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ANALYSIS OF A STAND-ALONE HYDRAULIC OFFSHORE WIND TUBRINE COUPLED TO A PUMPED WATER STORAGE FACILITY

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ABSTRACT: The concept of wind turbines operating with hydraulic transmissions is gaining popularity. With the promise of improved energy density and lower maintenance costs, a number of manufacturers are evaluating such a novel concept. In the current study an offshore-specific hydraulic wind turbine design is proposed. This is based on the use seawater as the working fluid in a large scale, open-loop transmission system. The possibility of coupling such a system to an onshore pressurized water storage facility is also evaluated. Simulation results indicate the potential for such an energy storage system, to address a supply-demand mismatch without requiring intermediate conversion to electricity, as in current pumped storage facilities used in conjunction with wind turbines.

Keywords: Wind Energy, Hydraulic Transmission, Energy Storage

1 INTRODUCTION

Existing offshore wind turbines on the market are inherently based on conventional onshore turbine technology that has been adapted to the offshore environment [1]. In the early stages of development the priority was to prove the concept of a turbine operating in the open sea, with all the advantages that such an environment brings with it. Progress has always been of an incremental nature, building on the same ideas [1]. The National Renewable Energy Laboratory (NREL) in the United States lists the development of offshore specific technologies as a priority to maximise the value of offshore wind generation [2].

The main reason for introducing new design concepts is driven by the number of problems that are being encountered in current offshore turbines. One of the main issues is gearbox failure, with current designs requiring a replacement every 4 years [3]. With the gearbox contributing to around 10% of the turbine cost [3], such frequent replacements are very detrimental to the overall viability of offshore wind energy conversion.

Another design driver is to simplify the turbine technology, in attempt to reduce problematic electrical systems that are typically situated in the nacelle. The American Wind Energy Association (AWEA) lists electrical system failure as having the highest repair and maintenance requirement, with 27% of turbine repairs being due to electrical system failure [3].

Large-scale electrical generators and electrical transmission systems require a substantial amount of copper. Historically, copper prices have been very unstable [4], with substantial fluctuations having occurred in the past decade [5]. A reduction in dependency on copper would allow for an improved ability to forecast wind farm viability and encourage more investment.

One of the most prominent solutions to the above problems is a shift to hydraulic-based wind turbines [6], where the gearbox and generator are replaced by a large positive-displacement pump using a hydraulic pipeline. The principle aims of this paper are to:

- Give an overview of developments in hydraulic-based wind turbine technology.
- Introduce the concept of using an open-loop system with seawater as the working fluid, illustrating a mathematical model for simulating the performance of such a system.
- Discuss the performance characteristics of the system, including an energy storage facility.

Hydraulic transmission applied to wind energy is not a new concept. Early works by JERICO (1981) and L. Rademakers (1988) showed that a lack of component availability is the main aspect hindering implementation [7].

Hydraulic transmission systems for large-scale wind turbines are already being developed in industry. A Norwegian company, ChapDrive™ [8], is currently developing a design that uses a low-speed radial piston pump connected to a high-speed hydraulic motor that is in-turn connected to a synchronous generator. The use of a variable displacement pump, such as the radial piston pump, allows for a continuously variable transmission (CVT) without a mechanical gearbox. The company also claims that their design does not require any frequency converters or transformers. This leads to a substantially lighter nacelle and an overall reduction of 20% in the price of power [8]. The company has tested 225kW and 900kW prototypes, and recently erected a 5MW turbine [8].

Another hydraulic energy transmission system is being developed by a Scottish company, Artemis Intelligent Power, Ltd. [9]. The company specialises in digital displacement transmission (DDT), which is a type of fluid transmission system that uses radial piston pumps and motors in conjunction with high-speed solenoid valves for transmission control. The company has been acquired by MHI (Mitsubishi Heavy Industries, Ltd.) who plan to implement the technology in a 7 MW offshore wind turbine called SeaAngel™ [9].

