Numerical Analysis of Two Different Hydraulic Power Take-Off Configurations for Renewable Energy Applications

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Abstract — **Renewable sources of interest for the development of Hydraulic Converters are wave and wind energy. In both areas of application, the size of the installed machine is growing. This means handling more torque at the crankshaft, approaching the limit for electromechanical components available in series production. The hydraulic PTO stands out as an answer to the problem, being able to convert high torques at low speeds into low torques at high speeds, through robust and well-known systems and components.**

Alongside a review of the state of the art of hydraulic PTOs and the main control strategies applicable to wave energy extraction systems, the present work applies a hydraulic PTO to the ISWEC, an Inertial Sea Wave Energy Converter designed for the Adriatic Sea. When designing such hydraulic PTOs, it is necessary to consider components and control methods allowing control of the force - torque applied by the PTO to the prime mover and guaranteeing a constant generator speed. Sub-optimal configuration can result in very inefficient energy conversion, so understanding the design trade-offs is key to the success of the technology.

Numerical analyses of a wave to wire time domain model are performed to compare the performances of two different hydraulic circuit configurations, both aiming to optimize the power production of the wave energy converter (WEC) considered.

Keywords — **Renewable energy, Wave Energy Converter (WEC), Hydraulic transmission, Hydraulic Power Take-off, Latching-Declutching Control.**

I. INTRODUCTION

As an answer to the worldwide energy request, over the last decades renewable sources have been widely investigated. Some of these sources exploit the energy available in order to produce mechanical motion that needs to be further transformed into electrical energy. The system that is involved in the process of energy conversion from mechanical into electrical is the power take-off.

Among the different renewable sources, ocean waves convey power through motions that feature high torques and low speeds: as the size of the conversion system increases, the torques become too high to be handled by electro-mechanical power take-offs, up to a level that commercial units are no longer available. The power absorbed from ocean waves can also be very irregular, with peaks 10 times as big as the mean values [1], and the offshore application requires systems with low maintainability needs. Therefore, wave energy converters find themselves in need of a robust power conversion system capable of handling highly variable powers with high torques and forces at low speeds, and with acceptable efficiencies.

These operating limits are included in the scope of hydraulic actuation, which is able to handle high forces and torques through robust and well-known systems and components. Moreover, the hydraulic power take-off features the possibility of smoothing out the strongly fluctuating absorbed power providing a more stable output by decoupling the input power source from the output shaft through means of accumulators. Furthermore, the hydraulic PTO allows rectifying the motion through the adoption of simple valves.

Hydraulic PTOs have thus become a common choice among developers, especially for big size conversion systems. While hydraulic drive systems have been known a long time, their application to such irregular sources like ocean and sea waves calls for the development of adequate control strategies, necessary for the system to adapt to different sea states and conditions.

To date, a reasonable amount of prototypes of WECs have been developed: examples of such prototypes, like the SEAREV [2], the Wavestar [1] and the Pelamis [3], all show the opportunity of obtaining promising results with hydraulic transmissions, incentivising further studies in the field.

Nowadays, the focus of research resides in the control strategies to be applied to the conversion system.

The first two chapters of this paper provide a review of the state of the art of hydraulic power take-offs, focusing both on the conversion unit itself and its most common configurations and both on the main control strategies developed over the years, with a particular interest on their possible applications to hydraulic units. In the second part of the paper, a numerical analysis is carried out applying a hydraulic power take-off to the ISWEC. Two passive control laws are applied to the PTO and compared to each other, the former being a simple control on the generator torque and the latter a combination of the generator control with a sub-optimal declutching logic that clutches the pump when its speed reaches a defined threshold and declutches it when it inverts its spinning direction. The simulations have been run on a set of waves numerically generated based on the JONSWAP (Joint North Sea Wave Project) spectra and annual occurrences of the Adriatic offshore seaside. No losses in the conversion system are considered to this point.

II. The Hydraulic Power Take-Off

The PTO includes all those components between the prime mover and the generator that contribute to the energy conversion. In a hydraulic PTO, the prime mover is coupled to a hydraulic actuator, like a double-acting cylinder or a pump, that pressurises a fluid inside a circuit. At the other end of the system, the pressurised fluid is used to move a rotary generator, by means of a hydraulic motor or a high-head water turbine. Between the input and the output sources, depending on the configuration chosen, there may be accumulators to smooth out the fluctuations and directional valves to rectify the incoming alternate flow. A hydraulic transmission of this type works as a gearbox, reducing the output torque, and at the same time allows rectifying the input motion and attenuating its fluctuations.

