Modeling and control of an open accumulator Compressed Air Energy Storage (CAES) system for wind turbines

Mohsen Saadat, Farzad A. Shirazi, Perry Y. Li

Department of Mechanical Engineering, University of Minnesota, Minneapolis, MN 55455, USA

HIGHLIGHTS

- Open accumulator enables near constant pressure compressed air energy storage.
- Maximizes energy production, levels load, downsizes electrical parts, meets demands.
- Near isothermal liquid piston compressor/expander increases thermal efficiency.
- Power dense hydraulic power path and energy dense pneumatic power path.
- Distributes control efforts according to hydraulic/pneumatic paths' bandwidths.

ABSTRACT

This paper presents the modeling and control for a novel Compressed Air Energy Storage (CAES) system for wind turbines. The system captures excess power prior to electricity generation so that electrical components can be downsized for demand instead of supply. Energy is stored in a high pressure dual chamber liquid-compressed air storage vessel. It takes advantage of the power density of hydraulics and the energy density of pneumatics in the "open accumulator" architecture. A liquid piston air compressor/expander is utilized to achieve near-isothermal compression/expansion for efficient operation. A cycle-average approach is used to model the dynamics of each component in the combined wind turbine and storage system. Standard torque control is used to capture the maximum power from wind through a hydraulic pump attached to the turbine rotor in the nacelle. To achieve both accumulator pressure regulation and generator power tracking, a nonlinear controller is designed based on an energy based Lyapunov function. The nonlinear controller is then modified to distribute the control effort between the hydraulic and pneumatic elements based on their bandwidth capabilities. As a result, liquid piston air compressor/expander will loosely maintain the accumulator pressure ratio, while the down-tower hydraulic pump/motor precisely tracks the desired generator power. This control scheme also allows the accumulator to function as a damper for the storage system by absorbing power disturbances from the hydraulic path generated by the wind gusts. A set of simulation case studies demonstrate the operation of the combined system when the nonlinear controller is utilized and illustrates how this system can be used for load leveling, downsizing electrical system and maximizing revenues.

1. Introduction

Renewable energy sources such as wind and solar energy are clean and available as long as the wind is blowing or the sun is shining. However, they suffer from intermittency and that they are often not available when the demand is high. For example, wind energy tends to be more abundant at night when power demand is low (Fig. 1). Variations in wind speed and solar intensity make integrating wind and solar energy into the electric power grid a challenge. An energy storage system can provide steady and predictable power by storing excess energy and releasing it when the demand is greater than supply [1,2].

In this paper, we consider an energy storage concept for wind turbines especially those that are off-shore. The capacity factor of current off-shore wind turbines are typically less than 50% [3] so that the electrical generator and collection and transmission
systems are significantly under-utilized. For off-shore wind turbines, collection and transmission is a major balance-of-plant cost. By storing the energy prior to generation of electricity, the electrical components can be downsized for demand instead of supply [4,5].

Compressed Air Energy Storage (CAES) is advantageous for this application in that: (1) it has a relatively high energy density (at 350 bar, ~ 3 MW × 8 h energy requires 500 m³) compared to pumped hydro (~144,000 m³ at 100 m elevation); (2) it is scalable (energy capacity scales linearly with storage vessel volume); (3) potentially cost effective and has a long life cycle relative to electric batteries; (4) it is not dependent on specific geographic sites as needed by a conventional CAES or pumped hydro storage system [8]. In a conventional CAES system, excess electricity is used to drive an air compressor that compresses air into an underground salt cavern; and the energy is then retrieved by pre-compressing and improving the efficiency of natural gas combustion in a gas fired turbine [9,10]. Such a system is relatively inefficient (<50%), requires use of hydrocarbon fuel, and depends on geographic sites.

An alternate novel Compressed Air Energy Storage (CAES) concept for wind turbines was proposed in [11] in which compressed air is stored in high pressure (~200–350 bar) vessels (Fig. 2). Excess energy from the wind turbine is stored locally, prior to electricity generation, as compressed air in a storage pressure vessel. This allows electrical components to be downsized. The compressor/expander used to store and extract energy operates nearly isothermally so that it is efficient. A variable hydraulic drive, instead of a mechanical gearbox, is used for power transmission. This improves the reliability of the transmission system and allows the generator and the storage system to be housed down-tower, thus reducing construction and repair costs. In addition, a cost effective fixed speed induction generator can be used instead of the combination of a permanent magnet synchronous generator and power electronics for frequency and voltage conversion.