The initiatives mentioned above both implement a hydraulic transmission, and although they have their advantages, the potential benefits of a hydraulic-based turbine are not being fully exploited [10]. Of particular note is the fact that none of the designs makes an attempt to centralise electric power generation. The individual turbines still require an electric generator.

One of the attributes of being in the offshore environment is the abundance of seawater. By moving to an open-loop transmission system that uses seawater as the working fluid, hydro-electric conversion can occur on a centralised generation platform that could also be located onshore. This would substantially reduce generator costs, and the dependence on copper. Moreover, seawater has a relatively high bulk modulus, implying that it has the capability of transporting large amounts of energy though a relatively small diameter pipeline. It is abundant and at a steady low temperature, eliminating the need for a cooling system.

The concept of open-loop transmission is being utilised at the Delft University of Technology (TUDelft) in their Delft Offshore Turbine (DOT) [1], [6]. The idea behind this turbine design is to have the rotor directly drive a fixed displacement radial piston pump in the nacelle. This pump is connected via closed loop hydraulic transmission (using an oil circuit) to a variable displacement swash-plate pump at the base of the turbine. The swash-plate pump draws in seawater and transmits it to a centralised generation platform. At the platform, a pelton wheel and generator convert the pressurized fluid into electrical energy [10].

Simulations of the performance of such a system have already been carried out [7], [11], [12] and results indicate that if such a turbine is to compete with current designs, losses in the energy conversion processes need to be minimised. Simulations by Laguna [7] on the DOT indicate energy conversion losses are around 3 times greater than conventional turbine losses. The technology definitely needs to be cheaper and simpler than what is currently the norm to encourage a shift. Finally, the results [7] indicate the since some of the fundamental designs aspects of a tradition turbine are no longer in place, one can consider the use of alternative control schemes that increase the power harvesting capabilities of the turbine.

The present study makes an attempt to simplify the turbine design by eliminating the intermediate closed-loop oil circuit. This would imply a directly driven swash-plate pump that would pump seawater from the base of the turbine to the nacelle and back down to the generation platform. A schematic is shown in figure 1 below.

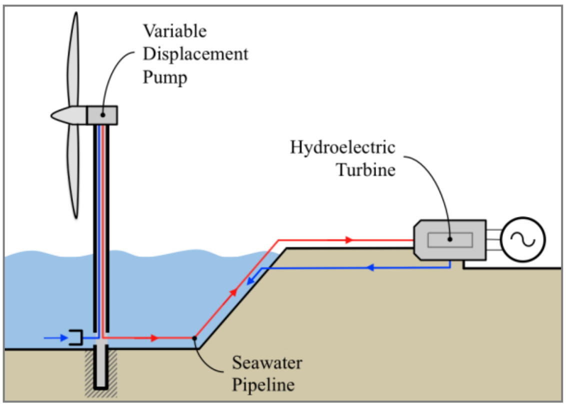

Figure 1: The proposed open-loop system.

This alternate design is much simpler and cheaper. Eliminating the closed-loop oil circuit means that the radial piston pump and hydraulic motor, along with the oil reservoir, cooling system and other components are not required. A performance analysis has already been carried out on such a system [13]. The study also considers the possibility of using the pressure in the hydraulic circuit as a control variable, along with swash-plate control and rotor-pitch control. Results [14] indicate that below rated wind speed, operating efficiencies for such a system are up to 5% less than for a conventional turbine. However, above the rated wind speed, the efficiency can be as much as 30% greater [14]. Preliminary performance data also indicates that using hydraulic pressure control, the annual energy yield can improve by 15% in regions with an average wind speed of 7ms^{-1} [14].

Current work is focused on a more holistic approach, whereby the non-linear behaviour of the pelton wheel is also included in the model and a more elaborate view of the energy extraction characteristics can be obtained. Previous simulations were based on optimum pelton wheel efficiency. However, in reality this efficiency varies with operating conditions [12], and this behaviour can now also be simulated.

One of the main challenges is to solve, or at least mitigate, one of the age-old problems with renewable energy resources: an intermittent or sporadic supply, and an independent, variable demand. Wind energy in particular, is often heavily criticised for its variable output characteristics [2].

