Fluid power network for centralized electricity generation in offshore wind farms

A Jarquin-Laguna
Offshore Engineering, Delft University of Technology, Stevinweg 1, 2628 CN, Delft, The Netherlands
E-mail: a.jarquinlaguna@tudelft.nl

Abstract.
An innovative and completely different wind-energy conversion system is studied where a centralized electricity generation within a wind farm is proposed by means of a hydraulic network. This paper presents the dynamic interaction of two turbines when they are coupled to the same hydraulic network. Due to the stochastic nature of the wind and wake interaction effects between turbines, the operating parameters (i.e. pitch angle, rotor speed) of each turbine are different. Time domain simulations, including the main turbine dynamics and laminar transient flow in pipelines, are used to evaluate the efficiency and rotor speed stability of the hydraulic system. It is shown that a passive control of the rotor speed, as proposed in previous work for a single hydraulic turbine, has strong limitations in terms of performance for more than one turbine coupled to the same hydraulic network. It is concluded that in order to connect several turbines, a passive control strategy of the rotor speed is not sufficient and a hydraulic network with constant pressure is suggested. However, a constant pressure network requires the addition of active control at the hydraulic motors and spear valves, increasing the complexity of the initial concept. Further work needs to be done to incorporate an active control strategy and evaluate the feasibility of the constant pressure hydraulic network.

1. An offshore wind turbine for centralized electricity generation
Electricity production from offshore wind is currently done using multimegawatt (multi-MW) wind turbines organized in a wind farm. Inside the farm, different network topologies are used for the collection and transmission of the electrical power into an offshore transformer platform. Here, further conditioning of the overall produced electrical power takes place before it is transmitted to the grid onshore [1].

An innovative and completely different wind-energy conversion system is studied in this paper. By means of a hydraulic network, a centralized electricity generation within a wind farm is proposed [2]. The gearboxes, generators and power electronics from individual turbines are removed and replaced with a fluid power transmission coupled to a seawater pump. Hence, the rotor-nacelle assemblies are only dedicated to pump seawater into the hydraulic network and mechanical power is converted to hydraulic power without any electrical intermediate conversion. Electricity is generated at a central offshore platform through a Pelton turbine and is transmitted to shore in a similar way as conventional wind farms. A schematic of the concept is shown in figure 1.
Previous work has shown the performance and transient behavior of a turbine with a single water pipeline by means of time domain simulations [3]. Results showed that a variable speed operation of the rotor is possible through a passive control method using a nozzle at the end of the water pipeline. However, it is uncertain if the same passive control method will be sufficient for cases in which multiple turbines are connected to the same hydraulic network. This paper presents some intermediate results for this ongoing research by analysing two turbines within a wind farm interconnected in different hydraulic network configurations. Although it is possible to model and connect several turbines using the proposed approach, only two turbines are analysed to keep a simple analysis and have a better understanding of their dynamic interaction. Time domain simulations of the reduced wind farm are used to evaluate the performance and rotor speed stability of both turbines under typical wind conditions.

The wind farm simulation model from [4] was used to define the ambient wind field and wake effects. The dynamic model of the individual hydraulic turbines including aerodynamic effects, hydraulic transmission and control strategy is presented in [3]. In addition, the model is modified in order to exclude the single water line and nozzle. Instead, the hydraulic network model is used as described in the following section. The numerical model and control strategy of the Pelton turbine is not included in this study as this component is physically decoupled from the rest of the system and does not have an influence on the stability of the individual turbines. The performance of the Pelton turbine is therefore neglected in the presented results. The main parameters of a single turbine are shown in table 1.

2. The fluid power network
One of the key aspects for having a centralized electricity generation is the use of hydraulic networks to collect and transport the pressurised seawater from the individual turbines to the generator platform. Two configurations are presented, the first one assumes that every turbine has an independent water pipeline and a nozzle, see figure 2. The second one considers that the water pipelines from individual turbines are connected through a common line to a single nozzle, see figure 3.
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 10.6 m/s         | Nominal oil pressure 350 bar      |
| Design tip speed ratio $\lambda$ 7.55 | Nominal water pressure 150 bar     |
| Max power coefficient $C_p$ 0.485 | Nominal motor speed 1500 rpm      |
| Rated mech power 4.4 MW           | Nominal water flow rate 15242 lpm |
| Max blade tip speed 80 m/s        | Nozzle diameter 43.2 mm           |

Figure 2. Network with independent pipelines and nozzles.

