not restrained. See further in Chap. 7.

usually composed of various sections of different materials (chain, wired-ropes, polyester, etc.). Some additional elements, such as floats or clump weights can be attached to the line to give it a special shape.

Depending on the objectives of the simulation, the mooring system can be modelled with different levels of accuracy, and thus different computational efforts. Hence, it is important to understand the level of detail and accuracy required in order to select the most appropriate modelling approach. Essentially mooring models may be split into two main categories: quasi-static and dynamic models.

Quasi-static models depend only on the position of the fairlead and the anchor at specific time-step. Therefore, they do not solve differential equations for the motion of the lines, which considerably reduces the required computational effort. Quasi-static models may be split into two types:

- **Linearized mooring model**. The most common quasi-static model is the so-called "linearized mooring model" which consists of modelling the mooring loads in the different directions of motion by a simple spring effect. The computation of the restoring effect is straightforward, but it is only effective when the device has small motions, around its undisturbed position. Whenever this approach is built-in in the wave-to-wire model it is necessary to input the mooring spring stiffness matrix (which is multiplied by the displacement vector at each time-step to define the tension at the fairlead connection).
- **Quasi-static catenary model**. The quasi-static catenary modelling approach consists of computing the tension applied by a catenary mooring line on a device using only the position of the fairlead and the anchor. This mooring modelling approach requires the inclusion of the nonlinear quasi-static catenary line equations in the wave-to-wire model, which are solved at each time-step in order to determine the value of the tension at the fairleads. This modelling approach is very simple and requires little computational effort, but it is only valid for relatively small motions about the mean position.

Quasi-static models are usually reliable to estimate the horizontal restoring effect on a device that experiences small motion amplitudes, but they are not reliable to estimate the effective tension in the line, especially in extreme weather conditions. In this case, dynamic models are necessary to compute the loads in the lines, and thus the restoring effect of the station-keeping system. This feature is available in some commercial modelling software such as OrcaFlex[3] [10] or ANSYS AQWA [11]. Although wave-to-wire tools may be coupled to dynamic mooring models this

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[3]Dynamic models represent the mooring lines by a finite-element description. The equation of motion is solved at each node in order to compute the tension in the line. Consequently, the elasticity and stiffness of the line, the hydrodynamic added-mass and drag effects, and the seabed interactions, among others, can be modelled accurately. The numerical methods implemented in such codes allow making numerical predictions under extreme loads and fatigue analysis of mooring lines possible, however, the computational effort required is in general considerable. The most widespread commercial code available in the market is Orcaflex a user-friendly numerical tool that allows the user to study the most common problems in offshore industry.

is not a common approach. Wave-to-wire tools are designed to model the opera-
tional regime of WECs (i.e. during power production) where a linear (or partially
non-linear) wave/device hydrodynamic interaction is fairly valid. In general, the
nonlinear behaviour of WECs under extreme wave conditions is not properly
represented in wave-to-wire tools. Therefore, combining wave-to-wire and dynamic
mooring models does not allow the full capabilities of the dynamic model to be
exploited. Moreover, usually most WECs need to enter a survival mode (with no
energy production) in extreme wave conditions in order to avoid structural damage
which, to some extent, decreases the usefulness of wave-to-wire models since their
main feature is to assess the energy conversion efficiency.

10.2.5 Radiation Force

In addition to the usual instantaneous forces proportional to the acceleration,
velocity and displacement of the body, the most commonly-used formulations of
time-domain models of floating structures incorporate convolution integral terms,
known as 'memory' functions. These take account of effects which persist in the
free surface after motion has occurred. This 'memory' effect means that the loads on
the wet body surface in a particular time instant are partially caused by the change
in the pressure field induced by previous motions of the body itself. Assuming that
the system is causal, this is, $h(t) = 0$ for $t < 0$, and time invariant[4] these convolution
integrals take the form

$$F_r(t) = \int_0^\tau h(t - \tau)\dot{z}(\tau)d\tau, \qquad (10.9)$$

where $h(t - \tau)$ represents the impulse-response functions (IRFs) or kernels of the
convolutions and $\dot{z}(\tau)$ the body velocity towards any DoF. In the case of 6 rigid
DoFs, $h(t - \tau)$ is a 6×6 symmetric matrix where the off-diagonal entries represent
the cross-coupling radiation interaction between the different oscillatory modes.

Apart from a few cases which may be solved analytically, the IRFs are derived
computationally. The most common method does not involve the direct computa-
tion of the IRFs, but derives the IRFs from the frequency-dependent hydrodynamic
data obtained with standard 3D radiation/diffraction codes (such as ANSYS Aqwa
[11], WAMIT [12], Moses [13] or the open source Nemoh code [14]) generally
used to model WECs.

