Offshore Floating Wind Turbine-driven Deep Sea Water Pumping for Combined Electrical Power and District Cooling

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Abstract: A new concept utilising floating wind turbines to exploit the low temperatures of deep sea water for space cooling in buildings is presented. The approach is based on offshore hydraulic wind turbines pumping pressurised deep sea water to a centralised plant consisting of a hydro-electric power system coupled to a large-scale sea water-cooled air conditioning (AC) unit of an urban district cooling network. In order to investigate the potential advantages of this new concept over conventional technologies, a simplified model for performance simulation of a vapour compression AC unit was applied independently to three different systems, with the AC unit operating with (1) a constant flow of sea surface water, (2) a constant flow of sea water consisting of a mixture of surface sea water and deep sea water delivered by a single offshore hydraulic wind turbine and (3) an intermittent flow of deep sea water pumped by a single offshore hydraulic wind turbine. The analysis was based on one year of wind and ambient temperature data for the Central Mediterranean that is known for its deep waters, warm climate and relatively low wind speeds. The study confirmed that while the present concept is less efficient than conventional turbines utilising grid-connected electrical generators, a significant portion of the losses associated with the hydraulic transmission through the pipeline are offset by the extraction of cool deep sea water which reduces the electricity consumption of urban air-conditioning units.

1. Introduction

The energy demand to cool buildings for thermal comfort in Europe is expected to increase by three times the 2006/7 levels by the year 2020 [1]. Existing offshore wind turbine technology is based on well-established land-based technologies and is only capable of producing electrical energy. There is an ever growing interest to exploit the enormous wind resources available in deep waters through the commercialisation of floating wind turbine technologies [2]. Deep water sites with sea depths typically exceeding 150 m also offer an enormous potential for renewable thermal energy through the formation of thermal gradients with sea depth; also known as thermoclines. The sea temperature in

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the deeper water layers is stable and significantly lower than ambient temperatures in various countries’ urban areas that typically demand significant amounts of energy for cooling of buildings.

Current research is exploring the viability of hydraulic wind turbine concepts, whereby electrical generators on conventional machines are replaced by positive displacement pumps; with the turbines pumping sea water at high pressures to a centralised hydro-electric power station \cite{3, 4} instead of generating electrical energy. Apart from potentially reducing costs by minimising the use of copper and rare earth materials required for electrical systems, this new approach will facilitate the integration of wind farms with hydro-energy storage systems thus mitigating problems associated with grid congestion and stability. Recent numerical modelling at the University of Malta has demonstrated the potential of using hydraulic wind turbines to exploit deep sea water (DSW) for thermal energy production in conjunction with electricity generation \cite{5, 6}. This principle, referred to as the Offshore Wind and Thermocline Energy Production (OWTEP) concept, involves the use of hydraulic wind turbines which pump deep sea water through a pipeline to a land-based plant consisting of a hydro-electric power installation coupled to a district cooling/heating system. In the preliminary analysis, which involved the simulation of a single offshore turbine for Central Mediterranean conditions \cite{5}, DSW exiting the hydro-electric plant was modelled to flow through a counter-flow heat exchanger to heat or cool water flowing from a district heating/cooling system. The study showed that significant amounts of thermal energy can theoretically be supplied by the OWTEP concept offering improved potential for floating wind energy technology in regions such as the Mediterranean, where wind conditions are not as favourable as in the North Sea and where large coastal urban areas rely heavily on electricity for cooling as a consequence of warm climatic conditions.

