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Generation of a set of typical dynamic load regimes for common conversion devices

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ABOUT MARINET

MARINET (Marine Renewables Infrastructure Network for emerging Energy Technologies) is an EC-funded network of research centres and organisations that are working together to accelerate the development of marine renewable energy - wave, tidal & offshore-wind. The initiative is funded through the EC’s Seventh Framework Programme (FP7) and runs for four years until 2015. The network of 29 partners with 42 specialist marine research facilities is spread across 11 EU countries and 1 International Cooperation Partner Country (Brazil).

MARINET offers periods of free-of-charge access to test facilities at a range of world-class research centres. Companies and research groups can avail of this Transnational Access (TA) to test devices at any scale in areas such as wave energy, tidal energy, offshore-wind energy and environmental data or to conduct tests on cross-cutting areas such as power take-off systems, grid integration, materials or moorings. In total, over 700 weeks of access is available to an estimated 300 projects and 800 external users, with at least four calls for access applications over the 4-year initiative.

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The initiative consists of five main Work Package focus areas: Management & Administration, Standardisation & Best Practice, Transnational Access & Networking, Research, Training & Dissemination. The aim is to streamline the capabilities of test infrastructures in order to enhance their impact and accelerate the commercialisation of marine renewable energy. See www.fp7-marinet.eu for more details.

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Deliverable Leader: Pierpaolo Ricci TECNALIA

Contributing Authors: Joseba López TECNALIA
Francesco Bóscolo TECNALIA

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EXECUTIVE SUMMARY

Power Take-Off systems for marine energy technologies are of different types and, as such, their design is usually device-specific and very much dependent on the applicable conditions, the rated power and the device dynamics. Thus, a proper recommendation for testing schedule of a PTO system should necessarily require a preliminary detailed design and assessment which could not be part of this work. On the other hand, a careful dynamic analysis of all the available device configurations would be unsuitable and possibly not very useful since it is likely that any realistic design would involve the definition of purposely-developed control strategies and would have to meet specific requirements and challenges.

In this deliverable we present a simple approach for the definitions of the most relevant dynamic loads from marine energy devices. The method is based on the numerical modelling and simulation of global ocean energy technologies with a time-domain analysis (wave-to-wire or current-to-wire models), a necessary tool because of the highly non-linear behaviour that characterises most of the real PTO systems. The hydrodynamics of wave energy converter is modelled by using linear water wave theory and panel codes. Additional terms accounting for viscous losses, possibly validated against preliminary tank testing, might be included at a second stage. Tidal turbines can be modelled by applying Blade Element Momentum theory. However, examples on the dynamic analysis of tidal machines are not presented in this document and might be added at a later stage. Different device concepts and PTO models are available, depending on the actual system under analysis. Within this document simple linear and Coulomb-type PTO systems are proposed and compared with a more realistic hydraulic PTO which is capable of applying different control strategies.

Five fundamental dynamic conditions are proposed for further investigation, based on the environmental conditions occurring at a selected site (the bimep, in this case). The first two refer to supposedly optimal loading conditions, where the system is expected to be most efficient and they are defined by the most occurring and the most energetic sea state. A third dynamic load regime corresponds to a minimal loading situation, coincident with the lower bound of the operational range of the machine. The system will have to be analysed also in extreme loading conditions, i.e. in the worst environmental conditions where it is still required to produce electric power. Finally, a fifth condition can be devised based on very severe weather conditions that might constitute a danger for the safety of the equipment and a possible source of damage. In this case, specific control strategies and/or hardware might have to be included to assure the preservation of its energy extraction capabilities.

Those five dynamic situations have been numerically simulated and analysed for the case of a heaving point absorber connected to a hydraulic, a linear or a Coulomb PTO. The results show that the application of simplified PTO configurations might be accurate enough for the purpose of producing dynamic loading time series. Reference values for motions, velocities, forces and hydrodynamic power are proposed for all the cases. It is noticed how, in some situations, the definition of the design parameters should be adjusted to meet the requirements specified by suppliers. Two simple control strategies are applied for the case of a hydraulic PTO. The application of a PI control seems to be much more efficient in terms of preserving a better operational regime for the electric generator.
1 INTRODUCTION

1.1 ENERGY CONVERSION IN MARINE RENEWABLE TECHNOLOGIES

Any marine energy converter presents a conversion system which transforms the marine energy absorbed by the device into another form of energy which is usable for other purposes. Even though some concepts have been proposed to produce fresh water (see for instance [1] for a proposed evolution of the Duck concept for this purpose), a large majority of the technologies currently being developed are electricity-generating and therefore require the introduction of some type of electrical generator.

The usual procedure applied to produce electrical power is using conventional high-speed rotary electrical generators, which are almost standard for many other energy applications and are relatively reliable components in comparison with other elements of the energy chain. Synchronous generators (either Permanent Magnet or field would) are typically in use in conventional power stations, where it is possible to control the power output by maintaining a virtually constant rotational speed. In marine energy technologies, however, the requirement of constant rotational speed can only be met by applying large storage mechanisms (or conversion systems with an inherently large inertia) and it is likely that generators are required to operate at variable speed, particularly if taking into account the needs for sophisticated control strategies to improve power absorption (as is the case of many wave energy technologies). For such purposes, common squirrel cage induction generators or doubly-fed induction generators (DFIG) are often adopted. A review of different rotational generator options for a specific type of wave energy device is discussed in [2] whereas a general outline is also given in [3].

Most of the existing marine renewable technologies are composed of one or several prime movers which are put in motion by the action of the water. In such cases, energy transmission to rotational generators often require intermediate steps for conversion involving either mechanical or hydraulic system to transform a variable bi-directional signal into a steady one-directional one. This is specifically the case of many wave energy technologies, whereas tidal devices based on either horizontal or vertical axis turbines are more similar to the systems used in wind energy and, as such, might allow direct-drive connection to rotational generator. Alternatively, gearboxes are interposed between the turbine and the generator.

![Figure 1.1 Power Take-Off typologies for wave energy conversion](image)

Direct-drive conversion has been proposed also for wave energy devices based on translating prime movers. In this case, the most common solution is the introduction of a permanent magnet linear generator which permits a much
simpler and virtually more efficient system although power electronics and possibly additional storage elements will have to be provided to guarantee grid-connection.

Even though Power Take-Off systems for marine energy technologies are diverse and very specific depending on the device they are installed on, it is possible to attempt a rough classification of different typologies that present separate challenges and requirements from the point of view of their design and operation.

For tidal devices, although some concepts with a distinctly different configuration have been proposed, it can be said (see [6]) that the large majority of the technologies currently being developed are based on a type of turbine which is either connected to a gearbox or direct-driven by a rotational generator.

For wave energy, on the contrary, the existing diversity of concepts and designs make very difficult the task of identifying common basis for PTO systems. A basic classification derived by an analysis of the existing technologies is presented in [7] and [8] and is summarised in Figure 1.1. It is customary to make a distinction between pneumatic, hydraulic, mechanical and direct-driven PTOs with the former three generally employing a more conventional rotational generator at the final step of the energy chain.

Not all of these PTOs are applicable to any kind of device. For instance, air turbines are almost solely utilised in Oscillating Water Column devices whereas overtopping devices always present hydraulic turbines as conversion mechanisms. A review of the existing marine energy concepts and their PTOs can be found in the previous deliverable [3].

Within this document, we will limit the representation of a few case studies to a short list of the most common conversion systems and device configurations that will be presented in §4. Additional considerations on other types of technologies are briefly summarised also in §2

1.2 The Design of Power Take-Off Devices: Requirements and Challenges

It is clear that the design of the power take-off equipment is one of the most fundamental phases of the development of marine energy technologies since its efficiency and reliability directly affects the actual feasibility of the technology itself.

Unfortunately, these PTO systems are most often novel concepts which require the design and testing of new components for which previous experience is not available and a supply chain is hardly in place. Even in cases that allow the adoption of commercially available solutions, the actual estimation of the efficiency and reliability of the PTO is particularly difficult because of the peculiarity of conditions derived by the marine environment and the different operational loads and velocities that the system is required to withstand.

It is no surprise, therefore, that the majority of failures occurred to date in marine energy occurred on the PTO system and a substantial amount of research is still required to address this problem effectively, particularly in what refers to the testing of realistic conditions that are representative of the real sea environment.

