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DOI
10.1088/1742-6596/753/10/102004

Publication date
2016

Document Version
Final published version

Published in
Journal of Physics: Conference Series

Citation (APA)
Jarquin-Laguna, A. (2016). Simulation of an offshore wind farm using fluid power for centralized electricity generation. Journal of Physics: Conference Series, 753(10), 1-13. [102004]. https://doi.org/10.1088/1742-6596/753/10/102004

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Simulation of an offshore wind farm using fluid power for centralized electricity generation

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Abstract. A centralized approach for electricity generation within a wind farm is explored through the use of fluid power technology. This concept considers a new way of generation, collection and transmission of wind energy inside a wind farm, in which electrical conversion does not occur during any intermediate conversion step before the energy has reached the offshore central platform. A numerical model was developed to capture the relevant physics from the dynamic interaction between different turbines coupled to a common hydraulic network and controller. This paper presents two examples of the time-domain simulation results for an hypothetical hydraulic wind farm subject to turbulent wind conditions. The performance and operational parameters of individual turbines are compared with those of a reference wind farm with conventional technology turbines, using the same wind farm layout and environmental conditions. For the presented case study, results indicate that the individual wind turbines are able to operate within operational limits with the current pressure control concept. Despite the stochastic turbulent wind input and wake effects, the hydraulic wind farm is able to produce electricity with reasonable performance in both below and above rated conditions.

1. Introduction
A typical offshore wind farm consists of an array of individual wind turbines several kilometers from shore. Each of these turbines captures the kinetic energy from the wind and converts it into electrical power in a similar way as is done with onshore technology. However, one main characteristic of a wind farm as a collection of individual turbines, is that electricity is still generated in a distributed manner. This means that the whole process of electricity generation occurs separately and is then collected, conditioned and transmitted to shore. When looking at a wind farm as a power plant, it seems reasonable to consider the use of only a few generators of larger capacity rather than around one hundred of generators of lower capacity. The potential benefits, challenges and limitations of a centralized electricity generation scheme for an offshore wind farm are not known yet.

This work explores a particular concept in which a centralized electricity generation within a wind farm is proposed by means of a hydraulic network using fluid power technology [1]. The basic idea behind the concept is to dedicate the individual wind turbines to create a pressurized flow of seawater. Then, the flow is collected from the turbines and redirected through a network of pipelines to a central generator platform. At the platform, the overall pressurized flow is converted first into mechanical and later into electrical power through an impulse hydraulic turbine.
This paper continues with previous work [2, 3] in an effort to assess the trade-offs implied by the proposed hydraulic concept. To this aim, time domain simulations are used to evaluate the performance and operational parameters of individual turbines coupled to a common hydraulic network for an hypothetical wind farm with centralized electricity generation. In the first part of this work, an overview of the wind farm model is presented together with the control strategy of the hydraulic components; the second part describes a case example where the results are compared with those of a typical wind farm.

2. Wind farm model overview
The overall wind farm model, incorporates the dynamic interaction between the individual turbines, the hydraulic network, the Pelton turbine and the controller. The model is described as a set of coupled algebraic and non-linear ordinary differential equations which are solved by numeric integration using Matlab-Simulink. The hydraulic wind power plant model is composed by the following subsystems:

2.1. Wind turbines
2.1.1. Aerodynamic model
The aerodynamic characteristics of a horizontal axis wind turbine rotor are a function of its rotational speed \( \omega_r \), the pitch angle of the blades \( \beta \) and the relative velocity of the upstream wind speed \( U \) with respect to the rotor. The torque \( \tau_{aero} \), and axial thrust \( F_{thrust} \) performance are described through their non-dimensional steady-state coefficients as a function of the upstream wind speed.

\[
\tau_{aero} = C_t(\lambda, \beta) \frac{1}{2} \rho_{air} \pi R^3 U_{rel}^2 \\
F_{thrust} = C_{Fax}(\lambda, \beta) \frac{1}{2} \rho_{air} \pi R^2 U_{rel}^2
\]

where \( \rho_{air} \) is the air density, \( R \) is rotor radius and the tip speed ratio \( \lambda \) is defined as the tangential velocity of the blade tip and the upstream undisturbed wind speed.