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[4]A time-invariant system is a system whose output does not depend explicitly on time. This
mathematical property may be expressed by the statement: If the input signal $x(t)$ produces an
output $y(t)$ then any time shifted input, $x(t + \tau)$, results in a time-shifted output $y(t + \tau)$.

The output of these numerical tools includes the frequency dependent added mass, $A(\omega)$, and damping, $B(\omega)$, coefficients along with the added mass coefficient in the limit as the frequency tend to infinity, A_∞ (**see section**: **hydrodynamics**). The IRFs are normally obtained by applying the inverse discrete *Fourier* transform to the radiation transfer function, $H(\omega)$, given by

$$H(\omega) = [A_\infty - A(\omega)] + B(\omega). \qquad (10.10)$$

Usually the direct computation of the convolution integrals is quite time consuming. Therefore, alternative approaches have been proposed to replace the convolution integrals in the system of motion equations, such as implementing a transfer function of the radiation convolution [15], or state-space formulations [16–18].

The state-space formulation, which originated and is generally applied in control engineering, has proved to be a very convenient technique to treat these sorts of hydrodynamic problems. Basically, this approach consists of representing the convolution integral by (ideally) a small number of first order linear differential equations with constant coefficients. For causal and time invariant systems the state-space representation is expressed by

$$\dot{X}(t) = AX(t) + B\dot{z}(t)$$
$$y(t) = \int_0^\tau h(t - \tau)\dot{z}(\tau)d\tau = CX(t), \qquad (10.11)$$

where the constant coefficient array A and vectors B and C define the state-space realization and x represents the state vector, which summarizes the past information of the system at any time instant.

Different methodologies have been proposed to derive the constant coefficients of the differential equations (i.e. the array A and the vectors B and C): (i) directly from the transfer function obtained with standard hydrodynamic 3D radiation-diffraction codes or (ii) explicitly from the IRF (i.e. the Fourier transform of the transfer function). Since typically the time domain modeling of WECs involves the use of 3D radiation-diffraction codes, which give the transfer function as an output, the first alternative is more convenient and is the approach generally used as it avoids additional errors being introduced by the application of the Fourier transform to obtain the IRF.

Next, a parametric model that approximates the transfer function by a complex rational function, computed for a discrete set of frequencies, is run. The most common methodology is based on the so-called frequency response curve fitting, which seems to provide the simplest implementation method (iterative linear least squares [19, 20]). The method provides superior models, mainly if the hydrodynamic code gives the added mass at infinite frequency, because it forces the structure of the model to satisfy all the properties of the convolution terms.

The least squares approach consists of identifying the appropriate order of the numerator and denominator polynomials (rational function) and then finding the parameters of the polynomials (numerator and denominator). The parameter estimation is a non-linear least squares problem which can be linearized and solved iteratively. This operation can be performed using the *MATLAB* function *invfreqs* (signal processing toolbox) which solves the linear problem and gives as output, for a prescribed transfer function, the parameters vector [21]. To convert the transfer function filter parameters to a state-space form the signal processing toolbox of MATLAB includes the function *tf2ss,* which returns the A, B and C matrices of a state space representation for a single-input transfer function.

10.2.6 PTO Force

The simplest way to represent the PTO force involves considering a linear force that counteracts the WEC motion. This force is composed of one term proportional to the WEC velocity and another proportional to the WEC displacement, i.e,

$$F_{pto}(t) = -D\dot{z}(t) - kz(t). \tag{10.12}$$

The first term of Eq. 10.12 is the resistive-force component where D is the so-called damping coefficient. This term refers to a resistive or dissipative effect and is therefore related to the WEC capacity to extract wave energy. Furthermore, the second term of Eq. 10.12, represents a reactive-force proportional to the displacement, where k is the so-called spring coefficient. This term embodies a reactive effect related to the energy that flows between the PTO and the moving part of the WEC. The reactive power is related to the difference between the maximum values of kinetic and potential energy. Ultimately, the reactive-force component does not contribute to the time-averaged absorbed power since the time-averaged reactive power is zero.

To maximize the overall energy extraction (rather than the instantaneous power) it is necessary to continually adjust the characteristics of the control system in order to keep the converter operating at peak efficiency.

Fundamentally there are two main strategies to control WECs: passive control and active control. Passive control is the simplest control strategy as it consists of only applying to the floater an action proportional to its velocity (resistive force) by adjusting the damping coefficient and setting the reactive-force component of the PTO to zero. Conversely, active control requires tuning both PTO parameters, D and K, which, as mentioned above, implies bidirectional reactive power flowing between the PTO and the absorber.