From the point of view of the requirements, the basic functions that a PTO is expected to satisfy are:

- Power transmission over a relatively large range of conditions
- High efficiency possibly over most of the operational conditions defined
- High reliability to assure continuous energy production and limited downtime
- Ease of maintenance
- Preferably storage capabilities to deal with power qualities issues and reduce the requirements in terms of electronic equipment
- Minimal environmental impact and damage in case of failure

Nowadays, the requirements on reliability and easy maintenance are perhaps the most challenging to meet and extensive testing programs are definitely needed to provide adequate solutions. Moreover, those solutions will typically be device- and site-specific making it difficult to organise a general set of recommendations valid for all the technologies.

However, any conversion system will require, at some point, a preliminary definition of a range of operational conditions on which it is expected to perform. This set of conditions will likely be dependent on the technology and the deployment site but it should be possible to provide some understanding by analysing the actual dynamics of
some general device types in some specified environmental conditions, either by performing small-scale tank testing or numerical analysis based on mathematical models for the hydrodynamic behaviour of the device. Thus, PTO design and optimisation might start at the level of small-scale testing at specific testing rigs that would allow the assessment of different load conditions by using as input the dynamic loading regimes defined previously. Clearly, the definition of special conditions and challenges would require a detailed analysis of the actual technology and PTO under study. For example, one might be interested to analyse the end-stop issue related to translating wave energy devices which is peculiar of this kind of technologies and might require purposely defined solutions. Care should be taken also in considering control strategies, particularly when they are required to improve power absorption or assure survivability of the equipment. Even though some general theoretical considerations on control strategies can be made at a very high level (see [9] for a brief outline of the problem of the control in wave energy), it is clear that practical aspects of any control implementation can be analysed only by focussing on a specific PTO technology.

1.3 OPERATIONAL CONDITIONS OF PTO SYSTEMS

Since marine energy finds itself at a very early stage, it is not easy to find information on the range of operational conditions on which PTO systems do actually work and how their efficiency varies depending on them. Most of the developers claim that their device will be able to produce energy above a specified value of the applicable environmental input parameter (current velocity for tidal turbines, significant wave height for wave energy devices).

Power production is generally limited to an upper bound dependent on the rated power of the generator. It is likely that marine energy technologies will apply working strategies similar to the ones applied in wind energy, i.e. allowing continuous power production above a lower bound of either current velocity or significant wave height and then limiting the power production to the rated limit for velocity or wave height larger than a specified upper bound. In very severe conditions, however the large forces and motions occurring on floating devices might determine, in some cases, the need for safety mechanism that prevent ruptures and serious damages to the PTO components. In those situations, the application of safety brakes (which dissipate energy through friction) or other mechanisms to avoid the occurrence of extreme forces and pressures might lead to the definition of additional operation modes, that we might call “survivability modes”.

This is particularly true if the PTO is required to provide some additional function other than the sole power transmission, such as, for example, the damping of the motion to assure structural safety. Again, in this case, specific control strategies might have to be defined, similarly to what is done, for example, in wind energy in very large wind velocity conditions where the blade pitch angle is controlled to reduce the loads on the rotor.
2 CHARACTERISATION OF OCEAN ENERGY DYNAMICS: MODELLING APPROACHES AND ASSUMPTIONS

2.1 HYDRODYNAMICS OF OCEAN ENERGY DEVICES: BASIC APPROACHES

A typical approach for concept proof and optimization of a marine energy device is the mathematical and numerical modelling of its dynamics and interactions with the environment. Due to the existing differences between the various concepts, neither standard theoretical assessment procedures nor globally applicable computational tools have been identified for wave or tidal technologies. Marine energy technologies can be categorised depending on the operating principle responsible of the energy extraction. Wave energy technologies are usually divided between:

- Oscillating Water Columns (OWC)
- Overtopping Devices
- Wave Activated Bodies

Tidal stream devices are instead classifiable in four groups:

- Horizontal axis systems
- Vertical axis systems
- Variable foil systems
- Venturi based systems

The assumptions and approaches used to model each of this kind of devices are generally different and might be device-specific even inside a defined category. Some common modelling bases, however, can be found taking into account the interaction between the converter and the sea, which is modelled under the same assumptions in several cases. Main references for the modelling of marine energy devices are the books from Newman ([10]) and Faltinsen ([11]). Fundamental theoretical and analytical results for wave energy have been given by Falnes ([12]). The modelling of many tidal technologies can also be linked to marine propellers technology ([13]).

Theoretical formulation for loads and motions of marine energy converters are available only for simple geometries and flows. With the constant development of more and more powerful computer, numerical computation of these quantities has become consistently quick and reliable. In the recent years many commercial computational codes have been released for the simulation of fluid phenomena.

Typically for floating wave energy devices the hydrodynamic analysis is based on the potential flow theory and linear water wave theory with the subsequent definition of proper hydrodynamic coefficients for the estimation of the wave loads. These coefficients can be found in literature for simple cases but are more usually computed with the use of Boundary Element Method software packages, such as WAMIT ([14]) or AQWA ([15]). The estimation of the performance and the analysis of the dynamics are carried out through two different types of approaches: frequency-domain and time-domain methods. The former is relatively easy to implement and is based on the assumption of linear relationship between motion amplitudes and loads. It is usually applied on a preliminary modelling phase when a quick evaluation of a configuration is needed. The latter makes use of the Cummins equation ([16]) and allows non-linear models with possibility of taking into account a detailed Power Take-Off representation and a complete wave-to-wire model.

The choice of the adequate modelling approach appears to be, therefore, still strongly dependent on the technology considered. The same applies to other kind of wave energy devices such as OWCs for which the numerical modelling approach is usually similar to the one applied to floating devices. Different approaches are required for overtopping devices for which characterisation relies more on physical modelling and tank testing.

On what concerns tidal current turbines, a lot can be learned from the technology transfer from wind turbines and ship propellers: The basic performance of a marine current turbine, like a wind turbine, can be modelled satisfactorily using blade element momentum (BEM) theory (see [17]) but many uncertainties are due to the account for turbulence effects in unsteady flow conditions ([18]) and the possible interaction with surface waves, as mentioned previously about the resource assessment. The sensitivity of performance and loading of marine current turbines to these effects has been subject of recent studies such as the one from McCann ([19]). However, much of
the ‘generic’ work on horizontal axis tidal stream devices concerns open-bladed horizontal axis device (such as MCT) and enclosed blade-tip devices (Clean current, OpenHydro, Lunar, etc) have received less attention.

2.2 THE DYNAMICS OF OCEAN ENERGY DEVICES: METHODOLOGIES

Hydrodynamic optimization of wave energy concepts is usually easily carried out by the mean of a frequency-domain analysis. To prove the validity of a PTO design or a control strategy, however, a more realistic approach is often needed, particularly when real sea-state performance is the determining criterion to validate a solution. In such cases, a time-domain model is generally unavoidable.

When nonlinear effects such as viscous forces or other interactions are considered, the linearity assumption is no longer valid. The most general and fundamental approach to deal with these cases is a full nonlinear time-domain analysis, that requires the solution of the flow equation in the time domain possibly with complex and time-consuming CFD codes that, at the current stage, are really difficult to implement and not always accurate enough.

Another way to take into account nonlinearities, particularly when they can be modelled as time-varying coefficients of a system of Ordinary Differential Equations (ODEs), is to apply the linear time-domain model based on the Cummins equation, whose use is widespread in seakeeping applications. This is based on a vector integro-differential equation which involves convolution terms responsible for the account of the radiation forces.

For the case of a single body floating in heave, the Cummins equation can be expressed in the form:

\[
(M + A_{33e})\ddot{x}(t) + \int_{-\infty}^{t} K_{33}(t - \tau)\dot{x}(\tau)d\tau + \rho g S x(t) + F_{ext}(x, t) = F_e(t)
\]  \hspace{1cm} (2.1)

where \(M\) is the mass of the floater, \(A_{33}\) is the added mass at infinite frequency, \(K_{33}\) is the Radiation Impulse Response Function (RIRF), \(\rho\) is the density of sea water, \(g\) the gravity acceleration, \(S\) the waterline area and \(F_e\) is the wave excitation force.