To date, there is no industrial standard of a hydraulic power take-off unit applicable to WECs, but most of the investigations are carried out with very similar configurations.

A. Constant Pressure Configuration

The constant pressure configuration is based on the idea of keeping constant the pressure drop across the motor. It is the simplest and by far the most common system investigated in literature; examples can be found in the works of Falcao [5] and in the Pelamis [3], SEAREV [6] and Wavestar [1].

A simplified scheme of the constant pressure configuration is shown in Fig. 1. The WEC body acts on the cylinder/pump, pressurising the fluid inside the circuit: since the motion is alternate, the high pressure and low-pressure rams continuously switch along with the actuator motion. A 2-position control valve is used as a Gratez bridge in order to rectify the flow, so that the motor is not required four-quadrant mode. Downline the rectifier bridge, two hydraulic accumulators keep the high and low pressures as constant as possible, so that the generator might work in steady-state condition. The generator can be fixed or variable speed, and the motor can be fixed or variable displacement. A pressure relief valve is employed for safety reasons.

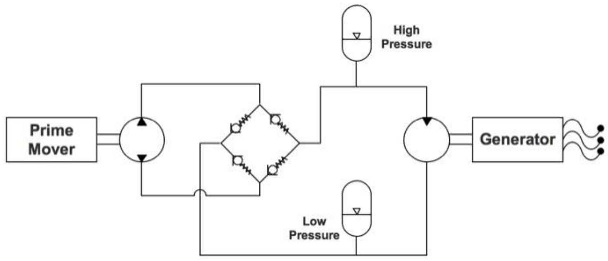

Fig. 1 Constant Pressure Hydraulic Power Take-Off

Since the pressures are kept as constant as possible, the force exerted by the PTO to the absorber is a Coulomb-like force, i.e. proportional to the pressure drop. This prevents the implementation of control strategies based on linear damping and spring forces like reactive phase control, theoretically diminishing the amount of power absorbable. On the other hand, force control of such PTOs can be achieved through valve transitions, instantaneously varying the pressure difference between a discrete set of values: possible control strategies are declutching control or discrete displacement control. Of course, in a real system, the accumulators have a finite volume, so the pressures are not perfectly constant and a certain degree of PTO force variation can be achieved even without valve transitions.

Constant pressure hydraulic PTOs prevent high levels of control with regard to power absorption, but show optimal results in rectifying and attenuating the fluctuations of the input motion, allowing the generator to work in more steady conditions.

B. Variable Pressure Configuration

The variable pressure configuration avoids the implementation of the accumulators: the hydraulic actuator directly feeds the motor, so that the pressure drop across it is directly related to the motor response to the incoming flow. The generator can be fixed or variable speed, and the motor can be fixed or variable displacement. A rectifier bridge can be implemented if the motor cannot reverse its flow by means of pilot valves. A pressure relief valve is employed for safety reasons. Examples of a variable pressure PTO can be found in the works of Costello and Ringwood [7]. A simplified scheme of the variable pressure configuration is shown in Fig. 2, where the rectification block is not implemented.

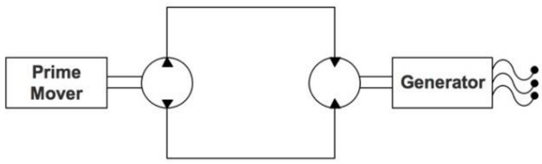

Fig. 2 Variable Pressure Hydraulic Power Take-Off

Force control can be achieved through pressure modulation, directly monitoring the difference of pressure between the two rams and therefore acting on the system parameters, like the generator breaking torque or the motor displacement. The PTO force can be continuously varied between its maximum and minimum values, allowing better control.

A variable pressure system is able to achieve better controllability than a constant pressure system, but the lack of accumulators can account for strong fluctuations in the motor-generator subsystem. A study conducted in ([1] and [7]) compares these two configurations: as expected, the variable pressure system shows better results in terms of controllability, but overall the constant pressure configuration outperforms the variable one, reaching higher power generation and considerable stronger efficiencies.

C. Variable-Constant Pressure Configuration

The variable-constant pressure configuration tries to include the advantages from both the variable and the constant pressure configurations, at the cost of more complicated systems with more components and thus lower conversion efficiencies [8]. The main idea of such a configuration is to exploit both the larger controllability of variable PTOs and the better working conditions of a generator of constant pressure ones. Such a PTO is made up of two parts: the variable pressure part of the system is coupled to the absorber, while the constant pressure part is coupled to the generator. An in-between shaft is employed to link the two parts. A simplified sketch of the configuration described is shown in Fig. 3. Variable-constant hydraulic PTOs are among the less studied in the field; to date, it is not easy to find detailed information on their performances.