Figure 3. Network with parallel pipelines with common line & nozzle.

The dynamic response of compressible laminar flow of a Newtonian fluid through a rigid pipeline has been described through what is referred as the dissipative pipeline model. The model assumes axisymmetrical flow with negligible motion in the radial direction and also neglects non linear convective terms; thermodynamic effects are also neglected [5]. The underlying idea of different approximation methods is to describe the solution to this model through a set of linear ordinary differential equations which are intended for use in time-domain simulations. In this way the pipeline flow dynamics can be incorporated and coupled to other algebraic or differential equations describing other subsystems of the wind turbines, i.e. aerodynamics, structural, control etc.

This paper uses a modal approximation for different input-output causalities (or boundary conditions) based on the work of [6]. Other models for transient laminar flow can be used in this approach as long as they are represented with linear ordinary differential equations. In the case of transient turbulent flow, different models in the form of non linear differential equations have been proposed and might also be used with the current approach, however their implementation and validation into more complex hydraulic networks is outside of the scope of this paper. Two different models are considered in this paper depending on the boundary conditions at upstream and downstream sides of the line, the Q-model and the Qp-model. The Q-model uses the flow rates as given inputs and the pressures as outputs. The Qp-model uses the upstream flow rate and downstream pressure as inputs, and the upstream pressure and downstream flow rate as outputs. The state-space representation of both models is shown in equations 1 and 2. Using this description for individual line elements, it is possible to assemble the model for a whole hydraulic network.
The definition of matrices $A$, $B$, $C$ and $D$ for both models is done in terms of the properties of the hydraulic fluid and physical parameters of the line; their definition is shown in [7].

2.1. Independent hydraulic lines with nozzle
In this configuration, each water pump of the turbines is connected with a single pipeline and nozzle. Each pipeline is modelled with a Q-model, where the input upstream flow rate $Q_{p,i}$ is given through the rotational speed $\omega_{p,i}$ of the water pump and the downstream flow rate $Q_{i}$ is given by the nozzle flow rate, see figure 4.

In order to connect the downstream side of each pipeline to a nozzle, the following considerations are used:

- The flow rate at the downstream side of each hydraulic line is equal to the flow rate through its nozzle, $Q_i = Q_{nz}$
- The pressure difference through the nozzle determines the pressure at the downstream side of the hydraulic line, $\Delta p_{nz} = p_{i, out} - p_{atm}$

A first order differential equation is used to describe the volumetric flow rate through the nozzle $Q_{nz}$ by taking the momentum equation of a fluid particle into account. Here $L_{nz}$ is the nozzle length, $A_{nz}$ is the nozzle cross sectional area and $C_d$ is the discharge coefficient to account for pressure losses due to the geometry and flow regime at the nozzle exit.

$$\rho_w Q_{nz} L_{nz} = \Delta p_{nz} A_{nz} - \rho_w Q_{nc} |Q_{nc}| \frac{2 A_{nz} C_d}{C_d}$$  \hspace{1cm} (3)

The flow rate $Q_{p,i}$ and transmitted torque $\tau_{p,i}$ of the water pumps, is given through the following relations,

$$Q_{p,i} = V_{p,i} \omega_{p,i} - C_{v,i} p_i$$ \hspace{1cm} (4)
$$\tau_{p,i} = V_{p,i} p_i + B_{p,i} \omega_{p,i} + C_{f,i} V_{p,i} p_i$$ \hspace{1cm} (5)
where $V_p$ is the pump volumetric displacement which describes the amount of volumetric fluid obtained per rotational displacement of the driving shaft; $C_v$ is the laminar leakage coefficient which is used to describe internal leakage losses as a function of the pressure difference across the pump. For the transmitted torque $\tau_p$, besides the ideal torque $V_p \omega$, a friction torque is added and described with a viscous component defined with the coefficient $B_p$. A torque loss caused by the mechanical internal friction is also present and described through the loss coefficient $C_f$ respectively [8].