In this formulation, all the possible nonlinearities are included in the term \(F_{ext}\), which represents the external forces that are applied to the system due, for example, to the PTO or to the moorings and that could be possibly linked to other independent variables that form a set of ODEs.

We notice that the excitation force in the time-domain is formally related to its frequency-domain expression by an inverse Fourier Transform which is, in general, not causal.

For regular monochromatic waves, we can express the excitation force as

\[
F_e(t) = |\Gamma(\omega)|A\cos(\omega t + \phi)
\]  \hspace{1cm} (2.2)

where we have introduced an excitation force coefficient \(\Gamma\) which is proportional to the wave amplitude. In irregular waves, the excitation force can be simply modelled as a linear superposition of \(N\) independent sinusoidal components (ideally, \(N \rightarrow +\infty\)) such as:

\[
F_e(t) = \sum_{i=1}^{N} |\Gamma(\omega_i)|A_i\cos(\omega_i t + \phi_i)
\]  \hspace{1cm} (2.3)

The amplitudes \(A_i\) of each frequency component are defined from the energy spectral density \(S(\omega)\) as Rayleigh distributed random values with mean square \(2S(\omega)\Delta\omega\). Phases \(\phi_i\) are randomly selected assuming a uniform distribution within \([0,2\pi]\). With this assumption, the randomness of the elevation process is properly reproduced and its statistical properties are correctly modelled.

The time-varying RIRF can be derived by a number of methods, including linear time-domain BEM codes or indirectly by firstly solving the linear problem in the frequency domain with the same tools outlined in the previous section and then using the computed frequency-dependent hydrodynamic coefficients in the equation:
The integral in equation has to be evaluated after a truncation at a properly defined frequency. Since the radiation damping coefficient $B(\omega)$ tends asymptotically to zero as the frequency tends to infinity, it is sufficient to introduce an upper limit for this coefficient to be negligible. For instance, taking as the truncation frequency the one above which $B(\omega)$ is less than one thousandth of its maximum value produces satisfactory accuracy.

To solve directly equation (2.1) we need to find the solution of the convolution integral at every time instant, which is possible, once the RIRF is known, by performing a numerical integration along the previous time history. If we use classical ODE single-step numerical methods for the direct integration of the Cummins equation, this operation can be implemented inside the numerical algorithm but turns out to be computationally demanding, especially if many degrees of freedom are considered. Moreover, in many applications, such as control theory, a state-space representation is usually preferable since it is suited to many analysis tools used in automatic control.

There has been a lot of work, therefore, done by many researchers, in order to define solution methods that could allow to get rid of the convolution integral. A review of these methods together with an analysis of the numerical error involved in their application is given in [20]. On the following we will apply the Prony method as introduced in [21] in all the numerical simulations shown in this deliverable.

### 2.3 Examples of Dynamic Models

Based on the approaches summarised before, we present hereafter some examples for the definition of the dynamics of typical marine energy converters.

The objective of these models is the realization of preliminary numerical simulations to identify a set of suitable time series of dynamic loads to be applied as inputs for either numerical assessment or testing of detailed PTO configurations.

Since the approach carried out in this document is rather general and no specific device design has been defined at the beginning, it is clear that these models are rather simple and are mostly useful for a first assessment and sensitivity study.

More advanced approaches might be required when considering a detailed PTO configuration.

#### 2.3.1 Wave-Activated Bodies (WAB)

A vast diversity of wave energy concepts can be assumed to pertain to the category of wave-activated bodies. The basic principle underlying this definition is the fact that those devices extract wave energy by putting in motion one or more bodies following the free surface motion.

The primary step is therefore the conversion of wave energy into kinetic energy which is subsequently transformed in another form depending on the type of PTO employed. An understanding of the hydrodynamics of these devices is absolutely necessary to be capable of modelling the dynamics of the prime movers.

Perhaps the simplest example of WAB device is a point absorber heaving with respect to the seabed, as shown on the left of Figure 2.1. This device can be modelled by a single degree-of-freedom model as outlined in the previous sections.

Alternative concepts might include a two-body system as seen on the right of Figure 2.1, whose model would require at least two degrees of freedom for the account of the dynamics of the two bodies. Multiple-body systems could be based also on the relative motion in other degrees of freedom, such as the rotation with respect to the transverse axis as seen in Figure 2.2.

Since the basic input to the PTO is represented by forces and motions of the prime movers, several types of conversion mechanism are applicable to these devices.

High-pressure oil-hydraulic systems are particularly suitable to convert energy from very large forces acting on relatively slowly moving bodies, as is the typical case of WABs. They can also include gas accumulators that work as storage units and allow smoothing the power output from the hydraulic motor.
Alternative hydraulic designs apply water as working fluid instead of oil and utilise high-head hydraulic turbines instead of volumetric hydraulic motors. These configurations have been applied in particular on near-shore devices which perform the conversion to electricity onshore (such as the Oyster).

Mechanical conversion systems capable of converting alternating translating motion into a continuous rotation have been proposed as well. Though their efficiency can be very high, the need for constant maintenance has reduced their applicability in real seas and only a few developers are still considering them as an option.

![Image of point-absorber technologies](image1)

**Figure 2.1 Example of point-absorber technologies (seabed-referenced on the left and two-body system with submerged mass reference on the right)**

![Image of attenuator-type device](image2)

**Figure 2.2 Example of attenuator-type device (energy extraction is achieved through the relative rotation of the two cylinders)**

Direct-driven conversion (Figure 2.3) has also been advocated for WABs, particularly heaving point-absorbers. It has the advantage of not requiring a mechanical interface and avoiding the losses occurring in the intermediate steps required by other systems. However, its development is still at a very early stage and its deployment is necessarily associated with the implementation of large-scale power electronics and storage facilities.
2.3.1.1 Linear PTO

A preliminary simple model, which permits also to check the quality of the time-domain results comparing them with the ones given by the frequency-domain, consists in assuming just a damper as Power Take-Off, capable of opposing a force proportional to the velocity of the buoy. Additionally, one can assume the presence of a spring term (i.e. proportional to the displacement), that is required to guarantee the stability of the system if no restoring force is present. In this case, since a floating body oscillating in heave experiences a restoring hydrostatic force, the inclusion of a spring term is not needed.

The PTO force is in this case described by the product of a constant coefficient $C_{PTO}$ (expressed in kg/s) and the velocity of the buoy (m/s):

$$F_{PTO}(t) = C_{PTO} \dot{x}(t)$$

(2.5)

The instantaneous power absorbed by the buoy, therefore, is given by
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\[ P(t) = C_{PTO} \dot{x}(t)^2 \]  

(2.6)

The assumption of linearity is not strictly applicable to realistic PTOs. However, as it will be shown, the application of a linear model can be a relatively good indicator to estimate forces and motions of the device.

### 2.3.1.2 Coulomb damping PTO

Most of the concepts for hydraulic energy transmission in wave energy are based on rectifying the flow through the use of check valves (see §2.3.1.3). This means that, if the compressibility of the cylinder chambers can be neglected, the piston (and therefore the floater) experiences a constant force that is opposite to its sense of translation and is proportional to the difference of pressure between high-pressure and low-pressure accumulators. The definition of a constant force which is only proportional to the opposite of the sign of the velocity can be resembled to the typical case of a Coulomb damping.

Thus, for the mathematical model of a point-absorber linked to a Coulomb-type PTO, we consider a PTO force described by the equation

\[ F_{PTO}(t) = C_{PTO} \text{sign}(\dot{x}(t)) \]  

(2.7)

One obvious consequence of this definition is that the PTO force is discontinuous at \( \dot{x} = 0 \) so that the numerical integration scheme might fail to converge due to infinite oscillations around the zero-velocity position. It makes sense to consider a modified version of the (2.1) that reads

\[
(M + A_{33}) \ddot{x}_3 = \begin{cases}
0 & \text{if } (\dot{x} = 0) \wedge \left( C_{PTO} \geq \left\{ F_r(t) - \int_{-\infty}^{t} K_{33}(t-\tau)\dot{x}(\tau)d\tau \ldots \right. \right.
\left. \ldots - \rho g S x_3(t) \right) \right) \\
F_r(t) - \int_{-\infty}^{t} K_{33}(t-\tau)\dot{x}(\tau)d\tau - \rho g S x(t) - C_{PTO} \text{sign}(\dot{x}) & \text{otherwise}
\end{cases}
\]  

(2.8)

This means that the acceleration of the buoy is maintained equal to zero whenever its velocity reaches zero until the sum of all the other forces acting on the buoy does not overcome the constant PTO.