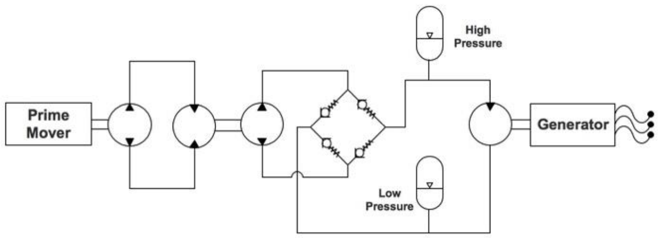

Fig. 3 Variable-Constant Pressure Hydraulic Power Take-Off

III. Control Strategies Applied to Hydraulic PTOs

The application of energy conversion systems to such irregular sources like ocean waves opens up to the problem of control. The incoming waves vary a lot over time, even on a daily and hourly basis [9]. Moreover, the differences between one sea state and another relative to a certain location are likely to be significant. As one can understand, the design of a PTO with no controllable parameters can lead to very poor efficiencies.

In order to understand the principles at the basis of the PTO control, it is important to understand how the conversion system affects the power production. The power take-off takes as input the absorbed power from the WEC body and releases the converted power to the electrical generator. On the other hand, the PTO responds to the WEC body with a reaction force that affects the absorbed power itself. Consequently, the power take-off has the main scope of both optimising the power absorbed and both guaranteeing high efficiency of the WEC with the implementation of an adequate force control.

In many cases, the control strategies developed are passive strategies, which means that the PTO parameters are optimised for the mean frequency of the sea site they are intended for and the process of optimisation is carried out offline, assuring that the natural frequency of the WEC is close to the frequency range of most of the sea states. On the other hand, active strategies work online in order to tune the PTO to the incoming wave; in this case, a wave prediction algorithm is usually needed. Examples of such strategies may be found in the works of Cargo [11].

Here a general outline of the most used and cited control strategies applicable to a hydraulic PTO units is given, with a particular focus on the two considered in this work: proportional control and declutching control.

A. Proportional Control

Proportional control is a control strategy applicable to constant pressure hydraulic PTOs that aims to keep constant the pressure drop across the motor and to tune it to its optimal value. It can be achieved by keeping the motor flow (or speed) proportional to the pressure drop across it or, analogously, by regulating the motor-generator torque proportionally to its angular speed. The parameter to be tuned is the constant of proportionality.

Examples of proportional control are to be found in the works of Falcao ([5] and [14]) and Ricci [15]. In [5] the author finds a relationship, for a heaving buoy, between the power absorbed and the PTO force, this latter supposed perfectly constant. An optimal value of the force for each given sea state is computed and it is shown that a proportional relationship between the pressure drop across the motor and the motor flow allows to tune the force to its optimal value. Numerical results of [5] show that a hydraulic unit with this type of control performs very similarly to a perfectly optimised linear resistive control. In [14] proportional control is coupled with phase control by latching, obtaining substantially stronger results, and in [15] it is coupled with phase control by means of extra-accumulators, again with remarkable improvements.

B. Declutching Control

Declutching control is an anti-casual sub-optimal strategy initially introduce in 2000 by Salter as unlatching control [12], and further investigated and optimised in 2009 by Babarit, Guglielmi and Clément [19]. It is proposed as a solution to the unwanted halting of the hydraulic actuator observed by Falcao in [5], which occurs when the PTO force (roughly constant) is too large with respect to the other forces acting on the absorber. Declutching control solves the problem with a bypass valve that decouples the actuator from the rest of the system for some parts of the cycle. Results of [19] show that, if the valve is operated in an optimal way, it allows achieving even better results than a configuration with multiple cylinders or multiple accumulators with different levels of pressure, at least on a theoretical level. In other terms, it is shown that with a constant pressure hydraulic unit, the PTO force has to be either null or maximum in order to maximise power absorption, and intermediate values are not to be taken on.

C. Others Control Strategies

Other control strategies as widespread as the previous ones are listed below. These strategies have not been taken into account in the system studied in this paper.

1) *Linear Resistive Control:* Linear resistive control, also known as linear damping control, is one of the simplest and most investigated control strategies in literature. As mentioned in [12], [13] and [17], the PTO action is modelled as proportional to the absorber velocity, with the constant of proportionality being negative, which accounts for resistive power only. This control strategy reduces thus to a simple amplitude control, and the control method cannot be optimal unless the resonant frequency is matched by other means. The kind of control cannot be applied in the constant pressure configuration due to the Coulomb-like damping force of the hydraulic unit.