Inevitably a wind turbine will at times be in a position to generate more energy than the actual demand. A solution to the variable output problem would be to store the extra energy produced for periods when for that period when there is not enough wind to satisfy the energy demand. With current turbine designs generating electricity in the nacelle, it is extremely difficult to store their generated output. Battery storage in the multi-megawatt scale requires quite a complex system at that scale, along with the environmental concerns associated with the chemicals used in batteries. Currently, one of the largest battery banks in the world is in situated in Fairbanks, Alaska. The system uses 1,376 Ni-Cd batteries along with sophisticated power electronics and monitoring systems. It is designed to provide spinning reserve and can handle a 26MW load for 15 minutes [15].

Another option is to utilise the concept of pumped hydro storage (PHS). This is a form of mechanical energy storage, whereby water is pumped to high elevations using pumps running on excess electricity in periods of low demand. When power is required, the water is allowed to drain to a lower elevation through a hydroelectric turbine [16]. PHS systems offer the largest storage capacity by a substantial margin, one of the few systems with supply times greater than 24 hours [16]. The technology is also highly developed and understood, the first form of central energy storage was a PHS system built in 1929 [17].

The main drawbacks of a PHS system are the infrastructural requirements. The energy storage capacity depends on two main aspects: the volume of water that can be stored and the height at which it can be stored [18]. Finally, it must also be noted that when using electrical energy to power pumps, a PHS facility is ultimately a net consumer of electricity. This is because it takes more energy to pump the water to the reservoir than the electricity retrieved as it falls through the turbine [19]. Losses associated with pumped storage are between 75 and 80% [16]. Despite these drawbacks it is still the most well understood and widely implemented system for storing energy in the world, with over 200 installations resulting in a total of 129GW in operation in 2012 [16].

PHS can be particularly relevant to island regions having small, isolated grids [20], [21]. The island of Gran Canaria, one of the Canary Islands, has population of 755,489 inhabitants (2004) and a surface area of 1532km^2 [20]. In 2002, 2,725GWh was generated in conventional power stations [20].

With an isolated grid, high levels of wind sourced energy penetration would imply substantial disturbances caused by wind speed fluctuations. This would give rise to supply and demand mismatch, along with dangerous frequency and voltage variations [22]. For this reason, grid-connected wind generation has to be limited to the synchronous capacity of the grid, which depends on the system load [20], [22].

The island of Gran Canaria has at its disposal two reservoirs with a height difference of 281m. By utilizing these reservoirs for a PHS facility, simulation results from Bueno and Carta [20] indicate that wind generated energy on the grid can be increased by around 22%. The system would require the installation of a 20.4MW wind farm and a 17.8MW modular pumping system to compensate for supply-demand mismatch [20].

A similar analysis was carried out in the Faroe Islands [21], where controllable energy storage was also shown to allow for a higher penetration of renewable energy. By modelling the concept of load following to control power from the storage facility, it was shown that fluctuations in wind farm output could be directly compensated for. The PHS was also successful as the primary means for controlling the frequency of the power system [21].

With the numerous positive characteristics of PHS, particularly to increase the penetration of grid-connected wind generation, it is definitely worth considering this form of storage when designing a large-scale hydraulic wind farm, with centralized electrical generation. The system would be able to directly utilise the hydraulic energy generated from the wind. It is therefore proposed that a storage facility be directly connected to the hydraulic transmission system, to correct supply-demand mismatch without requiring electrical energy conversion as in all previously discussed systems. This would substantially reduce conversion losses and simplify the system. Moreover, the storage facility would not be a net consumer of electrical energy, as in traditional PHS [19]. A schematic is shown in figure 2 below.

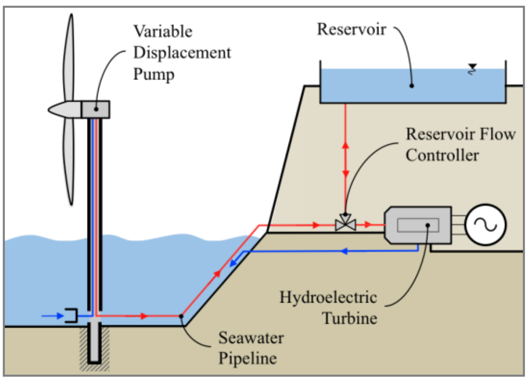

Figure 2: The proposed open-loop system with storage.