One of the main concerns of DSW cooling is its effect on the ecology of the body of water that is utilised as a heat sink. Abbasi and Abbasi \cite{7} observed that the thermal structure, salinity gradients and quantities of dissolved gases and nutrients can all be affected. However, these issues become particularly significant when pumping from depths in excess of 1000 m. Regarding the thermal structure, Chen et al. \cite{8} implemented a simplified two-dimensional model and showed that impacts of the discharge water on the overall temperature and overall ecology are acceptable. Briand \cite{9} analysed the effect of cooling systems in phytoplankton mortalities and advocates the use of deep-sea water for cooling since phytoplankton stocks are unaffected for intake temperatures lower than 15 °C. From an overall perspective, Looney and Oney \cite{10} describe DSW cooling in oceans, seas or lakes as a sustainable and reliable source of energy. Moreover, typical power plants using seawater cooling are allowed to reject water at up to 15 °C higher than their intake temperature \cite{11}, which is higher than that expected from the OWTEP system.

A more detailed investigation of the OWTEP concept is now being presented. This new analysis integrates a steady-state model for a single OWTEP turbine with that of a centralised sea water-cooled air-conditioning (AC) unit operating with a vapour compression cycle to chill water from a district cooling system. The main scope is to evaluate the energy savings achieved by having an AC unit utilising deep sea water from a single OWTEP turbine in lieu of sea surface water (SSW). A case study is presented based on a deep offshore (200 m) site in Maltese waters in the Central Mediterranean basin. The study will shed more light on the viability of integrating the OWTEP concept at deep offshore sites with large-scale air conditioners for district cooling applications in coastal urban environments.

2. Outline Description of the Combined OWTEP – AC System

Figure 1 illustrates the combined OWTEP-AC system modelled in this study. A single offshore hydraulic wind turbine pumps DSW at high pressure to an onshore hydro-electric power unit that consists of a Pelton wheel and synchronous generator. The generated electricity is primarily fed to the
Figure 1. The OWTEP wind turbine coupled to the large-scale AC unit and district cooling systems

grid, although a fraction is also consumed by the AC unit to drive the vapour compression cycle. The DSW from the OWTEP turbine is then fed into the condenser of the AC unit. Given that the supply of DSW is intermittent, depending on the availability of the wind at the offshore site, the AC unit also makes use of sea surface water (SSW) through an auxiliary make-up water pump to offset any shortfall in the supply of DSW, hence maintaining a constant sea water flowrate within the condenser.

3. Model for OWTEP System Performance

The analysis is based on a steady-state numerical model of a single-turbine OWTEP system developed by Buhagiar [6]. It includes a set of empirical equations that model the performance of the rotor, pump, pipeline and hydroelectric turbine.

3.1 OWTEP Mechanical/Hydraulic Model

Rotor modelling utilises discontinuous functions of angular velocity and torque with respect to wind speed. Variations in the control strategy corresponding to different regions of operation result in the respective angular velocity-wind speed relationships. These relationships [6] are derived for the standard wind turbine control scheme of aerodynamic optimization up to the rated wind speed and power regulation with pitch control between rated and cut-out wind speed conditions [6]. Rotor torque is computed as a function of the square of the angular velocity; a concept discussed by Laguna [3]. The use of a variable displacement pump allows optimization of the torque loading on the pump for the rotor operating state. The torque required to rotate the pump is regulated by adjusting the volume of fluid displaced per revolution of the pump as proposed by Diepeven and Laguna [3, 4]. This is analogous to generator torque control. Given that the OWTEP model is limited to steady-state operation, the torque developed by the rotor and that driving the pump are assumed to be equal.

Modelling of the pump is based on the method of lumped-parameters, as developed by Dasgupta and Mandal [12]. OWTEP pump efficiencies are described in terms of three coefficients that individually describe the damping, frictional and slip losses within the pump. These coefficients are constant throughout the entire operating range of the pump and are used to compute the instantaneous flow-rate and corresponding torque requirement. The driveshaft connecting the pump to the rotor is considered to be perfectly rigid and to induce negligible frictional losses. This implies that the rotor torque generated is entirely transferred to the pump. The frictional load induced on the pump by the pipeline is computed using the Darcy-Weisbach formula [13]. The Haaland approximation [14] is utilised to obtain an explicit formulation of the Colebrook and White formula for the turbulent friction factor. As indicated in Fig. 1, the pressurized DSW from the OWTEP pipeline is converted into electricity
through a hydro-electric power unit. It is assumed that this consists of a Pelton turbine coupled to a grid-connected fixed-speed generator. A spear-valve is utilised to adjust the area of the nozzle depending on the flow rate, and to maintain a fixed angular velocity at an optimum bucket-speed ratio over the range of operational wind speeds. In order to simplify the steady-state model, the electrical generator and driveshaft are modelled as having fixed efficiencies (99% and 95% respectively).