Control of WECs is an intricate matter mostly due to the randomness of ocean waves and the complexity of the hydrodynamic interaction phenomenon between WECs and the ocean waves. Furthermore, an additional difficulty arises from the sensitivity of

optimum control on future knowledge of the sea state (especially in the case of resonant point absorbers) [22]. However, control is crucial to enhance the system performance, particularly in the case of point absorbers where appropriate control strategies, normally highly non-linear, allow the otherwise narrow bandwith of the absorber to be broadened. In this framework the PTO machinery must have the capacity to cope with reactive forces and reactive power. Controlling the PTO reactive-force, so that the global reactance is cancelled [22], is the basis of these so called phase control methods. In this way the natural device response, including its resonant characteristics, are adjusted such that the velocity is in phase with the excitation force on the WEC, which is a necessary condition for maximum energy capture [22].

Several strategies have been suggested in the last three decades, but latching and declutching are the two most commonly used strategies categorized as phase control techniques. Latching control, originally proposed by Budal and Falnes [23], consists of blocking and dropping the captor at appropriate time instants to force the excitation force to be in phase with the buoy velocity, as described above. Extensive research has been developed in this topic, including amongst other researchers Babarit et al. [24]; Falnes and Lillebekken [25]; Korde [26] and Wright et al. [27]. Conversely, declutching control consists of manipulating the absorber motion by shifting between applying full load force or no force, allowing the absorber to move freely for periods of time. Declutching was introduced by Salter et al. [28] and latter extensively investigated by Babarit et al. [29].

The convergence into one, or possibly two or three different WECs, is still an open issue in the wave energy field. Currently there is a wide range of proposed concepts that differ on the working principle, the applied materials, the adequacy of deployment sites, and above all the type of PTO equipment and the control characteristics. Therefore, although the hydrodynamic wave/WEC interaction might be modelled using (to some extent) similar numerical approaches (independently from the technology itself), the development of generic wave-to-wire modelling tools is hampered by the wide variety of proposed PTO equipment and dissimilar control strategies, which require different modelling approaches.

Despite the number of existing PTO alternatives there are some fundamental considerations that may be made about the correlation between the type of PTO and the WEC class. In this regard it can be said that typically the PTO of OWCs consists of a turbo-generator group with an air turbine, whether Wells[5] or self-rectifying impulse turbine.[6] In the case of WECs within the class of wave-induced relative motion there are two main fundamental differences based in the amplitude of the oscillatory motion. In general the working principle of WECs with large captors and

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[5]The Wells turbine is a low-pressure air turbine that rotates continuously in one direction in spite of the direction of the air flow. In this type of air turbine the flow across the turbine varies linearly with the pressure drop.

[6]A self-rectifying impulse turbine rotates in the same direction no matter what the direction of the airflow is, which makes this class of turbine appropriate for bidirectional airflows such as in OWC wave energy converters. In this type of air turbine the pressure-flow curve is approximately quadratic.

so high dynamic excitation loads is based on motions of very small amplitude, which typify the use of **hydraulic systems**. On the other hand, WECs with small captors (i.e. point absorbers), and so lower excitation loads, require high displacements (within certain limits) to maximize the power capture. Those concepts are, by and large, heaving resonant WECs. In this case, the most frequently used PTO equipment is direct-drive **linear generators**, where the permanent magnet and the reluctance machines are the most noteworthy systems [30].

Recently, disruptive PTO systems based on dielectric elastomer generators (DEGs) [31] have been proposed, aiming to achieve high energy conversion efficiencies, to reduce capital and operating costs, corrosion sensitivity, noise and vibration and to simplify installation and maintenance processes. However, these systems are still in a very preliminary development stage. Therefore, as the aforementioned more conventional PTO alternatives still cover most of the technologies under development; a more detailed description of those systems is presented in this section:

- **Hydraulic systems**.