2) *Reactive Control:* Reactive control is an optimal control strategy achieved by means of reactive power that is used to adjust the power take-off stiffness to match the undamped frequency of the absorber to the incoming wave frequency [16]. It is usually coupled with an amplitude control that tunes the PTO damping value to achieve the overall optimisation of both phase and amplitude. Plummer and Schlotter in [8] investigate an example of a reactive control with a variable-constant pressure hydraulic unit.

3) *Phase control by latching:* First introduced by Budal and Falnes in 1980 [10], phase control by latching is a sub-optimal control strategy that can be applied when the resonant frequency of the absorber body is higher than the incoming wave frequency. Latching control acts locking the body for a certain period when its velocity goes to zero to shift its resonance period to higher values. Optimal phase control by latching has been investigated by Babarit and Clément in 2006 [18], while a sub-optimal strategy has been proposed by Falcao in [14] (coupling it to a constant pressure hydraulic PTO), resulting in a higher absorption power both in regular and irregular waves than a simple proportional control.

4) Phase control with Extra Accumulators: First introduced by Eidsmoen in 1996 [20], phase control by means of extra accumulators is a control strategy specifically developed for hydraulic units that tries to achieve phase control by using extra accumulators that are controlled via two-position hydraulic valves. The concept at the basis of this strategy is to open the valve of the extra accumulator at a certain time to match the phase of the excitation force. The performances in regular waves of a constant pressure hydraulic power take-off endowed with extra accumulators have been investigated in [15]. Results show that a remarkable improvement in power production can be achieved with respect to proportional control.

5) *Discrete Displacement Control (DDC):* it is one of the few strategies especially developed for hydraulic units. DDC requires the PTO to employ discrete displacement actuators, like multi-chamber cylinders. The concept at the basis of DDC is to switch between chambers in order to discretely vary the PTO force and match it to a reference force previously established. This is done by means of two-position hydraulic valves. Since the pressure drop across the motor is kept constant by means of accumulators, the PTO force depends only on the cylinder area, and therefore on the chambers selected. The wave power extraction algorithm of a DDC consists of two major steps: the reference force generation and the force-shifting algorithm. An accurate description is reported in [1].

IV. Hydraulic Transmission Numerical Model

As introduced in the first chapter, a numerical study has been conducted on the ISWEC system, coupling the gyroscope to a constant-pressure hydraulic power take-off system. The numerical model considered is a wave to wire model that simulates the behavior of the WEC, from the wave-hull hydrodynamic interaction to the power generation including the hydraulic PTO. For an accurate description of the working principle, internal mechanics and numerical model of the ISWEC please refer to [4]-[21]-[22]-[23]-[24]-[25]-[26]-[27]-[28]-[29].

A. *Hydraulic PTO Mathematical Model*

The hydraulic power take-off is a constant-pressure hydraulic unit with a rectification block, a 2-position control valve for declutching purposes and a pressure relief valve that links the high-pressure ram to the low-pressure ram on the motor side, as shown in Fig. 4.

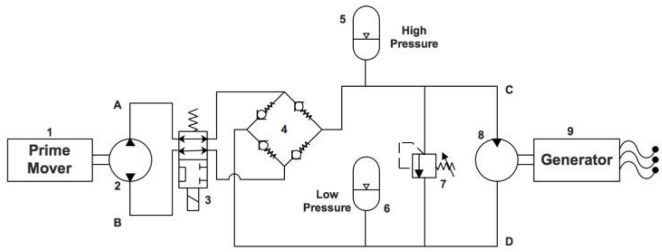

Fig. 4 Hydraulic Power Take-Off applied to the ISWEC

All components are considered ideal; hence, in normal operating conditions, the power absorbed on the pump side is equal to the power produced on the generator side. On the other hand, when the pressures overcome the high-pressure accumulator limit the flow is directly delivered to the low-pressure accumulator bypassing the hydraulic motor.

The power is conveyed from the hull to the PTO exploiting the gyroscopic effect obtained by the combination of the hull pitching oscillation with the spinning motion of a flywheel. The gyroscopic effect results in an oscillation of a gyroscopic system (prime mover -1-, Fig. 4). The gyroscope is coupled to the pump -2- along an axis ε perpendicular to the flywheel spinning axis ϕ. The governing equation of the gyroscope-pump subsystem is:

$$T_{gyro} = D_p(p_A - p_B) + (I_{gyro} + I_{pump})\ddot{\varepsilon} \tag{1}$$

Where:
- T_{gyro} is the driving torque on the ε axis due to the gyroscopic effect;
- I_{gyro} and I_{pump} are the gyroscope and pump moments of inertia respectively;
- $\ddot{\varepsilon}$ is the pump angular acceleration;
- Dp is the pump displacement;
- p_A and p_B are the pressures of the fluid at the pump ends.