A boost pump is included at the base of the wind turbine to pump the DSW to hub height. A conceptual system is considered in which the boost pump draws power from the high-pressure line. Based on the typical efficiency of the selected boost system components, the efficiency of this system is taken to be fixed at 80% [6]. The total pressure load induced on the pump is given by the sum of the nozzle pressure, pipeline frictional load, pressure drop across the boost system and the head corresponding to the Pelton wheel elevation above mean sea level.

3.2 The OWTEP Thermal Model

A thermal model is implemented to model the temperature rise in DSW as it flows through the wind turbine pump and pipeline. The pipeline is reduced to an internal forced convection problem, having a known inlet temperature and flow rate. The pipeline is discretised into one-dimensional tubular elements having an inlet and outlet temperature. The surrounding fluid temperature is constant for each element and is obtained from the National Oceanography Data Centre (NODC) temperature-depth profiles [15]. The thermo-fluid properties of seawater are calculated for each element using correlations developed by Sharqawy et al. [16]. Heat transfer processes are categorised into internal forced convection, conduction and external free-convection. Conduction is modelled by the application of Fourier’s law of conduction to a cylindrical surface. Internal forced convection is modelled using the Gnielinski empirical formula [17], which gives the Nusselt number at the internal flow-pipeline interface. External free convection is modelled using a set of correlations by Heo and Chung [18]. The temperature rise of the seawater as it passes through the pump was calculated to be negligibly small (≤ 0.2°C) and is therefore not included in the calculations. The model takes into consideration the dependency of the sea water viscosity on temperature at each pipeline element. It has been shown that this approach implicitly accounts for entropy generation across the pipeline [19].

4. Model for AC Unit Performance

Figure 2 illustrates a schematic diagram and the main parameters for the large-scale air-conditioning (AC) unit considered in this study. The refrigerant in the condenser is cooled using sea water which may consist of deep sea water (DSW) pumped by the OWTEP turbine \( T_{SW,1}=T_{DSW} \) or sea surface water (SSW) pumped through an auxiliary pump \( T_{SW,1}=T_{SSW} \). As already explained in Section 2, the condenser may also rely on a mixture of both DSW and SSW. In this case, the input sea water temperature at the condenser can be estimated from the energy conservation law:

\[
T_{SW,1} = -273 + \frac{(T_{DSW} + 273) \cdot Q_{DSW} + (T_{SSW} + 273) \cdot Q_{SSW}}{Q_{DSW} + Q_{SSW}}
\]  

(1)

where all temperatures are in deg. Celsius (°C) and not in Kelvin (K). A simple vapour compression cycle with R134a refrigerant is assumed, as plotted in Figure 3. Thermodynamic losses associated with the isentropic compression process 1-2 are ignored. It is also assumed that the refrigerant is not superheated, hence only a condensation process at constant temperature occurs in the condenser. The refrigerant is a saturated liquid at the condenser exit before flowing through the throttling valve. Pressure drops in the system are also ignored. The steady-state performance of the AC unit for a specified sea water flow rate \( Q_{SW} \) and temperature \( T_{SW,1} \) is computed by first applying the NTU method [20] for the heat exchange process in the condenser (see Figure 4). The temperature gain in the sea water across the condenser, \( \Delta T_{SW,12} \) is fixed.
The thermal effectiveness $\varepsilon$ of the condenser is computed from:

$$\varepsilon = 1 - e^{-NTU} \quad NTU = \frac{UA}{\rho \dot{Q}_{SW} c_{SW}} \quad (2)$$

where $U$ and $A$ are the condenser heat transfer coefficient and area respectively. The condensation temperature of the refrigerant is then found from:

$$T_{2r} = T_{3r} = T_{SW,1} + \frac{q_H}{\varepsilon \rho \dot{Q}_{SW} c_{SW}} \quad (3)$$