In principle, the Coulomb damping is equivalent to a hydraulic PTO with a stationary pressure difference between the two accumulators and negligible compressibility and losses in the pipes. However, since in the model presented below, those effects are actually taken into account, some differences can be observed.

### 2.3.1.3 Hydraulic PTO

A preliminary scheme of a hydraulic PTO is shown in Figure 2.5. This system consists of a hydraulic circuit where the transmission fluid is considered to be oil. The motion of the buoy is transferred to a double-acting piston capable of sliding within a hydraulic cylinder. The cross-section area of the piston rod is neglected. The two chambers of this cylinder are connected to a high pressure gas accumulator (b) and a low pressure gas accumulator (c) through a couple of control valves whose opening will be dependent on the sign of the velocity of the buoy and on the pressure. The compressibility of the oil inside the chambers of the hydraulic cylinder is taken into account through the introduction of two additional gas reservoirs (d and e). The thermodynamic transformations involving the gas contained in the accumulators are assumed to be isentropic, i.e. no heat exchange takes place between the accumulators and the environment. This is a fairly realistic assumption since the time-scale of the temperature changes in the water and inside the device that could determine heat exchange is expected to be much larger than the one corresponding to the pressure changes associated to the
compression and expansion cycles. A fixed-displacement hydraulic motor with no oil leakage and unitary volumetric efficiency links the high and low-pressure reservoirs producing a useful torque dependent on the pressure difference. The motor torque is counteracted by a resistive torque imposed by an asynchronous electrical generator that is directly connected to the hydraulic motor and feeds power into the grid.

\[ F_c(t) - \rho g S x(t) - \int_{-\infty}^{t} K(t-\tau) \dot{x}(\tau) d\tau - A_p (p_d - p_e) \]

\[ \dot{x} = \frac{\dot{x}}{m + A_w} \]

\[ V_d = -A_p \dot{x} + C_d A \sqrt{\frac{2}{\rho}} \max((p_d - p_b),0) - C_d A \sqrt{\frac{2}{\rho}} \max((p_e - p_d),0) \]

\[ V_e = A_p \dot{x} + C_d A \sqrt{\frac{2}{\rho}} \max((p_e - p_b),0) - C_d A \sqrt{\frac{2}{\rho}} \max((p_e - p_e),0) \]

\[ V_b = -C_d A \sqrt{\frac{2}{\rho}} \max((p_d - p_b),0) - C_d A \sqrt{\frac{2}{\rho}} \max((p_e - p_b),0) + q_m \]
\[
\dot{V}_c = C_d A_c \sqrt{\frac{2}{\rho_o} \max((p_c - p_d), 0)} + C_d A_t \sqrt{\frac{2}{\rho_o} \max((p_c - p_c), 0)} - q_m
\]  

(2.13)

Assuming the gas compression/expansion process inside the accumulators to be isentropic, the pressure inside every accumulator is given by:

\[
p_j = p_0 \left( \frac{V_{0j}}{V_j} \right)^{1.4}
\]  

(2.14)

The modelling of the connection between the motor and the generator is done in the following assuming that they rotate at the same velocity although a gear change could be interposed in case it was necessary. If the losses in the gearbox were negligible, the results in terms of power output and its variability should not be different from the ones here presented.

The flow rate across the motor can then be expressed as:

\[
q_m = \frac{D_m \omega_m}{\eta_v}
\]  

(2.15)

\[
\dot{\omega}_m = \frac{\eta_m D_m (p_b - p_c) - T_g}{J}
\]  

(2.16)

The useful power \( P_u \) can be written as:

\[
P_u = T_g \omega_m
\]  

(2.17)

Application of a constant resistant torque (and a largely variable rotational velocity) by the generator might be difficult and not desirable (depending on the type of electrical machine). One could think of controlling the rotational velocity of the generator through a PI controller. To model a PI controller it suffices to add one additional state and define a reference value for the angular velocity of the motor-generator:

\[
T_g^* = k_p \frac{D_m (p_b - p_c) - T_g}{J} + k_i (\omega_m - \omega_{ref})
\]  

(2.18)

The two PI parameters \( k_i \) and \( k_p \) should be selected according to the desired response. It is clear that, setting these constants to higher values would allow a quicker response and smaller oscillations of the angular velocity of the motor. However, the variation of the torque would also be really large and might constitute a source of problem for the functioning of the hydraulic motor.
3 DEFINITION OF APPLICABLE ENVIRONMENTAL LOAD REGIMES FOR OCEAN ENERGY DEVICES

3.1 ENVIRONMENTAL CONDITIONS FOR OCEAN ENERGY TECHNOLOGIES

It is clear that the definition of the appropriate rated power for ocean energy technologies is largely dependent on the resource available at the deployment site. One could even argue that some devices are actually scalable depending on the energy available so that a set of different configurations might be defined for deployment at different locations. The structural design of marine energy devices and their components (e.g. moorings) is particularly sensitive to the extreme environmental conditions occurring at the site so that the fundamental assessment of the structural configuration is typically carried out on a reduced set of sea states representative of the most severe conditions that are likely to occur along the lifetime of the device.

On the other hand, the PTO design should be based, as a first step, on the most frequent sea states on which the device is expected to be operational. It is desirable to design the PTO in such a way that its efficiency is maximal in correspondence with the most frequent conditions and therefore the correct identification of representative sea states for PTO design and testing is particularly important.

Wave energy resource is commonly described by the spectrum computed in the frequency domain over intervals of 30 to 60 minutes which define a sea state. Thus, every sea state can be described by a spectrum and every spectrum can be characterised by a series of parameters, the most important of which are the significant wave height and the energy period. Details on their definition and meaning have been the subject of recent standardisation within the activities of the IEC TC114 Committee ([22]).

The occurrences of each pair of values of significant wave height and energy period can be represented in a table or in a contour plot like the one shown in Figure 3.1 to provide a direct visualisation of the most common sea states occurring at a site.

![Figure 3.1 Occurrences of sea state at the bimep](image)

Figure 3.1 shows in particular the occurrences at the bimep site ([23]) based on four years of sample data generated by a calibrated numerical model ([24]). One can see that the most frequent sea states are characterised by a significant wave height of around 1.5 m and an energy period of approximately 8.5 seconds. In theory, being the most frequent, the red area represents a range of conditions in which the device would be operating most of the time. However, this does not necessarily correspond to the range of conditions in which it extracts the most of the energy.

In fact, it is well known that the incident wave energy flux is directly proportional in deep water to the energy period and the square of the significant wave height. Thus, if we compute the annual energy flux per sea state and quantify the contribution to the total in each cell, we obtain a plot like the one in Figure 3.2.
It is clearly visible that most of the energy incident at the bimep is associated with larger significant wave heights and energy periods than the ones identified previously. This consideration might have to be checked also against the actual hydrodynamic efficiency of the device, which would probably be different depending on the energy period. When confronted with the challenge of designing a PTO system for a wave energy converter at a specific site, then, it is logical to think of a desirable operating area described by a relatively large portion of the coloured area in Figure 3.1. This operational range could be limited by a lower bound representing the minimal requirements in terms of resource to produce energy and an upper bound that correspond to the most severe conditions in which the system is still capable of producing energy.

Since the efficiency of the PTO system is often variable depending on the operating conditions, a possible objective of the design would be to obtain the best efficiency in correspondence with the most frequent resource. Thus, any fundamental testing schedule for conversion system should include a set of dynamic loads corresponding to typical sea states representing “best-working” points.

Finally, in some cases it might be necessary to provide some understanding of specific “survival” operation modes corresponding to extreme environmental conditions in which the PTO might be required to perform different functions other than producing electricity.