The assembled model for two water pumps with independent pipelines and nozzles is shown in equation 6. The superscripts in vectors and matrices represents the element number and the subscripts indicate column vectors (i.e. $b_1^{(2)}$ is the first column of matrix $B$ of the second pipeline element).

$$
\begin{align*}
\begin{pmatrix}
\dot{x}^{(1)} \\
\dot{\theta}_d^{(1)} \\
\dot{x}^{(2)} \\
\dot{\theta}_d^{(2)}
\end{pmatrix} &= 
\begin{pmatrix}
A^{(1)}& C_1 & b_1^{(1)} & 0 \\
0 & 0 & 0 & 0 \\
b_2^{(1)} & 0 & A^{(2)}& C_2 & b_2^{(2)} \\
0 & 0 & 0 & 0 & \frac{\Delta p_{atm}}{\Delta p_{2}}
\end{pmatrix}
\begin{pmatrix}
\dot{x}^{(1)} \\
\dot{\theta}_d^{(1)} \\
\dot{x}^{(2)} \\
\dot{\theta}_d^{(2)}
\end{pmatrix} + 
\begin{pmatrix}
V_{in,1} |(1) \\
0 \\
V_{in,2} |(2) \\
0
\end{pmatrix} + 
\begin{pmatrix}
\omega_{p,1} \\
0 \\
\omega_{p,2}
\end{pmatrix}
\end{align*}
$$

2.2. Parallel hydraulic lines with common line and nozzle

Consider the case where lines from individual turbines are in parallel with a common line at the end. Each of the parallel lines is modelled with a Qp-model, where the input flow rate $Q_p$ is given through the rotational speed $\omega_p$ of the water pump and the pressure $p_{line}$ from the output of the common line. For the common line, a Q-model is used where the upstream input flow rate $Q_{line, in}$ is the sum of the individual downstream flow rates $Q_i$, which are the output of each Qp-model. At this node, the upstream pressure of the common line is the input for the parallel lines. An schematic of the model showing the input-output causality for each element is shown in figure 5.

Similarly to the previous configuration, the following considerations are used to connect the downstream side of the common pipeline to the nozzle:

- The flow rate at the downstream side of the common line is equal to the flow rate through the nozzle, $Q_{line} = Q_{nz}$
- The pressure difference through the nozzle determines the pressure at the downstream side of the common line, $\Delta p_{nz} = p_{line, out} - p_{atm}$

The assembled model for two parallel pipelines interconnected with a common line and a nozzle is shown in equation 7.
The presented models of hydraulic networks are assembled with the rest of the algebraic and differential equations which describe the main dynamic submodels of individual turbines as presented in [4] using the hydraulic transmission presented in [3] instead of the mechanical gearbox. The rotational speed of the motor-water pump assembly (high speed shaft) is used as an input to the hydraulic networks and the transmitted torque of the water pumps is used as feedback to the turbine models. In this manner a complete model with the individual turbines coupled to the hydraulic network is obtained. The complete wind farm model is now described through a set of coupled algebraic and differential equations which are solved numerically using a variable time step ODE solver via MATLAB Simulink. Simple test cases of the overall model (i.e. using a constant wind speed input) were used to check steady state results. Time domain simulations are performed to obtain the transient response to a stochastic wind input as described in the next section.

3. Results of time domain simulations
In order to evaluate the performance and functionality of the passive control concept, two turbines are placed inline with the mean wind speed direction. A wind field with a mean wind speed of 9m/s and 10% turbulence intensity (TI) is used as an input. Simulations of the reduced wind farm behavior are performed during 1000s with zero initial conditions. Wake effects for the downwind turbine are considered by using the thrust coefficient $C_T$ of the upwind turbine according to [4].