Resolving the equation (1) the angular speed of the pump can be obtained; therefore, the pump flow rate Q_p is easily computed as:

$$Q_p = D_p \dot{\varepsilon} \tag{2}$$

The power absorption of the pump can be defined as:

$$P_{abs} = T_\varepsilon \dot{\varepsilon} = D_p \dot{\varepsilon}(p_A - p_B) \tag{3}$$

Where T_ε is the PTO torque acting on the gyroscope shaft. The accumulators -5- and -6- implemented in the model are gas-charged accumulators. The gas is considered to behave isentropically:

$$pV^\gamma = const. \tag{4}$$

The two main parameters of the accumulators are the pre-charge pressure p_0 and the total capacity V_0. The pre-charge pressure is the pressure inside the accumulator when it is filled with gas only. The capacity is the total volume inside the accumulator, sum of the liquid and gas contributions: the higher the capacity, the higher the dampening effect on the pressure values. The accumulator model takes as input the mass flow rates from the pump and the motor in order to compute the volume of liquid stored:

$$V_{HP}(t) = \int_0^t (Q_p - Q_m)dt \tag{5}$$

Where:
- V_{HP} is the volume inside the high-pressure accumulator;
- Q_p is the pump flow rate rectified by the hydraulic Greatz bridge -4-;
- Q_m is the motor flow rate.

As far as the low-pressure accumulator goes, the equation is analogous with opposite signs.

Knowing the volume of liquid inside the accumulator, one can evaluate the volume occupied by the gas and therefore the pressure of both gas and liquid. The accumulators, through the following equation, set the pressures in the circuit:

$$p(t) = \frac{p_0 V_0^\gamma}{(V_0 - V(t))^\gamma} \tag{6}$$

The working equations of the motor are the same as the pump ones:

$$D_m(p_C - p_D) = T_{gen} + (I_{motor} + I_{gen})\dot{\omega}_m \tag{7}$$

$$Q_m = D_m \omega_m \tag{8}$$

Where:
- T_{gen} is the resistive torque of the electric generator;
- D_m is the motor displacement
- I_{motor} and I_{gen} are the motor and generator moments of inertia respectively
- p_C and p_D are the pressures of the fluid at the motor ends
- ω_m is the motor-generator angular speed.

Fig. 5 allows appreciating the dampening effect of the accumulators by comparing the power absorbed from the pump, highly irregular, with the power produced by the generator, quite smoother.

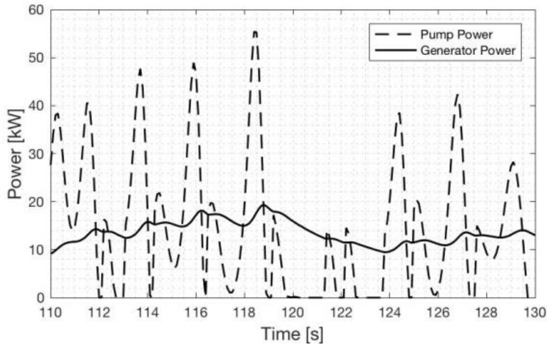

Fig. 5 Power absorbed and power generated

Similar considerations can be done by looking at fig. 6 as well, where the PTO torque trend is shown as a roughly constant-module square wave. Since the pump torque is proportional to its pressure drop, one can realize that the pressures are not actually constant throughout the simulation, but they keep a certain degree of variation: this is a consequence of the finite capacity of the accumulators.

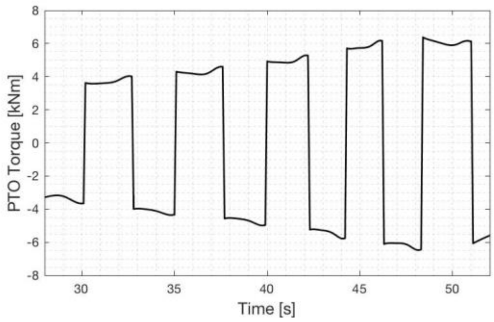

Fig. 6 PTO torque trend

The pressures at ports A and B of the pump continuously switch between the high and low values depending on its spinning direction.