\[
\lambda = \frac{\omega_r R}{U}
\]

This reduced order model does not include any aero-elastic or unsteady aerodynamic effects. Although these aspects are important for the loading of both rotor and support structure, their effects on the aerodynamic torque are considered less relevant from the performance and control point of view of the overall wind farm; the relatively large mass moment of inertia of the rotor in the angular degree of freedom, will absorb large peak fluctuations in the rotor speed derived from the unsteady aerodynamic effects on the rotor torque.

2.1.2. Hydraulic drive train model
The hydraulic drive train consists of a large positive displacement water pump directly coupled to the low-speed rotor shaft. Hence, the rotor-pump angular acceleration is described through the balance of the aerodynamic torque \( \tau_{aero} \), and the transmitted torque from the pump \( \tau_p \) as a first order differential equation; the mass moment of inertia of the rotor and pump is described by:

\[
J_r \ddot{\omega}_r - \tau_{aero}(U, \beta, \omega_r) + \tau_p(\omega_r, \Delta p_p, V_p) = 0
\]
The pump is mainly characterized through a variable volumetric displacement $V_p$, which determines the volume of fluid that is obtained per rotational displacement. Hence the volumetric flow rate of the pump $Q_p$ is ideally given by the product of its volumetric displacement and the rotor shaft speed; internal leakage losses are included as a linear function of the pressure drop across the pump $\Delta p$ with the laminar leakage coefficient $C_s$. In a similar manner, the transmitted torque is directly related to the volumetric displacement and the pressure across the pump; a friction torque is described with a viscous and a dry component defined with the damping coefficient $B_p$ and a friction coefficient $C_f$ respectively [4].

$$Q_p = V_p(e) \omega_r - C_s \Delta p_p$$  \hspace{1cm} (6)

$$\tau_p = V_p(e) \Delta p_p + B_p \omega_r + C_f V_p(e) \Delta p_p$$  \hspace{1cm} (7)

Here $e$ is introduced as the ratio of the current volumetric displacement and its nominal value per rotational cycle such that:

$$V_p = e V_{p,max}$$  \hspace{1cm} (8)

The variable $e$ from equation 8 is used as a control variable to modify either the volumetric flow rate or the transmitted torque of the pump. The dynamics of a general actuator used to modify the volumetric displacement of the pump, are approximated by a first order differential equation. The constant $T_e$ characterizes how slow or fast the actuator responds to a reference value input $e_{dem}$ according to the following equation:

$$\dot{e} = \frac{1}{T_e} (e_{dem} - e)$$  \hspace{1cm} (9)

The yaw degree of freedom of the individual turbines is not considered. Hence, the yaw controller of the turbines is not included. A schematic showing the different subsystems of a single turbine is shown in Figure 1.

2.2. Hydraulic network
The hydraulic network consists of a number of interconnected pipelines represented by linear transmission line models. The approach to construct this networks for time-domain simulations.
from individual pipelines was previously presented in [2]. The dynamic response of the compressible laminar flow of a Newtonian fluid through a rigid pipeline network is given by the following state-space model; the model includes inertia and compressibility effects which are necessary to describe the fluid transients or so-called ‘water-hammer’ effects. The model uses the volumetric flow rates from the individual rotor driven pumps and at the nozzle as an input, and the pressures at across the water pumps and nozzle as an output.

\[
\begin{align*}
\dot{x} &= A_Q x + B_Q \begin{pmatrix} Q_{p,1} \\ Q_{p,2} \\ \vdots \\ Q_{p,i} \\ Q_{nz} \end{pmatrix}, \\
&= C_Q x \quad (10)
\end{align*}
\]

The matrices \(A_Q, B_Q\) and \(C_Q\) are defined in terms of the physical parameters of the hydraulic lines and water properties such as water viscosity, water density, speed of sound in the water, length and internal radius of the pipelines.