Thus, for the definition of typical dynamic load regimes for marine energy devices, we might define a set of environmental conditions corresponding to the following four fundamental load cases:

- Optimal operating conditions
- No-load conditions
- Extreme load conditions
- Extreme safety conditions

### 3.2 Optimal Operating Conditions

As explained earlier, under this situation the PTO should work with the best efficiency. We might define the optimal operating conditions to coincide with the most frequent wave conditions because it is likely that the PTO itself would be designed to operate at best in this situation. However, from the simple discussion outlined in the previous section, it is clear that the most occurring sea state does not necessarily coincide with the one where most of the energy is extracted.

Thus, for the case of the bimep envisaged in this document, two basic options arise for a definition of the best operating conditions:

- Most occurring sea state \( (H_s=1.5 \text{ m}, T_e=8.5 \text{ s}) \)
- Most energetic sea state \( (H_s=2.5 \text{ m}, T_e=11 \text{ s}) \)
What do we mean by optimal for real PTO conditions? In general we make reference to the maximum efficiency, or, in other words, to the conditions in which the energy losses occurring in the energy conversion are minimal with respect to the energy produced.

Consequently, a complete characterisation of the PTO system should be known since the beginning and the numerical models should include proper account of all the losses in its components. Alternatively, optimal conditions might be derived based on requirements on the reliability and durability of the elements of the PTO. To this aim, either fatigue analysis or data from previous experiences might be necessary.

For the systems presented in §2.3, the following arguments apply:

- Linear and Coulomb PTO: no major differences from other operational ranges unless an empirically-defined efficiency is taken into account
- Hydraulic PTO: Efficiency of the motor and circuit (valves and pipes) should be checked and the system should be designed so as to guarantee its best performance at these conditions

### 3.3 No-load conditions

This should represent a condition for minimum operation of the PTO. Basically, it is likely that for energy production, the conversion system has to run with a minimal load and/or velocities. It makes sense to test its behaviour in sea state conditions which are not very energetic but are nevertheless sufficient to guarantee energy production.

Considering the resources at the bimep, a possible lower bound for PTO operation might be based on a sea state described by

- $H_s < 1$ m, $T_e = 5-10$ s

For the generation of representative dynamic loads in §4, we consider $H_s = 0.5$ m and $T_e = 7$ s.

Is the PTO in this situation behaving differently from the optimal situation? Possibly the control system would assure separate operating conditions. For example, it is to be expected that the damping of the PTO in this case is consistently smaller than in other more energetic sea states. Thus, we might consider different variables that impose a smaller load on the wave energy converter.

- Linear and Coulomb: small values of the coefficients. Efficiency not included
- Hydraulic PTO: The resistive torque (or the reference rotational velocity) would possibly be small. The efficiency of the motor and circuit (valves and pipes) should be verified.

### 3.4 Extreme load conditions

Each PTO will have an operational range on which it can keep on generating power. How can this be determined? Most likely, these conditions will be different for types of conversion and devices and it is impossible to set a standard set of conditions which is valid for all the possible systems.

However, some assumptions can be made based on experience and the consideration of specific PTOs. As regarding the environmental conditions observed at the bimep, one could establish an operational range based on a short return period (i.e. three or four months). Alternatively, considering the current state of the technologies and their size, a maximum operational range of the order of $H_s = 7$ m might be taken as a reference sea state, though some care should be taken when assessing certain types of devices. In terms of cycling loads, shorter periods might be more constraining and the influence of wave grouping could be decisive for some cases. Perhaps a set of representative spectral shape could be defined in the beginning to be tested for PTO operation.

It is likely that the PTO in this situation will apply the largest resistive loads to the system. Therefore, the dissipation due to friction and other mechanism would be maximal and the efficiency is possibly worse than in other cases. The objective of the PTO control in this condition would be to maintain the operation and assure the safety of the equipment at the same time (for example in terms of pressure, loads and velocities).

- Linear and Coulomb: Largest values. Efficiency not included
- Hydraulic PTO: The resistive torque (or the reference rotational velocity) would be the largest value that can be provided. The efficiency of the motor and circuit (valves and pipes) should be verified.
3.5 Extreme safety conditions

Due to the large forces occurring at sea, it is likely that WECs will not operate continuously in severe conditions. Thus, PTO systems will be locked or allowed to be by-passed. The exact mechanics of this condition is certainly dependent on the device and on the type of PTO so that it is difficult to define a set of general recommendations and series of dynamic loads to be used for the testing and assessment of PTO systems.

In terms of environmental conditions, we might expect these extreme conditions to correspond to the worst dynamic conditions, i.e. a return period of the order of 50 years. For the bimep site, this would correspond to a sea state of approximately $H_s=13\,m$ and $T_e=16\,s$.

From the dynamic point of view, the obtainment of a safety condition might be achieved by several possible options:

- A very large damping associated to a dissipative mechanism (i.e. not related to the energy conversion chain)
- A very stiff connection
- Some type of braking mechanism

The actual modelling of this situation is complicated and very much dependent on the specific conversion mechanism considered. Interaction with suppliers and developers is probably necessary at this point to assure the correct representation of the dynamic behaviour of the system in these conditions.
4 GENERATION OF DYNAMIC LOAD TIME SERIES FOR SPECIFIC CASE STUDIES: WAB

We consider a point-absorber like the one described in the beginning of §2.3.1. The device is composed by a cylindrical floating buoy of radius and draught equal to 5 m and extracts energy through its heave motion with respect to the seabed. On the following, we will consider deep water conditions; though this assumption might not be completely realistic for a device of this kind (the large depth would probably make unfeasible the concept).

The hydrodynamics are described by frequency-dependent hydrodynamic coefficients computed by using a BEM code as mentioned in §2.1. These coefficients are then applied to derive the RIRF and the excitation force for the time-domain analysis.

To assess the relevance of separate assumptions and provide generality to the results, four separate PTO systems are considered for the representation of the point-absorber dynamics:

- Hydraulic PTO with constant motor displacement and constant resistive generator torque (in this case the rotational velocity of the motor is highly changing)
- Hydraulic PTO with constant motor displacement and PI control of the rotational velocity (the velocity follows a specified reference value and is approximately constant whereas the generator torque is variable)
- Linear PTO with constant damping coefficient
- Coulomb PTO with constant opposing force

We will assume constant values for several geometrical and dynamic parameters of the hydraulic PTO for the sake of convenience. These values are the result of a preliminary sizing based on practical considerations and realistic requirements. They were not, however, aimed at any specified configuration and assembly within a real converter and might therefore need some changes assuming that a more detailed design is required.

Table 4.1 presents the design data that were kept constant along all the case studies here considered.

The definition of the other parameters with respect to the different operational conditions was based on previous analyses carried out in [25] with the purpose of characterising the system and define optimal values for best performance. Thus, the generator torque or reference rotational velocity specified at the beginning of each case study was established based on a preliminary sensitivity analysis.

Further analyses might be carried out considering the application of different control strategies in the same loading conditions and its consequences on the actual design and testing of the PTO system but they are out of the scope of this deliverable and will not be pursued here.

Since the objective of these two simple formulations is the representation of an equivalent PTO with comparable characteristics to a real one, the definition of the damping coefficient of the linear PTO and of the resistive force of the Coulomb system was based on the established values of the control parameters of the hydraulic PTO.

For instance, if we assume known the generator torque applied to the motor, we know that, on the average, the difference of pressure between the High-pressure and Low-pressure accumulator is given by

\[ \Delta p \equiv \frac{T_g}{D_m} \]  

(4.1)

If the capacity of the accumulators was sufficiently large, their pressure difference would be practically constant along the time. Furthermore, if the compressibility of the oil in the chamber is negligible, then the difference of pressure that the piston experiences is equal to the one existing between the two accumulators. Thus the force acting on the piston is equal to the product between the pressure and the surface area of the piston.

Taking the root-mean-square value of the velocity, we can provide an expression for the equivalent linear PTO coefficient:

\[ C_{PTO} \equiv \frac{(P_{HP} - P_{LP})S_{pav}}{\dot{x}_{rms}} \]  

(4.2)
which becomes

\[ C_{PTO} \approx \frac{T_x S_{pist}}{D_m x_{rms}} \]  \hspace{1cm} (4.3)

Analogously, for the case of a Coulomb-type PTO, the expression of the equivalent force coefficient is

\[ C_{PTO} \approx \frac{T_x S_{pist}}{D_m} \]  \hspace{1cm} (4.4)

Thus, the two simplified PTO schemes can be used for comparison by applying the values defined by (4.3) and (4.4).