![Figure 6. Wind field grid for two inline turbines; Mean wind speed of 9m/s with 10% TI.](image)

Three different hydraulic networks are evaluated for the same wind field using the same turbines. Model 1 considers independent pipelines and nozzles for both turbines; model 2 connects the parallel lines to a common line and nozzle; model 3 uses the same configurations as the previous model but with an increased nozzle diameter (therefore modifying the passive control strategy to a higher tip speed ratio and lower pressure). An overview of the physical parameters used is shown in table 2.

In the upper graph of figure 7, it is observed a deficit in the effective wind speed at the second turbine (WT 2) due to the wake effects from the first turbine (WT 1). The reduced wind speed is seen by the second turbine after 77.7s approx, which is the time required for the wake from the first turbine to travel with the mean wind speed velocity. With this hydraulic network configuration, the dynamics of each turbine are independent from each other. The passive control strategy works adequately for separate turbines and their behavior only depends on the local effective wind speed. For short periods where the rotor speed exceeds the maximum value, collective pitch action $\beta$ is observed to maintain...
Table 2. Main physical properties of the hydraulic network.

| Network Design parameter | Pipeline WT1 | Pipeline WT2 | Common Main aspects for pipeline centralized generation |
|--------------------------|--------------|--------------|--------------------------------------------------------|
| Model 1 Single Line     | 1000         | 1000         | -                                                      |
| Line length [m]          |              |              | × independent lines - no network                       |
| Line diameter [m]        | 0.50         | 0.50         | ✓ nominal tip speed ratio                               |
| Nozzle diameter [mm]     | 43.2         | 43.2         | ✓ dynamic stability                                     |
| Model 2 Parallel Line    | 500          | 500          | 500                                                    |
| Line length [m]          |              |              | ✓ allows hydraulic network                             |
| Line diameter [m]        | 0.50         | 0.50         | ✓ nominal tip speed ratio                               |
| Nozzle diameter [mm]     | -            | -            | 6.11                                                   |
|                          |              |              | ? dynamic stability unknown                             |
| Model 3 Parallel Line    | 500          | 500          | 500                                                    |
| Line length [m]          |              |              | ✓ allows hydraulic network                             |
| Line diameter [m]        | 0.50         | 0.50         | 6.83                                                   |
| Nozzle diameter [mm]     | -            | -            | 6.83                                                   |
|                          |              |              | ? dynamic stability unknown                             |

Figure 7. Model 1 wind farm simulation results, independent hydraulic network.
the turbine within operation limits as shown in the second and third graphs. A smooth power output from each turbine is obtained in the last graph of figure 7. The independent behavior of turbines also occurs in conventional wind farms which include individual generators per turbine. In Model 2, the turbines are connected in parallel with a common line and the dynamics of individual turbines are no longer independent. The pressure at the upstream side of the common line results in a similar torque feedback for both turbines. Since both turbines experience different effective wind speeds at a specific moment in time, the passive control strategy is strongly limited.

![Inline turbines with parallel pipelines and common nozzle](image)

**Figure 8.** Model 2 wind farm simulation results, parallel pipelines.

As shown in the results of figure 8, the passive control strategy is able to keep a relative stable operation of both turbines only during the first 400s. Afterwards, the torque unbalance in one of the turbines is high enough to increase the rotor speed of the first turbine while breaking the second turbine until it stops. During deceleration of the second turbine, its water pump also reduces the flow rate and pressure in the common line is reduced. A lower pressure creates a higher torque unbalance in the first turbine to accelerate it even more and eventually reaches nominal rotor speed where pitch action is required. During the interaction, the transient flow rates and pressures result in an undesirable oscillatory power output. These results show the importance of having a hydraulic network which is coupled to the dynamics of single turbines resulting in unstable operation using the current passive control strategy.
Figure 9. Model 3 wind farm simulation results, parallel pipelines with augmented nozzle.