In this paper two systems will be modeled:

- A single hydraulic turbine with an open-loop system using a seawater pipeline and a pelton-type hydroelectric turbine with flow control (Figure 1).
- An identical system, but with a hydro storage facility connected to the system (Figure 2).

2 MATHEMATICAL MODELLING

A mathematical model of the system discussed earlier is derived to allow for analysis of its performance characteristics. The model of the turbine is based on previous work [13], [14] and caters for the overall performance of the system.

Different physical aspects of the system are brought together using the *physical network approach* [23]. Parameters that affect only transient behaviour are neglected.

2.1 Rotor Model

The wind turbine rotor is based on the NREL 5MW Baseline Wind Turbine, a benchmark turbine developed by NREL [24]. The turbine uses generator torque control to optimise the rotor below the rated wind speed, beyond which an active pitch control scheme regulates output power to the rated value. The rotor characteristics are shown below.

Table 1: Rotor Characteristics of the NREL 5MW Baseline Turbine [24].

Rating, Orientation	5MW, Upwind
Number of Blades	3
Rotor, Hub Diameter	126m, 3m
Hub Height	90m
Cut-In, Rated, Cut-Out Wind Speed	3ms^{-1}, 11.4ms^{-1}, 25ms^{-1}
Cut-In, Rated, Rotor Speed	6.9rpm, 12.1rpm
Maximum Power Coefficient ($C_{p\text{-}max}$)	0.482
Optimum Tip Speed Ratio (λ_{opt})	7.55

The rotor performance characteristics are based on previous work [13] where the rotor was modelled using blade-element-momentum theory with Prandtl tip and root loss corrections. The rotor power (P_R) and torque (M_R) are given by:

$$P_R(u) = \begin{cases} 0, & [u < u_{in}] \\ \left(\frac{\pi}{2}\rho_a R^2 C_{P-max}\right) u^3, & [u_{in} \leq u < u_r] \\ P_{Rat}, & [u_r \leq u < u_{out}] \\ 0, & [u \geq u_{out}] \end{cases} \quad (1)$$

$$M_R = \frac{P_R(u)}{\omega_R} \quad (2)$$

2.2 Positive-Displacement Pump Model

The variable displacement swash-plate pump was modelled using a lumped-parameter model with the method of constant loss coefficients. This method is based on the fact that the instantaneous efficiency of a pump can be expressed in terms of three dimensionless parameters that remain constant over different operating conditions [25], [26]. Each parameter (C_s, C_d, C_f) describes a form of energy loss. Based on these three parameters the pump mathematical model can be defined as follows:

$$M_P = V_p p_P \left[1 + C_d\left(\frac{\mu\omega_P}{p_P}\right) + C_f\right] \quad (3)$$

$$Q_P = V_p \omega_P \left[1 - C_s\left(\frac{p_P}{\mu\omega_P}\right)\right] \quad (4)$$

The constant loss coefficients are obtained from the nominal operational set point and from the no-load characteristics of the pump, both are typically available from manufacturer data.

In equations (3) and (4) the term (V_p) is the volumetric displacement of the pump (m^{3}rad^{-1}). By continuously varying this parameter the torque loading on the rotor can be optimised to consistently maintain the optimum tip-speed ratio in the region ($u_{in} \leq u < u_r$) [7]. This is analogous to generator torque control. The optimised volumetric displacement is given by:

$$V_p = \left(\frac{\frac{\pi}{2}\rho_a R^3 C_{P-max}}{\lambda_{opt}}\right)\left(\frac{\eta_{P-m}}{p_P}\right)u^2 \quad (5)$$

For a swash-plate pump this would correspond to a particular value of the swash-plate angle [7].