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>( V_{ob} ) (m(^3))</td>
<td>8</td>
</tr>
<tr>
<td>( V_{oe} ) (m(^3))</td>
<td>4</td>
</tr>
<tr>
<td>( V_{od}}=V_{oe} ) (m(^3))</td>
<td>0.05</td>
</tr>
<tr>
<td>( p_{od} ) (MPa)</td>
<td>10</td>
</tr>
<tr>
<td>( p_{oe} ) (MPa)</td>
<td>10</td>
</tr>
<tr>
<td>( \rho_c ) (kg/m(^3))</td>
<td>850</td>
</tr>
<tr>
<td>( A_p ) (m(^2))</td>
<td>0.2</td>
</tr>
<tr>
<td>( C_d )</td>
<td>0.99</td>
</tr>
<tr>
<td>( A_v ) (m(^2))</td>
<td>0.002</td>
</tr>
<tr>
<td>( D_m ) (m(^3)/rad)</td>
<td>1.2x10(^{-4})</td>
</tr>
<tr>
<td>( J ) (kgm(^2))</td>
<td>7.5</td>
</tr>
<tr>
<td>( \eta_m )</td>
<td>0.98</td>
</tr>
<tr>
<td>( k_p )</td>
<td>4</td>
</tr>
<tr>
<td>( k_i )</td>
<td>10</td>
</tr>
</tbody>
</table>

Table 4.1 Design values for the hydraulic PTO considered in the simulations

On the next subsections we will carry out different simulation runs for the operational conditions defined earlier. Due to the random nature of sea waves, even if a very long duration is considered, the resulting time series are different every time a set of different random phases for the wave components is considered. To characterise the statistics of this process, a large number of runs with an appropriately large duration would be required. From the point of view of the performance assessment, however, it has been shown (see [20]) that durations of 1800 s (half an hour) allow an average standard deviation of the estimated value smaller than the 10%. The chosen method within this document was to provide results corresponding to five runs of 1800 s duration each for each of the sea state considered. Additional safety factors might be included if these data were to be applied for design consideration.
4.1 Optimal Operating Conditions

As detailed in §3.2, there are two options for the definition of the sea state for assessing optimal operating conditions of the PTO, depending on whether we prefer the system to be more efficient on the most occurring or on the most energetic situations.

At the design stage, a proper analysis would require the run of several cases for all the relevant environmental conditions at a site, particularly if the objective is to characterise the behaviour of the PTO. Within this work, however, we limit the analysis to the two sea states defined in §3.2. We will call these two alternatives (a) and (b).

4.1.1 Sea state (a) ($H_s=1.5\ m$, $T_e=8.5\ s$)

As mentioned before, the control variables to be set for the hydraulic systems are specified in this case with the objective of maximising the average power absorption of the device. Taking as reference the results shown in [25], this corresponds to either a constant generator torque or reference rotational velocity on a specific sea state. It was shown that the optimal value for these variables is not dependent on the significant wave height but only on the energy period (at least for the model taken into consideration here).

Thus, for the present case, the values chosen were 67.5 Nm for the generator torque and 1200 rpm for the desired rotational velocity.

Figure 4.1 Sample of time series of the forces at the cylinder for different PTO in optimal load conditions (a)

Notice that this corresponds, following the procedure described previously, to a linear PTO damping coefficient of 1333 kNs/m and a constant Coulomb PTO force of 112.5 kN.

A sample of the time series of the forces acting on the cylinder is shown in Figure 4.6. The two control strategies applied to the hydraulic PTO produce very similar results and so does the linear PTO case. The Coulomb-type PTO, however, is associated with a constant force throughout the simulation as prescribed by (2.7). It should be expected, therefore, that this model is closer to the first two on averaged values rather than on time-changing variables.
These conclusions are also confirmed by Figure 4.2 where we can see again that the two control strategies produce very similar results if properly set for optimal power absorption. It is interesting to notice that a linear PTO tends to slightly underestimate the velocity whereas the Coulomb PTO determines much larger velocity peaks than the ones produced by the hydraulic system, a consequence of the application of a constant opposing force.

The corresponding power absorbed at the cylinder for the same sample can be seen in Figure 4.3. We should notice that, since this power is instantaneously absorbed by the system, its value can be occasionally negative for a very limited time (i.e. the pressure difference on the piston drives the buoy motion). This does not occur in case a linear or a Coulomb-type PTO is taken into account.
The profile of the power output is very similar between the different PTO types. However, the peaks resulting from the application of a Coulomb force are substantially smaller and sometimes even slightly displaced in time with respect to the other three.

For the case of the hydraulic PTO, the behaviour of the generator can also be analysed in such a way that the time series of motor torque and rotational velocity could be actually used as input for the validation and testing of electrical generators.

Figure 4.4 and Figure 4.5 show respectively the rotational velocity and the electrical power output for the two control strategies here applied.
As expected, the PI control is successful in maintaining the velocity very close to the reference chosen (1200 rpm) whereas the constant torque determines a very variable rotation which might be not acceptable for the operational efficiency of the hydraulic motor and of the generator.

The signal of the absorbed power for the two cases is however relatively similar, the main difference being sharper but smaller variations in the power given by the PI-controlled system. Also, the inertia of the system with constant torque is somewhat slower since it appears to be lagging by a few seconds on the output generated in the PI-controlled case.

The main statistical parameters referring to five simulated runs of the four types of PTO for the optimal conditions (a) are summarised in Table 4.2.

<table>
<thead>
<tr>
<th></th>
<th>Hydraulic PTO (constant torque)</th>
<th>Hydraulic PTO (PI controlled)</th>
<th>Linear PTO</th>
<th>Coulomb PTO</th>
</tr>
</thead>
<tbody>
<tr>
<td>Largest buoy motion/maximum positive stroke (m)</td>
<td>0.6160</td>
<td>0.6578</td>
<td>0.6241</td>
<td>1.6128</td>
</tr>
<tr>
<td>Smallest buoy motion/maximum negative stroke (m)</td>
<td>-0.6388</td>
<td>-0.6845</td>
<td>-0.6326</td>
<td>-1.4886</td>
</tr>
<tr>
<td>RMS buoy motion / RMS stroke (m)</td>
<td>0.1865</td>
<td>0.1737</td>
<td>0.1675</td>
<td>0.3451</td>
</tr>
<tr>
<td>Largest buoy velocity (m/s)</td>
<td>0.4416</td>
<td>0.4581</td>
<td>0.4635</td>
<td>1.2976</td>
</tr>
<tr>
<td>Smallest buoy velocity (m/s)</td>
<td>-0.4363</td>
<td>-0.45</td>
<td>-0.3771</td>
<td>-1.3246</td>
</tr>
<tr>
<td>RMS buoy velocity (m/s)</td>
<td>0.1267</td>
<td>0.1161</td>
<td>0.1109</td>
<td>0.2613</td>
</tr>
<tr>
<td>Maximum force at the cylinder (kN)</td>
<td>580.33</td>
<td>576.14</td>
<td>563.75</td>
<td>112.5</td>
</tr>
<tr>
<td>Minimum force at the cylinder (kN)</td>
<td>-594.66</td>
<td>-511.69</td>
<td>-501.61</td>
<td>-112.5</td>
</tr>
<tr>
<td>Maximum power absorbed by the buoy (kW)</td>
<td>254.80</td>
<td>262.22</td>
<td>261.30</td>
<td>149.01</td>
</tr>
<tr>
<td>Average power absorbed by the buoy (kW)</td>
<td>16.15</td>
<td>15.67</td>
<td>16.24</td>
<td>19.58</td>
</tr>
</tbody>
</table>

Table 4.2 Results of the dynamic analysis of a point-absorber in optimal load conditions (a)

These values might be taken as reference for the definition of appropriate dynamic load regimes. For instance, loading time series might be defined based on the average and the standard deviation of a particular variable. In this case, however, a distribution should be established. One might think of using a standard formula (for example a Gaussian) or determining a theoretical one based on the fitting of the values resulting from the simulation.