The enthalpy across an ideal throttling valve is constant; hence $h_4 = h_3r$. The mass flowrate of refrigerant $\dot{m}_r$ in kg/s and the net compressor work and cooling load in Watts are determined using:

$$\dot{m}_r = \frac{q_H}{h_2r - h_3r} \quad W = \dot{m}_r (h_{2r} - h_{1r}) \quad q_L = \dot{m}_r (h_{1r} - h_{4r}) \quad (4)$$

The study considers a district cooling system using water as a medium for cooling buildings. In the present model for the AC unit, it is assumed that the water is chilled by the evaporator of the AC unit from a temperature $T_{FW,1}$ down to $T_{FW,2}$ which is fixed at 12 °C (see Figure 2). Existing district cooling systems have a set of technical requirements that consumers connected to the system must meet. These often include a clause regarding the return temperature of the fluid to district system. Typical
scenarios require the district system to supply the cool water at for example 6.5 ±0.5 °C and the user must return the liquid at 15 ± 1 °C [21, 22]. Such parameters are subject to monitoring at 30 minute intervals so that the cooling load is constantly being monitored. In the current study thermal loads representing the consumers could not be so easily quantified. Instead, the ambient temperatures were fed into the model as the return temperature to the warm side of the district heat exchanger. Using a fixed outlet temp of 6 °C, as in typical district system, would imply that the temperature drop across the heat exchanger would be much higher since local ambient temperatures are well above 15 °C. Hence the fixed outlet temperature was shifted to 12 °C so as to allow for the use of ambient temperatures at the return side of the heat exchanger, while maintaining a temperature drop (and therefore heat transfer rate) that corresponds to typical district systems [23].

The flowrate of water from the district cooling system $Q_{FW}$ is allowed to vary depending on the values of input parameters $Q_{SW}$, $T_{SW,1}$ and $\Delta T_{SW,12}$. The efficiency of air-conditioning systems is quantified by computing the coefficient of performance ($COP_c$) which for cooling process is defined as [20]:

$$COP_c = \frac{h_{1r} - h_{AF}}{h_{2r} - h_{1r}}$$  \hspace{1cm} (5)

The $COP_c$ values for vapour compression AC units lies typically in the range of 2 – 6.5, with the geothermal and sea-water cooled systems being at the higher end of the range. The use of DSW in lieu of SSW lowers the condenser temperature, hence also lowering the work load on the compressor and augmenting the heat exchange in the condenser. The end result is an increase in the $COP_c$ value, as may be noted by the dashed lines in Figure 3.

5. Definition of Design and Environmental Parameters for the Combined OWTEP-AC System

The OWTEP system considered in this study uses the hypothetical NREL 5MW reference rotor [24]. This has a 3-bladed, 126 m diameter rotor, which is typically used as reference when comparing different wind turbine technologies. The constant loss coefficients for the wind turbine pump were computed for the nominal parameters shown in Table 1. Turbine operation was modelled in Maltese waters at an offshore site having a sea depth of 200 m and at a distance of 9.5 km from the shoreline. The design parameters of the OWTEP pipeline and the land-based hydro-electric unit considered in the study are presented in Table 2. The nominal design parameters of the AC unit used in the analysis are given in Table 3. The following three independent sources of environmental data were used to simulate the combined OWTEP-AC system on an hourly basis over a duration of one year:

1. Hourly wind data at the offshore wind turbine location and hub height of 90 m. These were extrapolated from the 80 m level of a land-based tubular, guyed meteorological mast located at the shore line at Ahrax Point in Malta [25]. The data, covering the period January 2007 – December 2011 was extrapolated to a point 9.5 km offshore using the CFD model WindSim™ [26]. However, only data for the year 2011 is being used in this paper. The annual average wind speed for the year 2011 at the offshore site was predicted to be equal to 6.71 m/s by the CFD model. This was found to be 0.3 m/s higher than that measured at 80 m above ground level at Ahrax Point during the same year. It is also around 8% lower than the five year average value for the offshore site under consideration.

2. Hourly temperature measurements at 3.5 m above ground level for January – December 2011 from an urban location in Hal Far, Malta. It was assumed that the hourly temperature values of the water from the district cooling system (denoted by $T_{FW,1}$, see Fig. 2) are equal to these ambient temperature measurements.

3. Monthly-averaged temperature-depth (thermocline) profile data from the National Oceanography Data Centre (NODC) [15], taken from the nearest data point on a 5 deg. mesoscale modelling grid.
Air density was assumed constant at 1.21 kg/m$^3$, based on the average temperature. The selected 134a refrigerant is set to be phased out in the European Union due to its high Global Warming Potential (GWP) [27]. Other refrigerants such as HFO-1234yf will be introduced to replace it. Such a refrigerant will exhibit nearly identical behaviour with around 99.7% less GWP [27]. Future models will be modified to use HFO-1234yf when more detailed data is available. The OWTEP pipeline was discretised into 1,523 elements in the thermal model described in Section 3.2 to enable computation of the deep sea water temperature $T_{DSW}$ at the pipeline exit. Figure 5 plots the monthly average wind speeds and temperatures used in the OWTEP model, with Month 1 denoting January.

### Table 1. Design Parameters of the Conceptual OWTEP pump.

| Parameter                                      | Value       |
|------------------------------------------------|-------------|
| Nominal Pressure Load                         | 150 bar     |
| Nominal Angular Velocity                      | 12.1 rpm    |
| Mechanical Efficiency at Nominal Conditions   | 97%         |
| Volumetric Efficiency at Nominal Conditions   | 98%         |
| Angular Velocity at No Load                   | 50 rpm      |
| Power Consumption at No Load                  | 30kW        |
| Maximum Volumetric Displacement               | 1.5 m$^3$/rev|
| Minimum Volumetric Displacement               | 0.05 m$^3$/rev|

### Table 2. Design Parameters of the OWTEP pipeline and Hydro-Electric Unit.

| Parameter                                      | Value       |
|------------------------------------------------|-------------|
| Pipeline Length                                | 9.5 km      |
| Pipeline Diameter                              | 0.5 m       |
| Pipeline Thickness                             | 75 mm       |
| Pipeline Material                              | Glass Reinforced Epoxy/Plastic |
| Pipeline Surface Roughness                     | 5 µm        |
| Pelton Wheel Diameter                           | 8 m         |
| Pelton Wheel Speed                              | 200 rpm     |
| Pelton Wheel Elevation                         | 20 m (a.m.s.l.) |
| Bucket friction coeff. and splitting angle     | 0.95, 15°    |
| Spear valve coeff. of velocity/contraction     | 0.99, 0.99  |

### Table 3. Design Parameters of the AC unit

| Parameter                                      | Value       |
|------------------------------------------------|-------------|
| Refrigerant                                    | R134a       |
| Condenser Area                                 | 1414 m$^2$  |
| Condenser Heat Transfer Coefficient            | 1000 W/m$^2$/K |
| Temperature difference across condenser ($\Delta T_{SW,12}$) | 10 °C |
| Evaporator Temperature of Refrigerant          | 6 °C        |
| Outlet Temperature of District Cooling Water ($T_{FW,2}$) | 12 °C |
The monthly average values for the DSW and SSW temperatures derived from the NODC database are also included. The differences in these two temperature values during the summer period are evident and represent the renewable energy potential for cooling available through DSW extraction.