2.3 Pipeline Model

The pipeline is modelled as a frictional load that acts between the pump and pelton turbine. This non-linear pressure load is modelled using the Darcy Equation [27]:

$$p_{fr} = \frac{8fL\rho}{\pi^2 D^5} {Q_P}^2 \quad (6)$$

Where the friction factor for the circular pipeline is computed as follows [23], [27]:

$$f = \begin{cases} \frac{64}{\text{Re}} & \text{Re} < 2300 \\ \left[-1.8\log\left(\frac{6.9}{\text{Re}} + \left(\frac{e/D}{3.7}\right)^{1.11}\right)\right]^{-2} & \text{Re} \geq 2300 \end{cases} \quad (7)$$

In the case of turbulent flow (Re ≥ 2300), the equation is based on the Colebrook and White formula [27], which also accounts for the internal surface roughness of the pipe. The Haaland approximation [28] is used to increase computational efficiency [23].

2.4 Pelton Wheel Model

The hydroelectric energy conversion occurs through the use of a pelton wheel. This consists of a wheel with a number of buckets attached to it. Nozzles convert hydraulic pressure into kinetic in the form of a water jet that impinges on the buckets causing the wheel to rotate. The wheel is connected to an electrical generator by a driveshaft.

There are other types of hydroelectric turbines, however a pelton-wheel is best suited for this application given the high pipeline pressure required by the system [12].

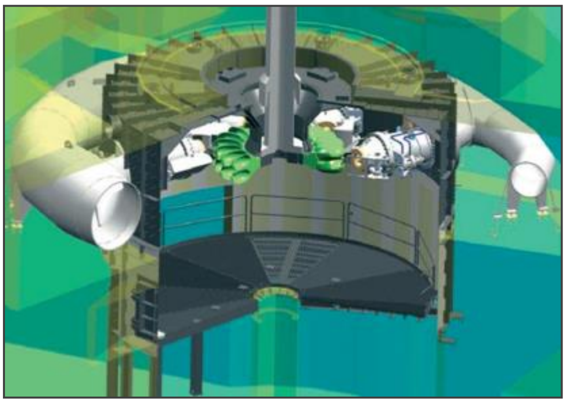

Figure 3: A vertical-axis, multi-jet pelton wheel [29].

The non-linear pressure load exerted by a pelton wheel is a function of the jet velocity and can be obtained by using the Bernoulli equation across the nozzle:

$$p_T = \frac{1}{2}\rho\left(\frac{v}{C_V}\right)^2 \tag{8}$$

The term C_V is the nozzle velocity coefficient, it accounts for the fact that some pressure is used to overcome friction at the nozzle walls. Typical values are 0.97 [30]. The flow rate through the pelton wheel is the product of the nozzle area and its velocity requirement [12]:

$$Q_T = \frac{Q_{Jet}}{C_C C_V} = vA_N \tag{9}$$

The term C_C is the nozzle contraction coefficient, it accounts for the fact that the jet will continue to contract after it leaves the nozzle until it reaches the "*vena contracta*". Typical values range from 0.92 to 1.0 for round orifices [30]. The product $(C_C C_V)$ is the nozzle discharge coefficient.

By equating forces on a pelton wheel bucket, it can be shown that the hydraulic power imparted by the jet onto the bucket is given by:

$$P_H = \rho A_N v(R_T\omega_T)(v - R_T\omega_T)(1 + k\cos\vartheta) \tag{10}$$

The term $(R_T\omega_T)$ gives the velocity of the buckets, and the ratio of this speed to the jet speed is the bucket speed ratio $\left(\phi = \frac{R_T\omega_T}{v}\right)$. This ratio is a crucial parameter in the performance of the pelton wheel since it can be shown that the turbine reaches its maximum efficiency when: $\phi = 0.5$ [12].

In order to maintain a constant bucket-speed ratio over the entire operating range it is necessary to continuously vary the nozzle area depending on the load. In practice this is achieved by means of a servo-controlled spear valve [31].