All the results from the simulations are nevertheless available and can be provided on request to the user. We should notice that, the application of a constant force in the Coulomb case, as seen also in the figures shown previously, produces larger buoy motions both on the average and on the extremes. This is compensated on the other hand by very smaller maximal forces since these are constant and equal to the established value in (4.4).

The extremes should be checked with respect to the maxima specified for the design parameters of the PTO. For example, a velocity larger than 1 m/s for the piston of a hydraulic ram would be probably not acceptable so that some braking mechanism should be provided to avoid the occurrence of such situation.

Other constraints might be imposed on the stroke, depending on the cylinder and on the mechanism used. The forces at the cylinder, on the other hand, are likely to be a decisive input for the design and assessment of the rod and of the chamber.
We should notice that the power absorbed by the system at the cylinder can be very changing and can be, at times, very large even though its actual average is less than the 10% of the maximum. This phenomenon is not necessarily related to such large variations in the actual electrical power output since some storage effect is provided by the accumulators. This reasoning might not be valid if a linear generator is taken into account.

4.1.2 Sea state (b) (\(H_s=2.5\) m, \(T_e=11\) s)

We define again control variables for the hydraulic PTO to maximise the average absorbed power. For the present case, the values chosen were 150 Nm for the generator torque and 2250 rpm for the desired rotational velocity, which corresponds to a linear PTO damping coefficient of 2197 kNs/m and a constant Coulomb PTO force of 250 kN.

Taking a sample of one of the runs carried out for this sea state, we can see that the forces and the velocities of the piston plotted respectively in Figure 4.6 and Figure 4.7 confirm the conclusions presented before for the most occurring sea state.

Notice that the velocities associated with a Coulomb PTO are very large and possibly already out of the bounds specified for this kind of systems. This result points to the possible adoption either of a larger force or an additional mechanism to reduce the peaks. On the other hand, velocities for the linear PTO are often slightly smaller than the ones corresponding to the hydraulic PTO.

The hydraulic power time series is very similar in all the cases with the individual peaks occurring mostly at the same instants. The magnitude of those peaks, though, is distinctly different with the Coulomb PTO producing larger power at most of the peaks with the exception of the ones associated with the largest forces (Figure 4.8).

Efficiency of the hydraulics should be checked in this range since the hydraulic power to the cylinder might be occasionally very large. The current model accounts for losses in the valves and therefore the calculation of the efficiency is very sensitive to the volume flow which is higher for higher velocities (as it happens for highly energetic waves as in this case). For instance, the efficiency recorded for the previous case was of the order of the 65% and it is around the 60% in this case. More accurate estimations should be carried out once the single components of the PTO are defined.
Figure 4.7 Sample of time series of the buoy velocities for different PTO in optimal load conditions (b)

The rotational velocity of the hydraulic motor for this case can be seen in Figure 4.9 for the two control strategies. As seen before, the electrical power output is obviously much smoother than the hydraulic one and the application of different control strategies determines different time scales for the power variation (Figure 4.10).

A summary of the results for the different PTOs for the most energetic sea state is given in Table 4.3. The motions of the buoy are to be noticed, being in the case of the Coulomb system possibly too large to allow the applications of common hydraulic cylinders.

One should pay attention to the very large forces acting on the cylinder which might determine fatigue issues, particularly if considering that these conditions are the ones where the device is expected to produce most energy.
D2.19 Generation of a set of typical dynamic load regimes for common conversion devices

Figure 4.9 Sample of time series of the rotational speed of the motor for the two hydraulic PTOs in optimal load conditions (b)

Figure 4.10 Sample of time series of the electrical power for the two hydraulic PTOs in optimal load conditions (b)

<table>
<thead>
<tr>
<th></th>
<th>Hydraulic PTO (constant torque)</th>
<th>Hydraulic PTO (PI controlled)</th>
<th>Linear PTO</th>
<th>Coulomb PTO</th>
</tr>
</thead>
<tbody>
<tr>
<td>Largest buoy</td>
<td>1.188</td>
<td>1.3189</td>
<td>1.0345</td>
<td>2.9805</td>
</tr>
<tr>
<td>motion/maximum positive stroke (m)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>-----------------------------------</td>
<td>---</td>
<td>---</td>
<td>---</td>
<td></td>
</tr>
<tr>
<td>Smallest buoy motion/maximum negative stroke (m)</td>
<td>-1.2861</td>
<td>-1.2531</td>
<td>-0.9661</td>
<td>-2.4760</td>
</tr>
<tr>
<td>RMS buoy motion / RMS stroke (m)</td>
<td>0.3222</td>
<td>0.3721</td>
<td>0.2749</td>
<td>0.6048</td>
</tr>
<tr>
<td>Largest buoy velocity (m/s)</td>
<td>0.6700</td>
<td>0.7144</td>
<td>0.5748</td>
<td>1.8236</td>
</tr>
<tr>
<td>Smallest buoy velocity (m/s)</td>
<td>-0.6577</td>
<td>-0.6835</td>
<td>-0.5527</td>
<td>-1.9372</td>
</tr>
<tr>
<td>RMS buoy velocity (m/s)</td>
<td>0.1761</td>
<td>0.2013</td>
<td>0.1427</td>
<td>0.3653</td>
</tr>
<tr>
<td>Maximum force at the cylinder (kN)</td>
<td>1226.8</td>
<td>1137.9</td>
<td>1261.7</td>
<td>250</td>
</tr>
<tr>
<td>Minimum force at the cylinder (kN)</td>
<td>-1248.7</td>
<td>-1141.7</td>
<td>-1214.4</td>
<td>-250</td>
</tr>
<tr>
<td>Maximum power absorbed by the buoy (kW)</td>
<td>813.61</td>
<td>808.59</td>
<td>725.21</td>
<td>484.3</td>
</tr>
<tr>
<td>Average power absorbed by the buoy (kW)</td>
<td>45.90</td>
<td>52.46</td>
<td>44.74</td>
<td>56.97</td>
</tr>
</tbody>
</table>

Table 4.3 Results of the dynamic analysis of a point-absorber in optimal load conditions (b)

4.2 MINIMAL LOAD CONDITIONS \((H_S=0.5 \text{ m}, T_E=7 \text{ s})\)

The situation corresponding to no load conditions might be defined differently depending on the type of PTO and of the requirements on its operational range.

In some cases, the system operation in power production mode might not be possible, especially if some of the components require a minimum load or velocity (for example, the hydraulic motor is likely to require a minimum rotational velocity to work properly). In those situations one might consider the use of a by-pass circuit in hydraulic systems to declutch the lines connecting to the accumulators. Alternatively other declutching mechanisms might be envisaged to avoid the operation of PTO components outside the design conditions. Simulations and/or testing in these conditions might be interesting to check the friction and leakage of the mechanical elements responsible for the motion transmission of the device (e.g. seals, cylinders, bearings etc.).

However, for the purpose of defining general dynamic cycles of operations, it makes sense to verify the power production in this situation, particularly if it is associated with challenging requirements on the generator.

The best control variables for power absorption were found to be either a constant generator torque of 20 Nm or a reference rotational velocity of 500 rpm. Thus, the applied equivalent linear damping coefficient is 813.5 kNs/m while the applied Coulomb force is 33.33 kN.

Samples for forces, velocity and hydraulic power are shown in Figure 4.11, Figure 4.12 and Figure 4.13 respectively. The same conclusions found previously for the applicability of the Coulomb PTO are valid in this case as well.

The motions for this case can be significant because the damping applied has been left relatively small to allow large power absorption, as seen by the plot of the forces.
Figure 4.11 Sample of time series of the forces at the cylinder for different PTO in no-load conditions

Figure 4.12 Sample of time series of the buoy velocities for different PTO in no-load conditions
The generated power is obviously very small, possibly not sufficient to run the generator. The variations in the output are relatively similar for the two control strategies applied. Even though the rotational velocity can be maintained at a high rate, the resistive torque might be too small to assure a proper working regime (friction on the bearings and other elements are not taken into account within this models).

Figure 4.13 Sample of time series of the power at the cylinder for different PTO in no load conditions

Figure 4.14 Sample of time series of the rotational speed of the motor for the two hydraulic PTOs in no-load conditions
D2.19 Generation of a set of typical dynamic load regimes for common conversion devices

Figure 4.15 Sample of time series of the electrical power for the two hydraulic PTOs in no-load conditions

Table 4.4 Results of the dynamic analysis of a point-absorber in no-load conditions
A summary of the results obtained for the minimal load conditions is given in Table 4.4. As said previously, these values might provide a good reference for testing of specific components (i.e. the efficiency of the generator) or the validation of the capability of power delivery even in low waves.