![Figure 5](image1.png) Monthly average wind speed, ambient temperature, SSW and DSW for the year 2011.

![Figure 6](image2.png) Monthly average thermocline profiles for the Central Mediterranean [10].

6. Results and Discussion

The flowrate- and pump pressure-wind speed characteristics for the modelled single OWTEP turbine are shown in Figure 7. The flowrate increases with wind speed up to the rated wind speed, beyond which it remains constant and equal to 0.32 m³/s. The OWTEP system maintains a nearly constant pressure with wind speed at the pump, increasing only marginally from 155 to 158.4 Bar at higher wind speeds. As may be noted in Figure 8, the electrical power from the hydro-electric unit at each wind speed is less than that of a conventional 5 MW turbine generator having the same rotor diameter. This is due to the frictional losses in the pipeline which reduce the transmission efficiency to between 79 and 82%. The OWTEP flowrate-wind speed characteristic in Figure 7 was prescribed to the AC unit model described in Sections 4 and 5 (Table 3) to investigate the variation of performance parameters with wind speed. It was initially assumed that the AC unit is only supplied directly with DSW, with the SSW pump (Figure 2) off. The results are plotted in Figures 9 – 12. The characteristics are plotted for three assumed values of \( T_{SW,1} \). Lower values of \( T_{SW,1} \) are observed to improve the AC unit performance, as a consequence of reduced power consumption by the compression process (process \( 1r-2r \) in Figure 3). This trend is observed at all wind speeds. As may be noted in Fig. 11, highest \( COP_c \) values at each value of \( T_{SW,1} \) are attained at low wind speeds, decreasing to a constant value at and above the rated wind speed. Lower wind speeds result in lower values of \( Q_{DSW} \) hence improving the thermal effectiveness of the condenser. The corresponding \( COP_c \) values obtained from an ideal Carnot cycle computed using the formula \( COP_c = T_1 / (T_2 - T_1) \) are also included in Fig. 11. The coefficient of performance computed for the thermodynamic cycle in Fig. 3 was found to be around 12 to 16% lower than the ideal Carnot value, depending on wind speed and the intake water temperature \( T_{SW,1} \). The mass flowrate of refrigerant \( (m_s) \) and volume flowrate of water \( Q_{FW} \) from the district cooling network (see Figure 2) were found to vary with wind speed with similar trends to \( Q_{DSW} \), with the dependency on the \( T_{SW,1} \) being very small.

Analysis using the one year data for Maltese conditions (Jan. – Dec. 2011) described in Section 5 was later conducted by prescribing the hourly predictions for \( Q_{DSW} \) and \( T_{SW,1} \) from the OWTEP model to the AC unit model. For each hour, the temperature of the district cooling water entering the evaporator \( (T_{FW,2}, \text{Figure 2}) \) was assumed equal to the ambient temperature measured at Hal Far, Malta. All other variables were kept fixed, as defined in Table 3. In order to be able to quantify the energy saved by utilising DSW supplied by the OWTEP system, three different system modes of operation were analysed for the AC unit shown in Figure 1. These are summarised in Table 4.
Figure 7. Variation of flowrate and pump pressure with wind speed for OWTEP turbine.

Figure 8. Comparison of electrical power characteristic of OWTEP turbine with conventional turbine of equal rotor diameter.

Figure 9. Variation of cooling load generated with wind speed at three sea water input temperatures.

Figure 10. Variation of net consumed power with wind speed at three sea water input temperatures.

Figure 11. Variation of the coefficient of performance with wind speed at three sea water temperatures.