  Hydraulics systems are difficult to typify because they can take many different forms. However, usually hydraulic circuits include a given number of pairs of cylinders, high-pressure and low pressure gas accumulators and a hydraulic motor. Depending on the WEC working principle the displacement of the pistons inside the cylinders is caused by the relative motion between two (or more) bodies or the relative motion between the floater and a fixed reference (e.g. sea bed). A rectifying valve assures that the liquid always enters the high-pressure accumulator and leaves the low-pressure accumulator and never otherwise, whether the relative displacement between bodies is downwards or upwards [32]. The resulting pressure difference between the accumulators, Δp_c, drives the hydraulic motor, so that the flow rate in it, Q_m, is obtained from

  $$Q_m(t) = (N_c A_c)^2 G_m \Delta p_c(t), \tag{10.13}$$

  where N_c is the number of pairs of cylinders, A_c the total effective cross sectional area of a pair of cylinders and G_m a constant. The pressure difference between the accumulators, Δp_c, is given by

  $$\Delta p_c(t) = \phi_h v_h(t)^{-\gamma} - \phi_l \left[\frac{V_0 - m_h v_h(t)}{m_l} \right]^{-\gamma}, \tag{10.14}$$

  where the sub-indices l and h refer to the low and high-pressure accumulators, respectively; ϕ is a constant for fixed entropy (an isentropic process is usually assumed in the modeling process), v is the specific volume of gas, γ the specific-heat ratio for the gas, m is the mass of gas, which is assumed to be unchanged during the process, and V_0 is the total volume of gas inside the accumulators, which also remains constant during the process, so that $V_0 = m_h v_h(t) = m_l v_l(t) = C^{te}$.

The total flow rate in the hydraulic circuit is given by the variation of the volume of gas inside the high-pressure accumulator, which is given by

$$Q(t) - Q_m(t) = -m_h \frac{dv_h(t)}{dt}, \qquad (10.15)$$

where Q is the volume flow rate of liquid displaced by the pistons. The useful power at a given instant, P_u, is, in any case, given by

$$P_u(t) = Q_m(t)\Delta p_c(t). \qquad (10.16)$$

- **Air Turbines**.
  Air turbines are the natural choice for the PTO mechanism of oscillating water columns (OWCs). In essence, OWC wave energy converters consist of hollowed structures that enclose an air chamber where an internal water free surface, connected to the external wave field by a submerged aperture, oscillates. The oscillatory motion of the internal free surface, in bottom fixed structures, or the relative vertical displacement between the internal free surface and the structure, in floating concepts, causes a pressure fluctuation in the air chamber. As a result, there is an air flow moving back and forth through a turbine coupled to an electric generator.

  The Wells turbine is the most commonly used option in OWCs, whose main characteristic is the ability to constantly spin in one direction regardless of air flow direction [33]. Nevertheless, there are other alternatives such as Wells turbines with variable-pitch angle blades [34] and axial [35] or radial [36] impulse turbines. A detailed review of air turbines used in OWCs is described by Falcão and Henrriques in Ref. [37].
  To numerically model OWCs the internal surface is usually assumed to be a rigid weightless piston since the OWC's width is typically much smaller than the wavelengths of interest [38].
  The motion of the water free-surface inside the chamber, caused by the incoming waves, produces an oscillating air pressure, $p(t) + p_a$ (p_a is atmospheric pressure), and consequently displaces a mass flow rate of air through the turbine, $\dot{m}$. This is calculated from

$$\dot{m} = \frac{d(\rho V p)}{dt}, \qquad (10.17)$$

where ρ is the air density and V the chamber air volume. Often, when modeling OWCs it is also assumed that the relative variations in ρ and V are small, which is consistent with linear wave theory. In addition, ρ is commonly related to the

pressure, $p + p_a$, through the linearized isentropic relation, the adequacy of which is discussed by Falcão and Justino [39]. Taking into account the previous assumptions the mass flow rate of air in Eq. 10.17 might be rewritten as

$$\dot{m} = \rho_0 q - \frac{V_0}{c_a^2}\frac{dp}{dt}, \tag{10.18}$$

where q is the volume-flow rate of air, ρ_0 and c_a are the air density and speed of sound in atmospheric conditions respectively, and V_0 is the air chamber volume in undisturbed conditions.

The mass flow rate, $\dot{m}$, can be related to the differential pressure in the pneumatic chamber, p, by means of the turbine characteristic curves. Thus applying dimensional analysis to incompressible flow turbomachinery, yields [39, 40]

$$\Phi = f_Q(\Psi), \tag{10.19}$$

$$\Pi = f_p(\Psi), \tag{10.20}$$

where Ψ is the pressure coefficient, Φ the flow coefficient and Π the power coefficient, given respectively by

$$\Psi = \frac{p}{\rho_0 N^2 D_t^2}, \tag{10.21}$$

$$\Phi = \frac{\dot{m}}{\rho_0 N D_t^3}, \tag{10.22}$$

$$\Pi = \frac{P_t}{\rho_0 N^3 D_t^5}, \tag{10.23}$$

in which ρ_0 is the air density, $N = \dot{\omega}$ the rotational speed (radians per unit time), D_t the turbine rotor diameter and P_t the turbine power output (normally the mechanical losses are ignored).