The generator -9- exerts a torque that is proportional to its rotational speed, so that it is always opposed to the spinning direction, which accounts for resistive power only:

$$T_{gen} = \beta\omega_m \quad (9)$$

The coefficient β works as a damping term: for both control strategies it is kept constant for a given sea-state and it is set in order to optimize the power production. The model of the generator torque is what underlies the proportional control.

Eventually, there are three types of hydraulic valves in the PTO: the check valves making up the rectifier bridge -4-, the declutching valve -3- and the pressure relief valve -7-. The pressure relief valve opens when a pressure limit threshold is reached and stays open until the pressure itself has dropped down by a given margin, defined as a percentage of the threshold. When the relief valve is open, part of the liquid flows from the high pressure to the low-pressure ram bypassing the motor and thus accounting for a difference between the power absorbed at the pump side and the power produced at the generator side. The rectifier bridge is simply modelled as an absolute value applied to the pump flow rate, so that the flow on the motor side of the circuit is unidirectional. The declutching valve, when it is actuated, is modelled to bring the pump flow rate and the pressure drop across it to zero.

B. *Hydraulic PTO Control Law*

Two different sea-state dependent control laws have been applied to the model and compared to each other. The first is a proportional control on the pressures achieved by keeping the generator torque proportional to its angular velocity. The damping coefficient of the generator torque is the parameter to be tuned. A relationship is found between the damping coefficient and the mean power production and an optimal value of the former is computed for each incoming wave.

The second control law combines the previous strategy with a sub-optimal declutching control that consists in clutching the pump when its speed exceeds a threshold value and declutching it when the pump inverts its spinning direction. One can refer to Fig. 7 to better understand the declutching control logic.

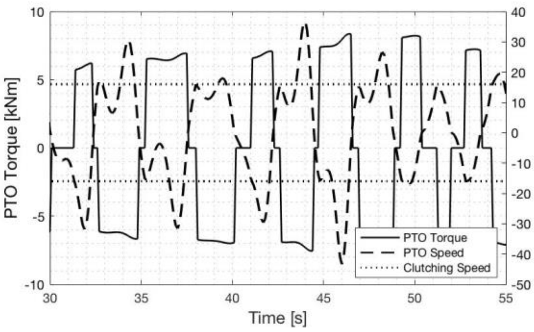

Fig. 7 Declutching control law: PTO torque and pump speed

The parameters to be tuned are the damping coefficient of the generator and the clutching speed, and an optimal combination of the two parameters is found for each sea-state studied.

The difference between the two control laws, as shown in Fig. 8, derives mainly from the diverse trend of the PTO torque. When the declutching valve is not used, the pump reaction torque behaves roughly like a square wave. When the declutching logic is applied, on the other hand, for some parts of the wave cycle (that is, when the pump is declutched from the rest of the system) the PTO torque is null.

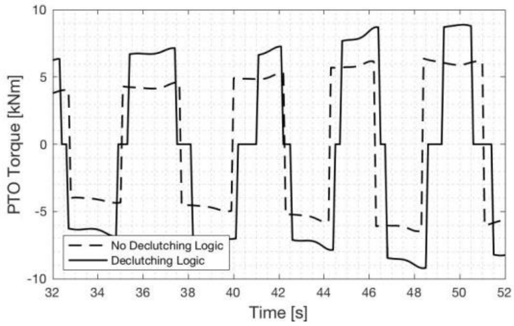

Fig. 8 PTO torque trend for both control laws

This allows achieving more control over the torque delivered by the PTO on the gyro. In fact, for the parts of the cycle where the kinetic energy of the gyroscope is low, no torque is applied, excluding the pump from the hydraulic transmission. On the other hand, when the speed of the gyroscope exceeds the clutching speed value, the PTO acts on the gyroscope with a braking torque.

Both control laws are passive strategies that require an evaluation of the current sea-state in order to tune the PTO parameters to the optimal values.

V. Numerical Results

Parametric simulations have been done in order to evaluate the performance of the hydraulic transmission and to compare the two control strategies employed. The simulations aimed to obtain the optimal control-parameters for both control logics. The comparison is eventually made on the optimized configurations employing the gross productivity as comparison parameter. The simulations have been run on 21 real sea waves, whose profile had been generated based on the JONSWAP (Joint North Sea Wave Project) spectra and annual occurrences of the Adriatic offshore seaside.