For a known generator mechanical power demand (P_G), the required nozzle area is given by:

$$A_N = \frac{P_G}{\rho\eta_{TG}(R_T\omega_T)^3(1 + k\cos\vartheta)}\left[\frac{\phi^2}{1-\phi}\right] \tag{11}$$

2.5 Storage System: Reservoir Model

As previously discussed, a reservoir can act as a form of energy storage, directly connected to the hydraulic transmission. In this case it will be modelled as being connected between the pipeline and the pelton turbine.

The flow into the reservoir must compensate for any discrepancies between flow supply from the pump and flow demand in the pelton wheel. From the principle of flow conservation:

$$Q_R = Q_P - Q_T \tag{12}$$

By convention, a positive value of Q_R implies that the reservoir is filling up, that is, extra energy is being stored. The change in reservoir head over an interval Δt is then given by:

$$\Delta h = \frac{1}{A_{RES}}\int_0^{\Delta t} Q_R(t)\,dt \tag{13}$$

The flow rate to the reservoir can also be defined in terms of the line pressure and the reservoir head, using the Bernoulli equation and making the area of the reservoir flow inlet/outlet into the transmission system subject of the formula:

$$A_R = \left|\frac{Q_R}{\sqrt{2\left|hg - \frac{p_T}{\rho}\right|}}\right| \tag{14}$$

By continuously altering this area, as in the case of the pelton nozzle, the optimum line pressure can be maintained. In practice this can be achieved using a slot-type, variable area orifice, much like a sluice gate [12].

3 COMPUTATIONAL IMPLEMENTATION

The above model was programed using a MATLAB® algorithm. The program is fed an array of time-varying wind speeds and power demands.

In order to solve the model and obtain the performance characteristics at every time-step, equations (1) to (14) above are solved sequentially and combined using the *physical-network approach* as discussed above. Other parameters such as component efficiencies can then be computed to give insight into the overall system performance.

The decision to store energy in the reservoir or drain from it is taken based on the flow deficit or surplus, which is computed using equation (12). The required flow rate is achieved by varying the passage area using equation (14).

In order to simulate a system without storage, the term Q_R is set to zero in equation (12), and equations (13) and (14) are neglected. In this case the pelton wheel can only be optimised when delivering maximum power; optimisation is carried out by varying the nozzle area. An iterative process is used to compute the required nozzle area at the operating conditions, and hence solve the model.

It must be noted that although simulations are time-based, results are all steady-state values. The model does not take transient parameters such as inertia and compressibility into account.

4 RESULTS

Results of the simulations are discussed in this section. In all cases the pipeline is taken to be 1km in length, with a 0.35m internal diameter.

4.1 System Performance with No Storage

The first results are of the stand-alone turbine without any form of energy storage.

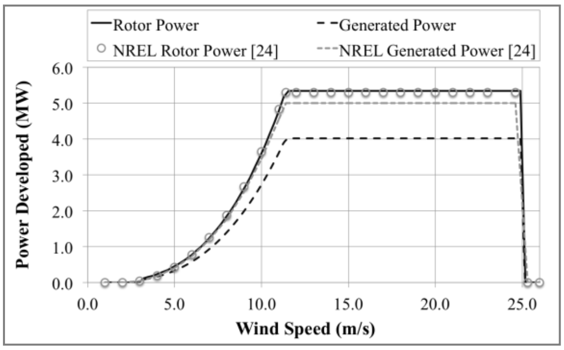

Figure 4: A graph of power versus wind speed. Shown are simulated results along with data from NREL [24] for the rotor being modelled.

As can be seen in figure 4 above, the mechanical power generated by the rotor is in excellent agreement with the NREL reference data [24]. This acts as a form of validation for the model, since there are no differences in the boundary conditions. However, the generated power is quite different. Losses in the transmission result from the pump, pipeline, pelton wheel and generator (the latter modelled with an efficiency of 95% [12]). Although these losses are substantially greater than the reference data, it must be taken into consideration that the hydraulic turbine is simulated with a control scheme of a traditional turbine for comparative reasons. In practice the control scheme would be tailored for the design, and this can allow higher energy extraction beyond the rated wind speed [7], [13]. In itself, this can already compensate for the resulting inefficiencies.