In the case of a Wells turbine, with or without guide vanes, the dimensionless relation between the flow coefficient and the pressure coefficient, Eq. 10.19, is approximately linear. Therefore Eq. 10.19 may be rewritten in the form $\Phi = K_t\Psi$, where K_t is a constant of proportionality that depends only on turbine geometry. Eventually, the relation between the mass flow rate and the pressure fluctuation can be written as

$$\dot{m} = \frac{K_t D_t}{N}p, \tag{10.24}$$

which is linear for a given turbine and constant rotational speed. The instantaneous (pneumatic) power available to the turbine is then obtained from

$$P_{\text{available}} = \frac{\dot{m}}{\rho_0}p, \tag{10.25}$$

and finally the instantaneous turbine efficiency is given by

$$\eta = \frac{P_t}{P_{\text{available}}} = \frac{\Pi}{\Phi\,\Psi}. \tag{10.26}$$

- **Direct drive linear generators**.
  The most typical applications of direct drive systems make use of rotating motions to convert mechanical energy into electrical energy. Generators in conventional power stations (e.g. coal, fuel oils, nuclear, natural gas), hydro power stations or direct-drive wind turbines all use rotating generators. However, in some particular cases linear generators are also used in applications with high power levels. This is the case of some hi-tech transportation systems, such as magnetic levitation (maglev) trains, and PTO systems for wave energy conversion.

The inherent complexity of extracting energy from waves, and ultimately the main difficulty with using linear generators for wave energy conversion, is related to the intricacy of handling high forces (depending on the size of the wave energy converter) and low speeds. In this context the viability of linear generators is restricted to heaving point absorbers which are characterized by higher velocities (higher that 1 m/s [41]) and lower excitation loads than the majority of the other categories of WEC. Nevertheless, the relevance of this PTO mechanism is highlighted by the large number of projects that have been focused on developing different heaving point absorber concepts equipped with linear generators (e.g. AWS, OPT, Seabased, Wedge Global, etc).

In the context of wave energy conversion there are different types of conventional linear generator that may be used. Namely

- Induction machines
- Synchronous machines with electrical excitation
- Switched reluctance machines
- Longitudinal flux permanent magnet generator.

Among these types of linear generators longitudinal flux permanent magnet generators (LFPM) have been the most common choice [41–43] for wave energy conversion. Normally, LFPM machines are also called permanent-magnet synchronous generators, as the armature winding flux and the permanent magnet flux move synchronously in the air gap. These machines have been extensively investigated for wave energy applications by Polinder and Danielsson [43, 44] amongst other researchers.

Figure 10.2 shows the cross-section of the magnetic circuit of a LFPM generator. The magnetic flux (indicated in Fig. 10.2 with dashed lines and its direction with arrows) from one magnet crosses the air gap and is conducted by the stator teeth through the stator coils. Then the flux is divided into two paths in the stator yoke and returns all the way through the stator teeth, crossing the air gap and through the adjacent magnets. The permanent magnets on the translator are mounted with alternating polarity, which creates a magnetic flux with alternating direction.

Fig. 10.2 Cross-section of a LFPM generator where the magnetic flux path is illustrated with *dashed lines* [45]
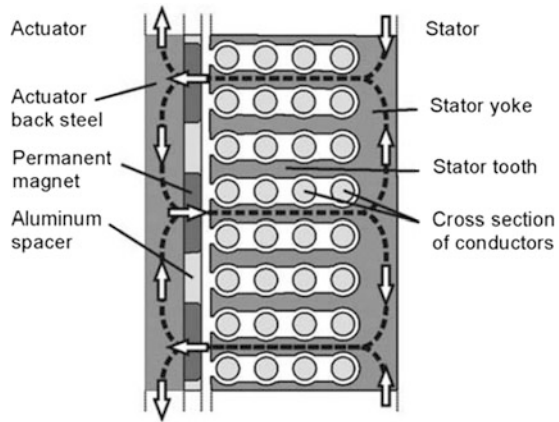
The relative motion between the stator and translator induces an electromotive force *emf* in the armature windings which drives a current whenever the armature winding is coupled to a load. In single body heaving point absorbers the translator is normally connected to the floater and the stator fixed to the sea bed, such as for the *Seabased* concept [46]. In the case of two body heaving concepts, the most common configurations have the stator attached to a submerged body and the translator connected to the floater. In turn, the current produced creates a magnetic flux that interacts with the flux of the permanent magnet leading to a force on the translator. In this way the floater mechanical energy is converted into electric energy consumed in the load.