4.3 EXTREME LOAD CONDITIONS ($H_s=7$ m, $T_E=13$ s)

The definition of the working range adequate to extreme load conditions is very difficult and possibly associated with the introduction of additional control strategies apart from the ones here analysed. This is particularly true if some of the dynamic variables actually exceed any specified limit prescribed by the PTO designer.

A detailed design of a PTO system, together with the equipment and control systems required to guarantee its proper operation, is out of the scope of this document and cannot be carried out without a refined specification of the wave energy technology and its dynamics.

It is however interesting to check the power output obtainable in this case with the same design variables defined previously to obtain an understanding of the loads and motions involved in the operation of a realistic PTO in such an energetic sea state.

The values of the control variables in this case are 490 Nm for the constant generator torque and 4900 rpm for the reference rotational velocity. The linear PTO damping coefficient is set equal to 4063 kNs/m whereas the Coulomb force is 817 kN.

![Figure 4.16 Sample of time series of the forces at the cylinder for different PTO in extreme load conditions](image)

As done previously, we show in Figure 4.16, Figure 4.17 and Figure 4.18 samples representing the time series obtained for the forces, velocities and power absorbed at the piston.

The design of some of the components of the hydraulic system might be based on these results. For instance, the maxima of the force on the piston could be taken as an input for the structural design of the rod (which has to withstand buckling) whereas the velocities and motions should be limited depending on the design requirements determined by the choice of the cylinder.

Notice that, in this case, the piston velocities are very large and would likely determine large leakages so that an over-damped solution (i.e. a larger generator torque or a smaller rotational velocity) might be preferable. An alternative option is the introduction of hydraulic brakes at the end of the chambers so as to limit the stroke of the piston as well.
The hydraulic power can be huge (up to 4 MW in this case) so that the efficiency of the system is necessarily reduced (in this model it goes down to 55-60%). Rotational velocity and electrical power output of the generator are plotted for the same time history in Figure 4.19 and Figure 4.20. The rotational velocity should be checked with respect to the operational requirements of the hydraulic motor and the generator. It should be noticed that gearboxes or other mechanisms might be introduced to allow decoupling and different transmission velocities. Also, the current model with constant resistive torque does not impose any minimal rotational velocity so that the hydraulic motor might occasionally reverse its flow in low-energetic periods in consequence of the high generator torque. Thus, a PI control of the rotational velocity seems necessary at this point, particularly if we considered the option of over-damping the motion which would require a constant larger torque.
(with more risk of flow-reversing). The over-damped solution could be achieved by reducing the reference rotational velocity (with the same motor displacement) so as to guarantee smaller motions and forces. This would clearly reduce the power output and possibly worsen the efficiency of the system but might be necessary (for example the value of 5000 rpm for the rotational velocity could be too large for most types of hydraulic motors available on the market).

![Graph 1: Sample of time series of the rotational speed of the motor for the two hydraulic PTOs in extreme load conditions](image1)

Figure 4.19 Sample of time series of the rotational speed of the motor for the two hydraulic PTOs in extreme load conditions

![Graph 2: Sample of time series of the electrical power for the two hydraulic PTOs in extreme load conditions](image2)

Figure 4.20 Sample of time series of the electrical power for the two hydraulic PTOs in extreme load conditions
<table>
<thead>
<tr>
<th></th>
<th>Hydraulic PTO (constant torque)</th>
<th>Hydraulic PTO (PI controlled)</th>
<th>Linear PTO</th>
<th>Coulomb PTO</th>
</tr>
</thead>
<tbody>
<tr>
<td>Largest buoy motion/maximum positive stroke (m)</td>
<td>2.5849</td>
<td>3.0227</td>
<td>1.987</td>
<td>6.4363</td>
</tr>
<tr>
<td>Smallest buoy motion/maximum negative stroke (m)</td>
<td>-2.8887</td>
<td>-3.0217</td>
<td>-2.1298</td>
<td>-6.4240</td>
</tr>
<tr>
<td>RMS buoy motion / RMS stroke (m)</td>
<td>0.6978</td>
<td>0.9120</td>
<td>0.5551</td>
<td>1.6560</td>
</tr>
<tr>
<td>Largest buoy velocity (m/s)</td>
<td>1.2405</td>
<td>1.2892</td>
<td>0.8743</td>
<td>4.2079</td>
</tr>
<tr>
<td>Smallest buoy velocity (m)</td>
<td>-1.1806</td>
<td>-1.3036</td>
<td>-0.9256</td>
<td>-4.1374</td>
</tr>
<tr>
<td>RMS buoy velocity (m/s)</td>
<td>0.3167</td>
<td>0.4169</td>
<td>0.2429</td>
<td>0.8756</td>
</tr>
<tr>
<td>Maximum force at the cylinder (kN)</td>
<td>3300.2</td>
<td>3194.6</td>
<td>3552.7</td>
<td>816.67</td>
</tr>
<tr>
<td>Minimum force at the cylinder (kN)</td>
<td>-3472.7</td>
<td>-3362.6</td>
<td>-3794.9</td>
<td>-816.67</td>
</tr>
<tr>
<td>Maximum power absorbed by the buoy (kW)</td>
<td>4070.1</td>
<td>4363.4</td>
<td>3512.7</td>
<td>3436.5</td>
</tr>
<tr>
<td>Average power absorbed by the buoy (kW)</td>
<td>259.29</td>
<td>341.57</td>
<td>242.71</td>
<td>413.16</td>
</tr>
</tbody>
</table>

Table 4.5 Results of the dynamic analysis of a point-absorber in extreme load conditions

The statistical characteristics of the most important parameters found for this case are presented in Table 4.5. The extreme values of forces, velocities and motions might be taken as a reference for design checking and assessment. The time series generated for this case might be defined as the most demanding schedules for PTO testing.
5 CONCLUSIONS: RECOMMENDATIONS FOR TESTING SCHEDULE OF PTO SYSTEMS

The definition of exhaustive recommendations for the testing of PTO systems would require a detailed design and analysis and, as such, is out of the scope of this deliverable. Also, it should be underlined that it could be carried out only after a detailed engineering assessment and a complete specification is available.

Within this document, we presented a preliminary methodology for the definition of the most relevant dynamic loads on the PTOs and an example of its application with reference to a heaving point-absorber oscillating with respect to the seabed and connected to a hydraulic PTO.

It is recommended that the definition of the appropriate loading conditions should start from the consideration of the applicable environmental conditions. Thus, the selection of an appropriate deployment site should be prior to any dynamic analysis.

If the weather characteristics of the deployment site are known, it is possible to carry out a preliminary selection of the most interesting sea states based on the definition of the likely operational range of the ocean energy technology. At least four different situations should be envisaged:

- Optimal loading conditions (most occurring sea state or the one associated with the largest energy contribution)
- Minimal loading conditions (low-energy sea state where the system is still expected to operate)
- Extreme loading conditions (high-energy sea state where the system will still be producing electricity but possibly with large dissipation and/or braking mechanisms)
- Extreme safety conditions (severe sea state corresponding to a survival conditions where the most important requirement is the preservation of the PTO elements)

Simulations were run on these different cases for several PTO types. It was found that a linear PTO, if correctly specified, can occasionally provide a relatively realistic output if compared to a hydraulic PTO. Forces and motions were defined for the four different cases together with the consideration of the required checks and control strategies for their operation. It was seen how a PI control strategy might be preferable to stabilise the operation of the generator.

Time series for the cases presented are available to the interested users. Alternatively, time series can be randomly generated by assuming random distribution formulae based on the mean and the standard deviation.

It should be noticed that no consideration on the scaling of such parameters has been given. This is a very important step that should be definitely analysed within the Marinet project, possibly with reference to real test rigs and configurations. The definition of the most adequate scaling factor is particularly important when considering friction and other dissipative effects and should definitely be addressed anytime the efficiency of a specific PTO configuration has to be assessed.
6 REFERENCES