Figure 12. Variation of the sea water exit temperature at the condenser with wind speed at three sea water temperatures.
Considering considerably lower (in the region of 2.6 – 7.8%). For (i.e. at 0.32 m$^3$/s). Temperature $T_{SW,1}$ is determined using Eqn. (3). In System B2, the AC unit is only reliant on DSW, with the SSW pump switched off. Consequently, the AC unit is operated intermittently depending on the availability of the wind offshore and the refrigerant flowrate ($\bar{q}_L$) is regulated accordingly. In the three systems, the AC unit is only operational when the ambient temperature is less than or equal to 22 °C, which is a realistic temperature to maintain a comfortable temperature in buildings.

The monthly average predictions for $T_{SW,1}$ and $T_{SW,2}$ are given in Figures 13 and 14. The latter parameter is of particular relevance when assessing the environmental impacts of sea-water based AC units on marine habitats. As may be observed in Figure 14, the use of DSW limits the exit temperature. This reduces risks associated with thermal pollution. The predictions for the cooling load generated ($q_L$) and net electricity ($W$) consumed by the AC unit per month for the three system modes are presented in Figures 15 and 16. $q_L$ and $W$ naturally peak during the summer period. Whilst values of $q_L$ for Systems A and B1 are almost equal, the net electricity consumed by System B1 is considerably lower (in the region of 2.6 – 7.8%). For System B2, $q_L$ and $W$ are significantly smaller because the AC unit is operating intermittently depending on the available wind offshore. Figures 17 and 18 are scatter plots for the instantaneous hourly values of $COP_c$ from Systems B1 and B2. The dependency of the $COP_c$ on wind speed for the two systems is considerably different. At very low wind speeds, below the cut-in wind speed of the wind turbine, the $COP_c$ value for System B2 is zero. At high wind speeds exceeding the rated value, the $COP_c$ values for both Systems B1 and B2 are equal given that the OWTEP wind turbine reaches full capacity in supplying DSW, hence the SSW pump in System B1 is off. At low wind speeds the $COP_c$ for System B1 is inferior and, unlike that for System B2, is also significantly dependent on the SSW temperature. The different curve formations at low wind speeds in Figure 17 result from different monthly average values for the SSW assumed in the model. The annual average $COP_c$ values for the three different systems were determined by simply dividing the annual sum of the hourly values of $q_L$ with the corresponding value for $W$. The results are summarised in Table 5 below together with the electricity production from the OWTEP turbine. The gross annual electricity consumed by the AC-unit is computed by assuming a constant electrical efficiency value of 80% ($\eta_e$) for the motor driving the refrigerant compressor. The electricity consumption of the SSW pump (Figure 2) is neglected as this was found to be very small. The values for the annual average $COP_c$ presented in Table 5 are somewhat higher than those observed in reality. This is because only a simply refrigeration cycle is being assumed, neglecting pressure losses in the AC unit as well as thermodynamic losses during the compression process of the refrigeration cycle. In addition, the electrical losses of the motor driving the compressor are not accounted for (through the inclusion of $\eta_e$) in the $COP_c$ computation.

It is estimated that, based on the one year wind data, the net electricity produced by the single turbine OWTEP system (11.21 GWh) is 9.1% less than that generated by a 5 MW conventional offshore turbine generator (12.33 GWh). See the power curve in Figure 8. The comparison assumes electrical losses amounting to 6% to transmit the electricity from the conventional turbine to the onshore grid. The deficit in the OWTEP turbine production of electricity is mainly due to the frictional losses.

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| System | OWTEP | SSW Pump | $COP_c$ | $T_{SW,1}$ |
|-------|-------|----------|--------|-----------|
| A     | OFF   | ON, Full Load | SSW, Full Load | $T_{SW}$ |
| B1    | ON    | ON, Intermittent | DSW+SSW, Full Load | Eqn. (3) |
| B2    | ON    | OFF       | DSW, Intermittent | $T_{DSW}$ |