From Faraday's law of induction the electromotive force emf, *E*, i.e. the voltage induced by the permanent magnet flux, may be written as

$$E = \omega \phi N, \tag{10.27}$$

where ω is the angular frequency, ϕ is the permanent magnet induced flux per pole and N is the total number of coil turns. The angular frequency is given by

$$\omega = 2\pi \frac{u_r}{w}, \tag{10.28}$$

in which u_r is the relative vertical speed between stator and translator and w the distance between the poles (i.e. the pole pitch). Simultaneously, there is also a

resistive voltage drop in the slots, the end windings and cable connections when the generator is loaded. This resistive voltage drop per unit of length of the conductor is given by

$$E = I\rho_{cu},$$ (10.29)

where ρ_{cu} is the resistivity of the conductor material (mostly copper) and I is the current density in the conductor. As a result the induced phase currents produce a magnetic field, divided into two components: one component is coupled to the entire magnetic circuit, i.e. the main flux, and the other component is leakage flux. The corresponding inductances are then defined accordingly as the main inductance, L_m, and the leakage inductance, L_l. In a symmetric system the synchronous inductance, L_s, expressed in terms of the main inductance and the leakage inductance, is given by

$$L_s = \frac{3}{2}L_m + L_l,$$ (10.30)

where the first term is the armature flux linkage with the phase winding, which will be described below, and the second term is leakage inductance of that phase.

In a simplistic way the main electrical characteristics of a LFPM generator may be described using a lumped circuit as illustrated in Fig. 10.3 for a single phase of the generator. A single phase might be then modelled by an electromotive force, E, (voltage induced by the permanent magnet flux), a resistance inside the generator, R_g, a inductive voltage modelled by the synchronous inductance, L_s, and a load resistance R_l (the load might be either purely resistive or may also have a reactive component).

Fig. 10.3 Lumped circuit diagram of one phase of a synchronous generator
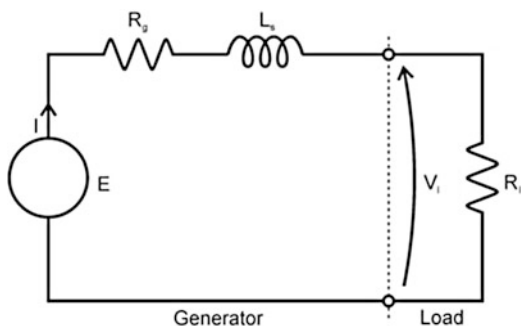
From the lumped circuit we can determine the load voltage given by

$$V_l = \frac{ER_l}{R_l + R_g + i\omega L_s},$$ (10.31)

the phase current by

$$I = \frac{E}{R_l + R_g + i\omega L_s},$$ (10.32)

and finally the power in the load is obtained from

$$P = \frac{E^2 R_l}{\left(R_l + R_g\right)^2 + \left(\omega L_s\right)^2}.$$ (10.33)

Regardless of the type of electrical machine there are fundamentally two main electromagnetic forces: the normal force, attracting the two iron surfaces, and the thrust force, acting along the translator, in the longitudinal direction in linear machines or tangential to the rotor surface in the case of rotating generators. The corresponding sheer, τ, and normal, σ, stresses are given respectively by

$$\tau = \frac{BA_e}{2}$$ (10.34)

and

$$\sigma = \frac{B^2}{2\mu_0},$$ (10.35)

where B is the air gap magnetic flux density (the SI unit of magnetic flux density is the Tesla, denoted by T), A_e is the electrical loading, measured in amperes per metre (A/m), and μ_0 the magnetic permeability of free space, also known as the magnetic constant, measured in henries per meter (H·m^{-1}), or newton per ampere squared (NA^{-2}). Typically the shear force density, Eq. 10.34, is limited in linear machines, since the air gap flux density is limited by saturation and cannot be increased substantially in conventional machines. Moreover, the electrical loading is also limited because current loading produces heat, and heat dissipation is by and large a drawback in conventional machines. Heat dissipation can be increased to a certain extent by improving thermal design (e.g. water cooling system), but it would not be expected to increase massively.

Besides the technical requirements for operating in irregular sea conditions with very high peak forces and relatively low speeds, the design of LFPM generators has a few additional complexities related to

(i) The design of the bearing system, which is quite intricate due to the high attractive force between translator and stator.
(ii) The mechanical construction with small air gaps. The stator construction of LFPM generators is simple and robust, however typically the air gap between the stator and the rotor has to be reasonably large, which reduces the air gap flux density and so the conversion efficiency. Essentially, the size of the gap

is imposed by manufacturing tolerances, the limited stiffness of the complete construction, large attractive forces between stator and translator, thermal expansion, etc.