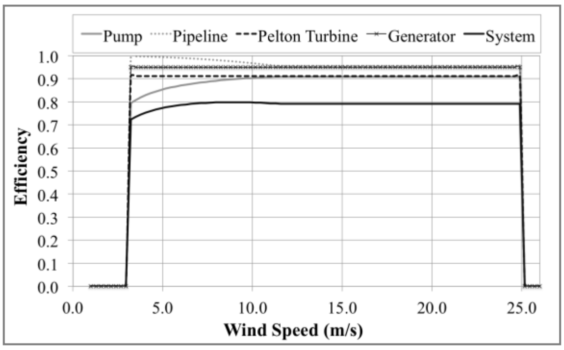

Figure 5: A graph of sub-component and overall efficiency versus wind speed.

Figure 5 above shows the efficiencies of the various sub-components. The major cause of energy loss is the pelton wheel, these losses can be categorised into frictional losses in the nozzle, hydraulic losses as the jet impinges on the buckets and finally windage and mechanical losses in the pelton shaft that also result in a minor loss of energy [12]. The pump efficiency increases as its pressure load increases and the pipeline induces slightly higher losses as the flow rate increases.

The steady-state values of pressure and flow rate are shown in figure 6 below.

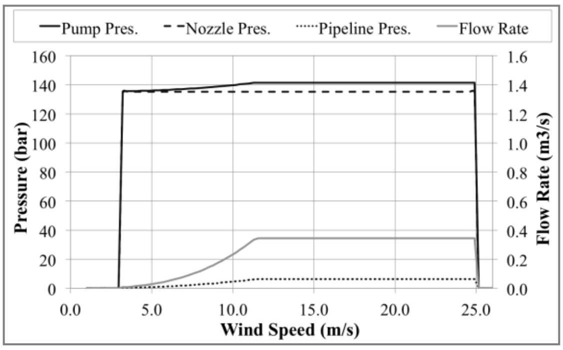

Figure 6: A graph of sub-component pressures and the flow rate through the system versus wind speed.

As shown in figure 6, the nozzle pressure remains at a steady 135 bar for this particular set-up. It can be shown that for a pelton turbine operating at a constant rotational speed, the pitchcircle-diameter of the buckets determines the resulting nozzle pressure [12]. The flow rate corresponds directly to the power variation, as required for operation at constant pressure. The pressure across the pipeline increases slightly with the flow rate, as expected.

4.2 System Performance with Reservoir Storage

Here the same turbine is connected to a pumped storage facility consisting of a 75x75m reservoir. The head difference is taken relative to the hydroelectric turbine. The program is fed hourly values of wind speed and power demand over a 24 hour period, both data sets are qualitatively based on those used by Bueno and Carta [20]. These values are averaged from 6 months of data and are used to illustrate the concept in a qualitative way.

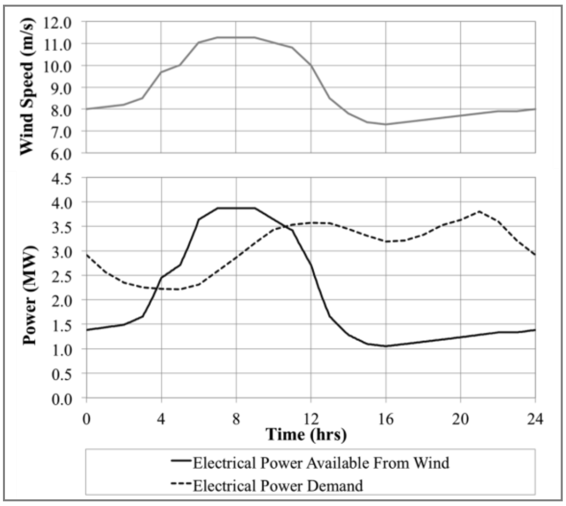

Figure 7: A graph of wind speed, corresponding rotor power and the actual power requirement from the turbine.