2.3. Nozzle and spear valve

At the end of the hydraulic network, a nozzle and spear valve is used to adapt the pressurized water flow into the Pelton turbine. The nozzle characteristics are included as a first order differential equation by taking the momentum equation of a fluid particle into account along the nozzle length \(L_{nz}\) as described in equation 11, [5].

\[
\rho_{hyd} L_{nz} \frac{d}{dh_s} Q_{nz} = \Delta p_{nz} A_{nz}(h_s) - \frac{\rho_{hyd} Q_{nz} |Q_{nz}|}{2 A_{nz}(h_s) C_d^2} \quad (11)
\]

Where \(\rho_{hyd}\) is the density of the hydraulic fluid, \(A_{nz}\) is the nozzle cross sectional area determined by the position of the spear valve, and \(C_d\) is the discharge coefficient to account for pressure losses due to the geometry and flow regime at the nozzle exit. The nozzle cross sectional area is described by the linear position of the spear valve \(h_s\) according to equation 12. It is assumed that the spear valve position is smaller than the fixed nozzle diameter \(d_s\). The geometric characteristics of the spear valve are included through the spear cone angle \(\alpha\) as presented in [6].

\[
A_{nz}(h_s) = \min \left( \pi \left[ h_s d_s \sin \left( \frac{\alpha}{2} \right) - h_s^2 \sin^2 \left( \frac{\alpha}{2} \right) \cos \left( \frac{\alpha}{2} \right) \right], \frac{\pi}{4} d_s^2 \right) \quad (12)
\]

Similarly to the pump actuator, the spear valve linear actuator is described through a first order differential equation. The parameter \(T_h\) accounts for the time constant of the actuator:

\[
\dot{h_s} = \frac{1}{T_h} (h_{s,\text{dem}} - h_s) \quad (13)
\]

The hydraulic power at the nozzle \(P_{hyd}\) is given by the product of the volumetric flow rate and the water pressure at this location.

\[
P_{hyd} = Q_{nz} \Delta p_{nz} \quad (14)
\]
2.4. Pelton turbine

The hydraulic efficiency of the Pelton runner \( \eta_P \) is obtained from momentum theory according to different geometrical and operational parameters as described in [7] and [8].

\[
\eta_P = 2k (1 - k) (1 - \xi \cos \gamma)
\]

(15)

where \( \xi \) is an efficiency factor to account for the friction of the flow in the bucket, \( \gamma \) is defined as the angle between the circumferential and relative velocities, and \( k \) is the runner speed ratio defined by the ratio between the tangential velocity of the runner at Pitch Circle Diameter (PCD) and the jet speed.

\[
k = \frac{\omega P R_{PCD}}{U_{jet}}
\]

(16)

For this case, the Pelton rotational speed is fixed to the synchronous speed of the generator. Thus, the efficiency of the Pelton turbine is only determined by the water jet velocity \( U_{jet} \), which is simply the volumetric flow rate divided by the cross sectional area and multiplied by a vena contracta coefficient \( C_v \) to account for the change in velocity immediately after the water jet exits the nozzle.

\[
U_{jet} = C_v \frac{Q_{nc}}{A_{nz} (h_s)}
\]

(17)

2.5. Environmental conditions

The dynamic wind flow models and wake effects for a given layout are based on an open source toolbox developed for ‘Distributed Control of Large-Scale Offshore Wind Farms’ as part of the European FP7 project with the acronym Aeolus [9]. The model assumes a 2D wind field generated at the hub height plane. The wind field does not account for wind shear or tower shadow effects and is generated at hub height plane. The mean wind speed has a constant value in the longitudinal direction and zero lateral component. Similarly, the wind speed direction is fixed with respect to the farm layout in longitudinal direction. The turbulent wind field is generated using a Kaimal spectrum; two spectral matrices together with coherence parameters are used to describe the spatial variations of the wind speed according to [10].

Three wake effects are considered: deficit, expansion and center, where wake deficit is a measure of the decrease in downwind wind speed, wake expansion describes the size of the downwind area
affected by the wake and wake center defines the lateral position (meandering) of the wake area. Expressions for wake deficit, center and expansion were developed in [11, 12]. To illustrate this, a small wind farm comprising of five turbines is shown in the layout of Figure 2. Figure 3 shows a snapshot of the wind field where the wake effects are observed.