(iii) The power electronics converter to connect the WEC voltage (which has varying frequency and amplitude caused by the irregular motion and continuously varying speed) to the electric grid (which has fixed frequency and amplitude).

(iv) The geometry of LFPM, however, limits the stator teeth width and cross-section area of the conductors for a given pole pitch. Increasing the tooth width to increase the magnetic flux in the stator or increasing the conductor cross-section demands a larger pole pitch and the angular frequency of the flux is thus reduced. This sets a limit for the induced emf per pole and consequently the power per air pat area.

10.2.7 End Stops Mechanism

End stops are mechanisms to restrict the stroke of the WEC moving bodies in order to restrain the displacement within certain excursion limits for operational purposes, depending on the WEC working principle. End stops mechanisms are particularly important in concepts operating at high velocities (e.g. heaving point absorbers). Virtual end stops may be incorporated in wave-to-wire models either as an independent additional force, representing a physical end stop, or included in the controller in order to avoid the bodies reaching the physical end stop, or to reduce the impact when limits are reached. Control methods for handling this kind of state saturation problem consist of adding spring and/or damper (to dissipate excessive power) terms to the calculation of the machinery force set-point. For instance, this additional force may be obtained from

$$F_{es}(t) = R_m\dot{\eta} - sign(\dot{\eta})K_{es}(|\eta| - \eta_{\lim})H(|\eta| - \eta_{\lim}) - D_{es}\dot{\eta}u(|\eta| - \eta_{\lim}), \quad (10.36)$$

where H is the Heaviside step function and K_{es} and D_{es} are the spring and damping constants for the end stop mechanism. The constant $\eta_{\lim}$ represents the excursion for which the mechanism starts acting [47].

10.3 Benchmark Analysis

This section presents a benchmark on existing wave-to-wire models and other modeling tools, such as CFD codes, based on the Reynolds-Averaged Navier-Stokes equation (RANSE). At present CFD codes are not the most suitable tools to model the entire chain of energy conversion (at least in a

Table 10.1 Benchmark on existing WEC modeling tools

Developer	Code name	Fluid model	Hydro model	Classes of WECs	PTO model	Multi-body	Accuracy	CPU time	Sea states		
									Small to moderate	Moderate to extreme	Severe
DNV—GL[1]	Wavedyn	Perfect fluid	Linear PFT	Moving Bodies	Non-linear	√	+	+++	√	X	X
Imosea[2]	Inwave	Perfect fluid	Partially nonlinear PFT	OWC Moving Bodies	Non-linear	√	++	++	√	√	X
ECN[3]	LAMSWEC	Perfect fluid	Partially nonlinear PFT	Moving Bodies	Non-linear	X	++	++	√	√	X
ECN	ACHIL3D	Perfect fluid	Linear PFT	Moving Bodies	Non-linear	√	+	+++	√	X	X
Sandia/NREL[4]	WEC—Sim	Perfect fluid	Linear PFT	Moving Bodies	Non-linear	√	+	+++	√	X	X
Principia[5]	Diodore	Perfect fluid	Linear PFT	Moving Bodies	Non-linear	X	+	+++	√	X	X
WavEC[6]	WavEC2wire	Perfect fluid	Linear PFT	OWC Moving Bodies	Non-linear	√	+	+++	√	X	X
Marin[7]	Refresco	Viscous fluid	RANSE	Moving Bodies	N/A	√	+++	+	√	√	√
ECN	Icare	Viscous fluid	RANSE	Moving Bodies	N/A	√	+++	+	√	√	√

[1] www.gl-garradhassan.com
[2] www.innosea.fr
[3] www.ec-nantes.fr
[4] www.energy.sandia.gov
[5] www.principia.fr
[6] www.wavec.org
[7] www.marin.nl

straightforward way) and evaluate different control strategies to enhance the device performance. Nevertheless CFD codes might be extremely useful to study flow details of the wave-structure interaction (e.g. detection of flow separations, extreme loading and wave breaking).

The main differences between the codes listed in Table 10.1 reside in the theory they are based on. For instance, modelling tools based on linear potential flow theory (PFT) are not very time demanding (especially when compared with CFD codes), although they allow the representation of a non-linear configuration of the PTO mechanism, which is the most realistic scenario for the majority of wave power devices. However, these tools have a rather limited range of applicability and fairly low accuracy, largely due to the linear theory assumptions of small waves and small body motions.