Figure 7 above illustrates a common situation with many renewable energy technologies. For this particular case, the turbine is not able to meet the demand except for an approximately 7-hour period, starting at hour 4, where the demand is actually being exceeded. For a standard system, the power deficit would need to be compensated for, typically using conventionally generated energy that uses non-renewable sources.

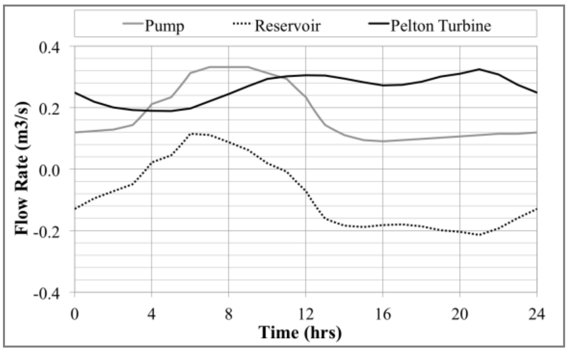

Figure 8: A graph of system flow rates. Note that a negative reservoir flow rate implies a draining reservoir.

With the possibility of energy storage, the supply-demand mismatch is addressed by the reservoir flow rates, as shown in figures 8 and 9. The system pressures and therefore efficiencies are maintained by controlling the flow rate from the reservoir using a variable area orifice.

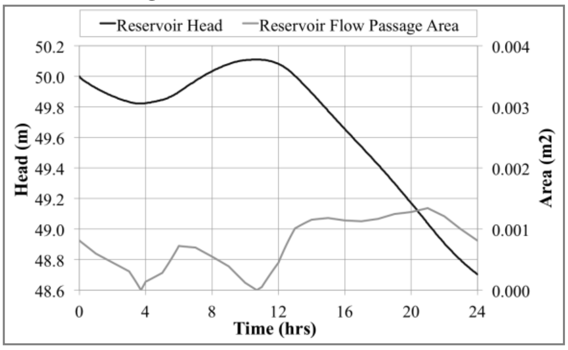

Figure 9: A graph of instantaneous reservoir head and flow passage area with time.

5 CONCLUSION

The concept of a wind turbine utilising a hydraulic transmission has been illustrated. Results show that energy conversion losses so far indicate an inferior performance when directly compared to standard turbines. However, using alternative control schemes [7] [13], such systems become more viable. Moreover, results for a turbine coupled with a storage facility indicate the potential for such a system to address the problem of supply-demand mismatch by directly storing hydraulic energy, without intermediate electrical conversion.

In practice the storage capacity will need to be optimised for site-specific conditions depending on electricity demand trends, wind farm size and the terrain available.

Future work will focus on more detailed analyses, along with a feasibility study for operating such storage systems and the use of seawater has a working fluid [32].

NOMENCLATURE

A_N	Nozzle Area	P_{Rat}	Rotor Rated Power
A_R	Reservoir Flow Area	Q_{Jet}	Jet Flow Rate
A_{RES}	Reservoir Area	Q_P	Pump Flow Rate
C_c	Contraction Coefft.	Q_R	Reservoir Flow Rate
C_d	Damping Coefft.	Q_T	Pelton Flow Rate
C_f	Friction Coefft.	R	Rotor Radius
C_s	Slip Coefft.	R_T	Pelton Wheel Radius
C_V	Velocity Coefft.	u	Wind Speed
D	Pipeline Diameter	v	Nozzle Jet Velocity
e	Pipeline Roughness	V_p	Pump Displacement
f	Friction Factor	ϕ	Bucket Speed Ratio
h	Reservoir Head	ϑ	Bucket Split Angle
k	Bucket Coefficient	μ	Seawater Viscosity

L	Pipeline Length	ρ	Seawater Density
M_P	Pump Torque	ρ_a	Air Density
p_{fr}	Pipeline Pres. Loss	η_{Pm}	Pump Mech. Eff.
p_P	Pump Pressure	η_{TG}	Pelton Mech. Eff.
p_T	Pelton Pressure	ω_P	Pump Ang. Velocity
P_H	Pelton Hydro. Power	ω_T	Pelton Ang.Velocity
P_R	Rotor Power		

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