**Table 4. Three System Operational Models Investigated**

System A represents a conventional AC unit which only relies on SSW to cool the refrigerant in the condenser. The SSW pump is assumed to operate at full load with a maximum flowrate of 0.32 m$^3$/s - equal to the maximum capacity of the OWTEP pump (see Figure 7). In System B1, the sea water consists of a mixture of DSW from the OWTEP turbine and SSW, with the SSW pump operating intermittently depending on wind speed to maintain a constant flowrate $Q_{SW}$ at the maximum value (i.e. at 0.32 m$^3$/s). Temperature $T_{SW,1}$ is determined using Eqn. (3). In System B2, the AC unit is only reliant on DSW, with the SSW pump switched off.
incurred in the DSW 9.5 km-long pipeline. On the other hand, Table 5 indicates that the full-load operation of the AC unit using **System B1** (DSW from the OWTEP turbine mixed with SSW) improves the annual cooling generation marginally but results in annual gross electricity savings of

**Figure 13.** Monthly average values for the sea water temperature at the entry of the condenser.

**Figure 14.** Monthly average values for the sea water temperature at the exit of the condenser.

**Figure 15.** Monthly values for the cooling generated.

**Figure 16.** Monthly values for the net electricity consumption.

**Figure 17.** Hourly coefficient of performance for System B1.

**Figure 18.** Hourly coefficient of performance for System B2.
0.67 GWh. This effectively reduces the deficit of the OWTEP electricity generation with respect to that from the conventional wind turbine from 9.1% down to 6.4%. Furthermore, Table 5 shows that the intermittent operation of the AC unit using System B2 results in a theoretical increase of the COPc of 64% (12.85 vs 7.83). Although the intermittent operation would limit the annual cooling production to 8.25 GWh, the conventional turbine would require an equivalent of 1.32 GWh annually to generate the same amount of cooling from an AC unit relying only on SSW (System A) in the absence of a DSW pipeline. This is significantly higher than the 0.8 GWh value required by the combined OWTEP-AC system with System B2. Consequently, under such intermittent operation, the effective deficit of the OWTEP system’s electricity generation with respect to that from a conventional wind turbine drops from 9.1% to 3.0%.

Table 5. Three System Operational Modes Investigated.

| System Mode | Net Annual Electricity OWTEP (GWh) | AC-Unit Operation | Annual Cooling Production \((q_L, \text{ GWh})\) | Net Annual Electricity Consumed by AC-unit \((W, \text{ GWh})\) | Annual Average COPc | Gross Annual Electricity Consumed by AC-unit \((W/\eta_\ell, \text{ GWh})\) |
|-------------|-----------------------------------|-----------------|------------------|--------------------------|---------------------|-----------------------------------|
| A           | -                                 | Full Load       | 33.53            | 4.28                     | 7.83                | 5.35                              |
| B1          | 11.21                             | Full Load       | 33.83            | 3.98                     | 8.49                | 4.68                              |
| B2          | 11.21                             | Intermittent    | 8.25             | 0.64                     | 12.85               | 0.80                              |

7. Conclusions and Future Work
This study has shown that while electricity generation using the OWTEP concept under the considered Central Mediterranean climatic conditions is less efficient in terms of rotor mechanical power to electrical power conversion than that generated by conventional wind turbine technology. Significant losses associated with hydraulic transmission across the pipeline are offset by the exploitation of DSW which reduces the energy consumption by sea water based air-conditioning units. This is a consequence of the fact that the use of cooler sea water significantly improves the performance of vapour compression systems. The installation of OWTEP turbines at sites closer to the coast and in deep waters having lower deep sea temperatures may likely result in better overall performance than for conventional turbines which only produce electricity. This is subject to further modelling of combined OWTEP-AC systems under different climatic/sea conditions. Further work will also investigate potential benefits of such systems over existing offshore wind technologies through a more detailed analysis involving costs models, more accurate modelling of the refrigeration cycle, the coupling of the OWTEP-AC model to a district cooling network model to simulate more accurately the cooling loads from buildings as well as the integration of hydro-energy storage system models.

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