Consequently, these limitations make the modelling tools based on linear potential flow theory inadequate to assess WEC survival under extreme wave loading or even throughout operational conditions when the motion of the captor is not of small amplitude. In order to overcome these limitations various models include some nonlinearities in the hydrodynamic wave-structure interaction. The most common approach consists of computing the buoyancy and Froude-Krylov excitation forces from the instantaneous position of a WEC device instead of from its mean wet surface, as considered in the traditional linear hydrodynamic approach. The major advantage of these partially nonlinear codes is widening the range of applicability from intermediate to severe sea-states.

10.4   Radiation/Diffraction Codes

Usually wave-to-wire models rely on the output from 3D radiation/diffraction codes (such as ANSYS Aqwa [11], WAMIT [12], Moses [14] or the open source Nemoh code [13]), which are based on linear (and some of them second-order) potential theory for the analysis of submerged or floating bodies in the presence of ocean waves. These sort of numerical tools use the boundary integral equation method (BIEM), also known as the panel method, to compute the velocity potential and fluid pressure on the body mean submerged surface (wetted surface in undisturbed conditions). Separate solutions for the diffraction problem, giving the effect of the incident waves on the body, and the radiation problems for each of the prescribed modes of motion of the bodies are obtained and then used to compute the hydrodynamic coefficients, where the most relevant are:

Added-Mass Coefficient:
The added mass is the inertia added to a (partially or completely) submerged body due to the acceleration of the mass of the surrounding fluid as the body moves through it. The added-mass coefficient may be decomposed into two terms: a frequency dependent parameter which varies in accordance to the frequency of the sinusoidal oscillation of the body and a constant term, known as the infinite added

mass, which corresponds to the inertia added to the body when its oscillatory motion does not radiate (generate) waves. This is the case when the body oscillates with "infinite" frequency or when it is submerged very deep in the water.

Damping Coefficient:

In fluid dynamics the motion of an oscillatory body is damped by the resistive effect associated with the waves generated by its motion. According to linear theory, the damping force may be mathematically modelled as a force proportional to the body velocity but opposite in direction, where the proportionality coefficient is called damping coefficient.

Excitation force coefficient:

According to linear theory the excitation coefficient is obtained by integrating the dynamic pressure exerted on the body's mean wetted surface (undisturbed body position) due to the action incident waves of unit amplitude, assuming that the body is stationary. The excitation coefficient results from adding to the integration of the pressure over the mean wetted body surface, caused by the incident wave in the absence of the body (i.e. the pressure field undisturbed by the body presence), a correction to the pressure field due to the body presence. This correction is obtained by integrating the pressure over the mean wetted body surface caused by a scattered wave owing to the presence of the body. The first term is known as the Froud-Krylov excitation and the second the scattered term.

10.5 Conclusion

Wave-to-wire models are extremely useful numerical tools for the study of the dynamic response of WECs in waves since they allow modelling of the entire chain of energy conversion from the wave-device hydrodynamic interaction to the electricity feed into the electrical grid, with a considerable high level of accuracy and relatively low CPU time. Wave-to-wire models allow the estimation of, among other parameters, the motions/velocities/accelerations of the WEC captor, structural and mooring loads, and the instantaneous power produced in irregular sea states. Therefore, these types of numerical tools are appropriate and widely used to evaluate the effectiveness of and to optimize control strategies.

Despite the usefulness of wave-to-wire models it is, however, important to bear in mind that they have some limitations that mostly arise from the linear wave theory assumptions which are usually considered in modelling the hydrodynamic interactions between ocean waves and WECs (e.g. linear waves, small response amplitudes). Although these assumptions are fairly acceptable to model the operational regime of WECs, which comprises small to moderate sea states, they are not appropriate to model the dynamic response of WECs under extreme conditions. Nevertheless, some sort of non-linear hydrodynamic modelling approaches might be included in wave to wire models (which extends the applicability of the model), such as the evaluation of the hydrostatic force at the instantaneous body position

instead of at its undisturbed position and/or the non-linear description of the Froud-Krylov term in the excitation force [48]. Ultimately, it is possible to trade off accuracy and CPU time by choosing the partial non-linear hydrodynamic approach for better accuracy, or the linear approach for faster computation.

Wave-to-wire models might be also used for modelling wave energy farms instead of single isolated devices. For this purpose the model must consider additional forces on each device resulting from the waves radiated from the other devices in the wave farm. Obviously this hydrodynamic coupling effect significantly increases the CPU time. Some simplification may be considered for faster computation however, such as neglecting the effect of remote WECs, the radiation force from which tends to be irrelevant when compared with that caused by neighbouring WECs. Moreover, the farm size and the hydrodynamic coupling between the WECs manifests an additional difficulty since it makes the application of BEM codes to generate the inputs required by wave-to-wire models (matrices of hydrodynamic damping and added mass) more time consuming.

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