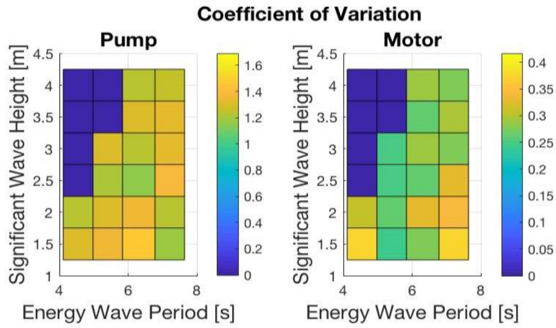

Fig. 9 Power Coefficient of Variation: Pump (left) and Motor (right)

By looking at Fig. 9 one can evaluate the performance of the hydraulic transmission when it comes to dampen out the irregularities of the input power. Such figure plots the coefficients of variation (COFs) of both pump and generator power, defined as the ratio between the standard deviation of the power and its average value:

$$COF = \frac{\sigma_P}{\overline{P}} \tag{10}$$

Results are plotted as a function of the energy wave period and the significant wave height. The blue region in the up-left corner concerns waves that have not been studied. The dampening effect is clear: the pump power COF lays on an average of 1.30, with peaks up to 1.69, while on the motor side the average value drops down to 0.31 and the peaks are kept under 0.42.

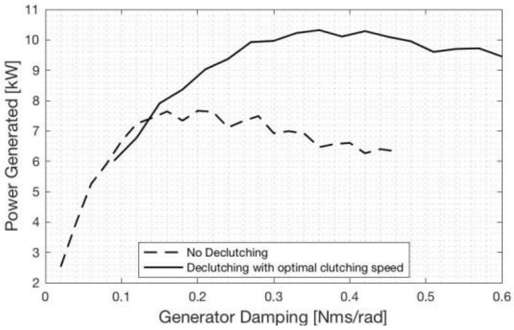

Fig. 10 Generator power vs generator damping coefficient: no declutching (dashed line) and declutching logic applied (solid line)

Fig. 10 allows to appreciate the influence of both the generator damping and the declutching logic on the mean power production for a given wave. The dashed line underlies the proportional control by plotting the relationship between the generator damping coefficient and the power production. The solid line, on the other hand, shows how the declutching logic improves the overall power production and shifts the optimal range of the generator damping to higher values. In particular, the higher the clutching speed, the higher the optimal value of the generator damping that maximizes the power production. This was actually expected: since the declutching valve brings the PTO torque to zero for some parts of the wave cycle, it makes it necessary to increase its value when the pump is clutched in order to keep a similar torque effective value over the cycle. Different curves are obtained varying the clutching speed, and an overall optimal working point is found.

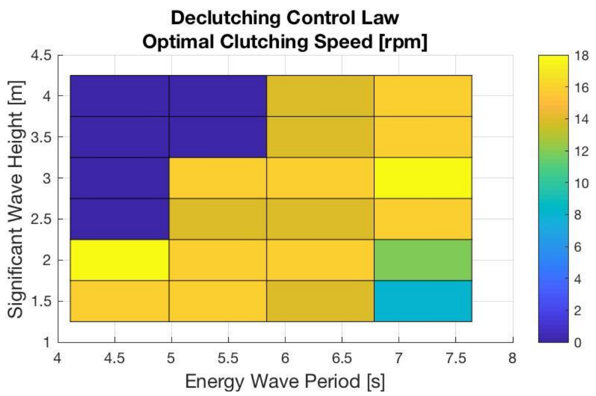

Fig. 11 Optimal Clutching Speed [rpm]

In Fig. 11 the optimal clutching speeds are reported. The plot shows that the optimal values remain all in a pretty narrow range, mainly between 14 and 16 rpm. This suggests for the employment of two different declutching logics. The first implies a constant clutching speed for all spectra, set to an average value estimated considering all the sea states investigated and their occurrences. The second is a more advanced control that adjusts the clutching speed tuning it to the current sea state so that full optimisation with every incoming wave can be achieved. In both cases the generator damping coefficient is set to its optimal value.

Synthetic results comparing the two declutching control laws with the proportional strategy are shown in Fig. 12 and Fig. 13: the plots show, for each wave investigated, the ratio between the mean power produced out of the declutching control law and the mean power produced out of the proportional control.