3. Variable speed control strategy
The so called variable-speed operation is of particular interest for this concept because by removing the individual generators and power electronics from the turbines, the hydraulic drives need to replace the control actions to obtain the variable-speed functionality.

3.1. Pump controller
As shown in equation 7, it is possible to manipulate the transmitted torque of the pump using two different control degrees of freedom (in contrast with the electro-magnetic torque in a conventional turbine): the volumetric displacement of the pump and/or the pressure across it. In this case, the volumetric displacement of the pump from each turbine is controlled under a relatively constant pressure supply. Hence, the rotational speed of each rotor is able to be modified independently according to the local wind speed conditions. This strategy is commonly known in hydraulic systems as ‘secondary control’ [13]. The required volumetric displacement of the pump $e_{dem}$ is shown in equation 18 as a function of the measured rotational speed of the rotor $\omega_{r, meas}$ and the measured pressure at the pump location $\Delta p_{p, meas}$. A low pass filter on the pressure measurement is employed to prevent actuation from the fluid transient fluctuations in the hydraulic network. The reference torque $\tau_{ref}$ is obtained from the steady-state torque-speed curves defined for different operating regions as in conventional variable-speed control strategies.

$$e_{dem} = \frac{\tau_{ref} (\omega_{r, meas}) - B_p \omega_{r, meas}}{V_p \left( 1 + C_f \right) \Delta p_{p, meas}}$$ (18)

3.2. Spear valve controller
In order to achieve a constant pressure in the hydraulic network, the linear actuation of the spear valve is used to constrict or release the flow rate through the nozzle area. The pressure control is based on a PI feedback controller and a cascade controller compensation to modify the linear position of the spear valve. A similar pressure control loop has been proposed in [6]; a schematic of the proposed controller is shown in Figure 4. In addition to the PI controller, a series of notch filters are added to prevent excitation from the low damped modes of the hydraulic network.

![Figure 4. Pressure control schematic based on the spear valve position of the nozzle.](image)
4. Simulation example

4.1. Wind farm conditions

The model described in the previous sections is used to assess the performance and operating conditions of a small hydraulic wind farm under specific wind conditions. Five turbines of 5MW each are interconnected, through a hydraulic network, to a 25MW Pelton turbine located at an offshore platform within 1 km distance from the individual turbines. Two different wind speeds corresponding to below and above rated conditions are simulated. First, a wind field with a mean wind speed of 9 m/s and 10% turbulence intensity (TI) is used as an input during 1000s. For above rated conditions, a mean wind speed of 15 m/s and 12% TI is employed. The main parameters are shown in Table 1.

Table 1. Main design parameters for the offshore wind turbine with fluid power transmission.

| Design parameter                  | Design parameter                  |
|-----------------------------------|-----------------------------------|
| Rotor diameter 126 m              | Drivetrain concept Hydraulic      |
| Rated wind speed 11.4 m/s         | Nominal water pressure 150 bar    |
| Design tip speed ratio λ 7.55     | Pump volumetric disp 10.2 L/rpm   |
| Max power coefficient $C_P$ 0.485 | Lines length 1 km                 |
| Rated power 5 MW                  | Lines diameter 0.5 m              |
| Max blade tip speed 80 m/s        | Nozzle nom diameter 43.2 mm       |

The results from the simulations are compared with those of a reference wind farm comprising of 5MW NREL turbines [14], using the same wind farm layout and environmental conditions. A schematic of the individual turbines and configurations used in the simulation example for both wind farms is shown in Figure 5. The capital letters A, B and C are used as a reference to present the results at specific points.

(a) NREL reference turbine.

(b) Hydraulic turbine.