In Model 3, an attempt to increase the stability of the passive control strategy for a network with parallel lines with single nozzle is tried by increasing the area of the nozzle. This effectively means that a higher tip speed ratio and lower pressure is imposed with the passive control strategy. Not only a relatively lower aerodynamic performance is sacrificed by operating at a high speed ratio, but also the rated wind speed, and thus rated mechanical power, is decreased due to the constraint on maximum tip speed allowed. On the other hand, by operating at a higher tip speed ratio, higher stability of the system is achieved by allowing larger rotor speed excursions during transient behavior. The results from figure 9 immediately show that stable operation is feasible. A higher rotor speed operation is observed, as expected for a higher tip speed ratio, and increased pitch action is present compared to the previous models. By operating with a reduced aerodynamic performance, the deficit in wind speed at the second turbine has a lower effect than in the first model with independent lines. A smooth power output is also observed as with Model 1 but with transient oscillations during start-up the first 200 s; for this pipeline configuration performance is compromised by rotor speed stability. The performance of the different models is summarized in table 3 where the averaged efficiencies of the rotor $C_P$, the hydrostatic transmission $\eta_{HST}$ and the hydraulic network $\eta_{hyd}$ are shown.
Table 3. Performance overview of time domain results.

|                     | Model 1 | Model 2 | Model 3 |
|---------------------|---------|---------|---------|
|                     | WT 1    | WT 2    | WT 1    | WT 2    |
| Averaged efficiencies|         |         |         |         |
| Power coefficient $C_P$ [-] | 0.459  | 0.465  | 0.337  | 0.178  |
| Eff. from the low to the high speed shaft $\eta_{HS[T]}$ [-] | 0.888  | 0.888  | 0.871  | 0.882  |
| Eff. from the high speed shaft to the nozzle exit $\eta_{pad}$ [-] | 0.913  | 0.915  | 0.920  | 0.920  |

4. Conclusion & Outlook

The numerical model of a reduced offshore wind farm with centralized electricity generation is presented. Centralized generation is possible by using water pipelines in fluid power network for the collection and transmission of pressurized water from the individual turbines. The mathematical model of a fluid power network is obtained through the assembly of individual pipeline elements. Hence, the coupling with different subsystems (in this case two turbines and nozzle) is achieved resulting in a system of ordinary differential equations representing the complete farm reduced model. Results from time domain simulations, with a stochastic wind field input, were used to evaluate the performance and functionality of two different hydraulic networks using a passive control method for the rotor speed.

It is shown that a configuration with parallel pipelines interconnected through a common line and nozzle, has strong limitations in terms of performance and stability for more than one turbine coupled to the same hydraulic network. Stability of the rotor speed for both turbines is possible by operating at a higher tip speed ratio, but the performance of the wind farm is decreased. For individual turbines with independent lines and nozzles this is not the case, however from the design principles for centralized electricity generation, a parallel configuration is preferred. Hence, a hydraulic network with constant pressure is suggested. On the other hand, a constant pressure network requires the addition of active control at the hydraulic motors and spear valves, increasing the complexity of the initial concept. Further work needs to be done to incorporate an active control strategy and evaluate the feasibility of the constant pressure hydraulic network.

References

[1] Liserre M, Cardenas R, Molinas M and Rodriguez J 2011 Overview of multi-MW wind turbines and wind parks IEEE Transactions on Industrial electronics 58 1081-95
[2] Diepeveen N 2013 On Fluid Power Transmission for Offshore Wind Turbines PhD Thesis (Delft: Technical University of Delft)
[3] Jarquin-Laguna A 2013 Modeling and analysis of an offshore wind turbine with fluid power transmission for centralized electricity generation Journal of Computational and Nonlinear dynamics under review
[4] Soltani M, Knudsen T, Grunnet JD and Bak T 2010 AEOLUS toolbox for dynamic wind farm model simulation and control European Wind Energy Conference
[5] D’Souza A and Oldenburger R 1964 Dynamic Response of Fluid Lines Journal of Basic Engineering, Transactions of the ASME 86 589-598
[6] Makinen J, Piche R and Ellman A 2000 Fluid transmission line modeling using a variational method Journal of Dynamic Systems, Measurement and Control, Transactions of the ASME 122 153-62
[7] Makinen J and Marjamaki H 2008 Modelling hydraulic systems as finite elements Proceedings of the 17th IASTED International Conference on Applied Simulation and Modelling Corfu
[8] Merrit H 1967 Hydraulic Control Systems (New York: Wiley) chapter 4 pp 65-72