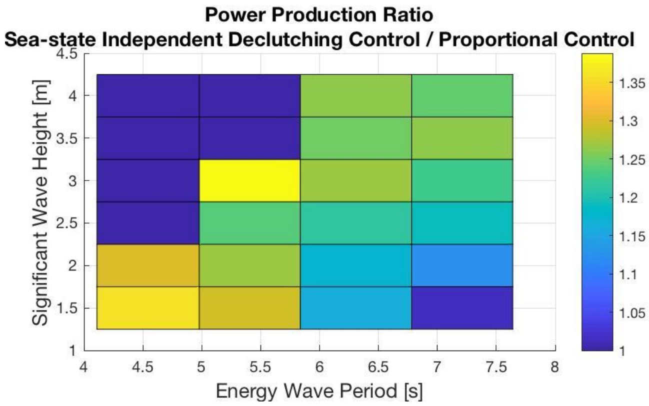

Fig. 12 Power Production: Sea-state independent Declutching Control over Proportional Control

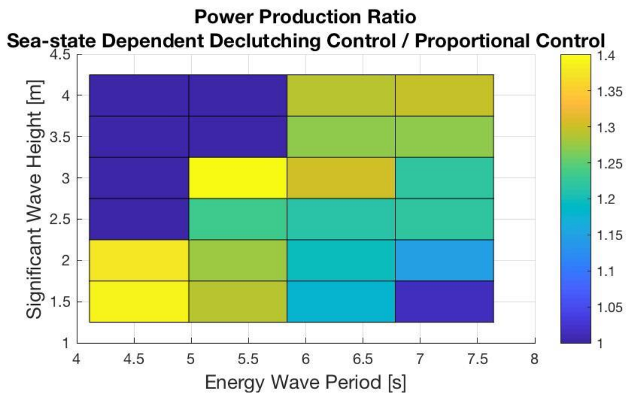

Fig. 13 Power Production: Sea-state dependent Declutching Control over Proportional Control

Fig. 12 concerns the sea-state independent logic with a constant clutching speed, while Fig. 13 concerns the sea-state dependent law with an adjustable clutching speed. The figures show how in both cases the declutching control allows to improve the power production for every sea-state considered, up to 38% with a constant clutching speed and up to 40% with an adjustable clutching speed. On average, the mean power production is increased by a 22% in the former case and by a 24% in the latter case.

VI. Conclusions

Due to the necessity of handling high torques and forces, in wave energy extraction systems hydraulic PTOs are nowadays becoming a common choice, and many developers have approached this solution. Still, there are a number of different possibilities in terms of both the PTO configuration (constant-pressure, variable-pressure, variable-constant pressure) and, most of all, in terms of the control strategies applicable (latching control, declutching control, discrete displacement control, etc.).

In the present work, a constant-pressure hydraulic PTO has been coupled to the ISWEC in order to investigate its performance. Two control laws have been applied and tested on the Simulink model of the WEC: the proportional control and a combination of the former with a sub-optimal declutching control. Both control laws are passive and sea-state dependent: a prevision of the incoming wave would be necessary for implementation on the full-scale device.

First, results show how a constant-pressure hydraulic transmission configuration improves the power quality of the WEC by smoothing the power peaks. In fact, the hydraulic accumulators coupled with a rectified bridge allow obtaining an average COF of 0.31 on the motor side compared to an average of 1.30, with peaks up to 1.69, on the pump side.

Moreover, the declutching control law, in comparison to a simple proportional control, allows to improve substantially the power production, up to 40% in the best case scenario. As already stated in [19], this result shows that allowing the pump to accelerate for some parts of the wave cycle leads to higher energy absorption, even though in those moments, the generator is decoupled and no energy is produced. On the other hand, the results of the present study show how it is not strictly necessary to employ an optimal command theory to compute the times of the declutching valve shifts, but a simple sub-optimal strategy with a sea-state dependent clutching speed is good enough to achieve considerable improvements.

Most importantly, it is shown how a constant clutching speed for all the sea states studied is able to perform similarly enough to a more complicated law that adjusts the clutching speed whenever the sea-state changes. This might lead to the implementation of a declutching logic independent of the sea state considered, that does not need a wave prediction algorithm, if not for the tuning of the generator damping.

Future works will focus on a more advanced control to consider both extreme and calm sea conditions. For what concerns the extreme wave conditions, the high-pressure accumulator will be overcharged and so a proper control of the motor speed will be necessary to discharge it quickly, employing the relief valve only in emergency conditions. Likewise, calm sea conditions must also be managed properly, avoiding the high-pressure accumulator to complete emptying and thus losing its damping effect. Moreover, a more advanced control will improve the COF of the extracted power, avoiding the use of super-capacitors for the electric power managing. Eventually, the hydraulic PTO model will be improved adding the contribution of tubes, valves, pump and motor losses and the effect of the oil compressibility in order to evaluate their influence on both the power production and circuit dynamics.

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