Figure 5. Simplified schematic with the main components involving the energy conversion for a reference offshore wind turbine and the proposed hydraulic concept.
4.2. Time-domain results

The results from the time domain simulations are presented for the main operational parameters of the five individual turbines such as mechanical power, rotor speed and pitch angle. For below rated conditions, Figures 6(a) and 6(b) show the transient response of the reference and the hydraulic wind farm respectively. The results show that for this case scenario and with the current control strategy, the hydraulic wind farm is able to generate electricity from the pressurized water flow to the central platform via a Pelton turbine. In terms of performance it is observed that the turbines from the hydraulic wind farm experience higher excursions of the rotor speed in comparison with the reference case; this effect is also reflected in the increased pitch action required for the same wind speed conditions. A possible explanation of the more pronounced changes of the rotor speed, is that the resulting torque demand generated by the hydraulic system, is slower than the reference case due to the higher fluid inertia of the hydraulic network. The slower response of the hydraulic wind farm also is observed during the first minute of electrical power production. In this time span, high frequency fluctuations are still present due to pressure waves in the hydraulic network as a result of transient operation; the fluctuations are eventually reduced. From a reliability point of view, the increased pitch action might have an important consequence on the life time of the pitch system.

For above rated conditions, the simulation results are shown in Figures 7(a) and 7(b). It is observed that both concepts are able to keep the rotor speed operating within a constant speed band while producing relatively constant power. Likewise, the pitch actuation is very similar in both wind farms, which is not unexpected since the same pitch controller is used. Once more, the transient operation in the electrical power production is more pronounced in the case of the hydraulic wind because of the high hydraulic inertia of the system. High frequency oscillations are observed in the electrical power due to the pressure waves travelling along the network.

| Wind farm concept                  | Averaged power [MW] | Efficiency [-] | Power coeff | A to B | B to C |
|-----------------------------------|---------------------|----------------|-------------|--------|--------|
|                                   | Mechanical point A  | Transmitted point B | Electrical point C | \(C_P\) | \(\eta_{AB}\) | \(\eta_{BC}\) |
| NREL reference                   | mean std            | mean std        | mean std     | mean   | mean   | mean   |
| WT1                               | 3.12 0.86           | 2.95 0.81       | 2.61 0.72    | 0.483  | 0.944  | 0.885  |
| WT2                               | 2.23 0.60           | 2.11 0.57       | 1.87 0.50    | 0.483  | 0.944  | 0.885  |
| WT3                               | 2.90 0.88           | 2.74 0.83       | 2.42 0.73    | 0.483  | 0.944  | 0.885  |
| WT4                               | 2.99 0.83           | 2.82 0.78       | 2.50 0.69    | 0.483  | 0.944  | 0.885  |
| WT5                               | 2.10 0.58           | 1.98 0.54       | 1.75 0.48    | 0.483  | 0.944  | 0.885  |
| Total                             | 13.3                | 12.6            | 11.1 1.90    | -      | -      | -      |
| Hydraulic with pressure control  |                     |                |             |        |        |        |
| WT1                               | 3.06 0.92           | -               | -            | 0.479  | -      | -      |
| WT2                               | 2.22 0.69           | -               | -            | 0.482  | -      | -      |
| WT3                               | 2.84 0.90           | -               | -            | 0.479  | -      | -      |
| WT4                               | 2.94 0.89           | -               | -            | 0.480  | -      | -      |
| WT5                               | 2.08 0.65           | -               | -            | 0.482  | -      | -      |
| Total                             | 13.1                | 11.6            | 10.2 2.71    | -      | 0.88   | 0.877  |
**Figure 6.** Time domain results for a wind farm comprising of 5 turbines subject to a wind field with a mean speed of 9 m/s and 10% turbulence intensity.
Figure 7. Time domain results for a wind farm comprising of 5 turbines subject to a wind field with a mean speed of 15 m/s and 12% turbulence intensity.
4.3. Performance comparison
The performance of the wind farm for the simulated conditions is summarized in the bar plots of Figures 8 and 9 for below rated conditions, and in Figures 10 and 11 for above rated conditions, where the averaged values with the standard deviation of the power transmission and conversion are displayed. The numerical values together with the averaged efficiencies are shown in Tables 2 and 3.

![Figure 8](image1.png)
**Figure 8.** Power performance for the reference wind farm, below rated conditions.

![Figure 9](image2.png)
**Figure 9.** Power performance for the hydraulic wind farm, below rated condition.

![Figure 10](image3.png)
**Figure 10.** Power performance for the reference wind farm, above rated conditions.

![Figure 11](image4.png)
**Figure 11.** Power performance for the hydraulic wind farm, above rated conditions.

After including the different performance of the main subsystems involved in the conversion and transmission of wind energy in a wind farm, the results show that the overall efficiency of a hydraulic wind farm is lower for an hydraulic concept compared to conventional technology; for the presented operating conditions the hydraulic wind farm overall efficiency was between 0.772 – 0.810 compared to 0.835 (excluding the aerodynamic performance). The most important losses from the hydraulic concept are attributed to the variable displacement pumps and friction losses in the hydraulic network. Despite having a slower response due to high water inertia, the hydraulic concept also showed higher standard deviations in the generated electrical power due to pressure transients in the hydraulic network.
Table 3. Performance overview of time domain results for above rated conditions.

| Wind farm concept | Averaged power [MW] | Efficiency [-] |
|-------------------|---------------------|---------------|
|                   | Mechanical point A  | Transmitted point B | Electrical point C | Power coeff | A to B η_{AB} | B to C η_{BC} |
| NREL reference    | mean std            | mean std        | mean std          | mean | mean | mean |
| WT1               | 5.28 0.22           | 4.99 0.21       | 4.41 0.18         | 0.249 | 0.944 | 0.885 |
| WT2               | 5.27 0.23           | 4.97 0.22       | 4.40 0.19         | 0.284 | 0.944 | 0.885 |
| WT3               | 5.28 0.22           | 4.99 0.21       | 4.42 0.18         | 0.251 | 0.944 | 0.885 |
| WT4               | 5.28 0.23           | 4.98 0.22       | 4.41 0.19         | 0.244 | 0.944 | 0.885 |
| WT5               | 5.27 0.23           | 4.98 0.22       | 4.41 0.19         | 0.277 | 0.944 | 0.885 |
| Total             | 26.4                | 24.9            | 22.1 0.92         | -    | -    | -    |

Hydraulic with pressure control

|                   | Averaged power [MW] | Efficiency [-] |
|-------------------|---------------------|---------------|
|                   | Mechanical point A  | Transmitted point B | Electrical point C | Power coeff | A to B η_{AB} | B to C η_{BC} |
| WT1               | 5.24 0.18           | -               | -               | 0.247 | -    | -    |
| WT2               | 5.22 0.19           | -               | -               | 0.282 | -    | -    |
| WT3               | 5.25 0.18           | -               | -               | 0.250 | -    | -    |
| WT4               | 5.25 0.18           | -               | -               | 0.243 | -    | -    |
| WT5               | 5.23 0.19           | -               | -               | 0.274 | -    | -    |
| Total             | 26.2                | 24.4 1.40       | 21.4 1.44        | -    | 0.931 | 0.87 |

5. Conclusion & Outlook

The numerical model of an hydraulic wind power plant, which is used to generate electricity in a centralized manner, was presented. The model allows to capture the most relevant physics of a wind farm including transient behaviour from the hydraulic network and Pelton turbine. Despite the stochastic turbulent wind input and wake effects, the hydraulic wind farm is able to produce electricity with reasonable performance. A constant pressure controller was used in the nozzle spear valve, to avoid the excitation of flow and pressure dynamics in the hydraulic network.

The performance of the hydraulic wind farm was compared with a reference wind farm using conventional technology. For the presented case study, results indicate that the individual wind turbines are able to operate within operational limits with the current control concept. Compared to the reference wind farm, the hydraulic collection and transmission has a lower efficiency due to the losses induced by the variable displacement water pumps and friction losses in the hydraulic network. Further work includes the evaluation of alternative control strategies and different load cases, such as extreme wind gust, start-up and shutdown conditions, to assist the performance evaluation of the proposed centralized electricity generation approach.

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