Two major challenges are: (1) compressor/expanders are generally not very efficient or powerful; (2) the pressure in the storage vessel reduces as compressed air in the storage vessel depletes, making it difficult for the air compressor/expander to maintain either its efficiency or power at all energy levels. The first challenge is overcome by developing a liquid piston air compressor/expander [12] with enhanced heat transfer using porous media [13] and droplet sprays [14], and reduced leakage. The latter is overcome by deploying an open accumulator configuration with a dual chamber storage vessel for both liquid and compressed air, such that energy can be stored/retrieved hydraulically and pneumatically [15]. By coordinating the hydraulic and pneumatic paths, the pressure can be maintained regardless of the energy content. Because hydraulic components are more power dense, they can be used for peak transient power while the air compressor/expander can be downsized for steady power. An appropriate controller that coordinates these two power paths is essential for the simultaneous pressure regulation, tracking of the desired generator power, and maximizing wind power capture in the presence of supply or demand power variations. The efficiency and performance of the CAES system depend significantly on the design of the controller.

Multiple facets of research is needed to achieve the proposed CAES. This paper focuses on the mathematical modeling and the controller design of the overall system, including the turbine, storage and generator. The control objectives are to maximize wind energy capture, meet the electrical demand, and to regulate the open accumulator pressure. A feature of the controller is that it respects the bandwidth limitation of the air compressor/expander relative to the hydraulic pump/motor to regulate accumulator pressure. In addition, simulation results are provided to illustrate the operating characteristics of the system. A portion of this paper was presented in [16] which contains a preliminary control design based on a feedback linearized plant and more limited modeling and illustrative results.

The rest of the paper is organized as follows. Section 2 is a brief overview of the CAES architecture. Detailed mathematical modeling is presented in Section 3. Nonlinear controller design approach and its modification to meet the actuator bandwidth limitation is presented in Section 4. A sample system sizing is given in Section 5. Simulation results to illustrate the performance and utility of the system are given in Section 6. Conclusions are given in Section 7.
2. System overview

The proposed CAES with an open accumulator architecture [15] is shown in Fig. 2. A variable displacement hydraulic pump (B) attached to the wind turbine rotor (A) in the nacelle converts wind power to hydraulic power. At down-tower (H), a variable displacement hydraulic pump/motor (C), a near-isothermal liquid piston air compressor/expander (F) and a fixed speed induction generator (G) are connected in tandem on a common shaft. They are powered by the pump (B) and exchange power with the storage vessel (E) with both liquid (hydraulic fluid) and compressed air at the same pressure. This allows energy to be stored in or extracted from (E) either hydraulically (as in a conventional hydraulic accumulator) or pneumatically (as in a conventional air receiver). In both cases, energy is stored in the compressed air. By coordinating the hydraulic and the pneumatic power paths, the pressure in (E) can be maintained constant regardless of energy content, unlike a conventional closed hydraulic accumulator with only a hydraulic port or a compressed air receiver with only a pneumatic port. For example, as compressed air is being released from (E), some liquid can be added to reduce the compressed air volume to maintain the pressure.

The pneumatic power branch makes better use of the vessel (E) volume than the hydraulic power branch (a compressed air tank stores 20 times more energy than a hydraulic accumulator at the same peak pressure and total volume [15]) but hydraulic pump/motors are more power dense than the pneumatic compressor/expanders. This architecture can take advantage of both by utilizing the hydraulic path to accommodate high power transient events such as wind gust or sudden generation power demand, and reserving the pneumatic path for steady power.

The liquid piston air compressor/expander (F) consists of an air compression/expansion chamber filled with porous media (F1), and a liquid piston pump/motor (F2). The porous material is used to increase the heat transfer surface area [13,17,18]. A liquid piston (water) can flow through the porous material and provides a tight seal for the compressed air.1 Heat transfer can also be enhanced additionally by water spray [14,19]. When storing energy pneumatically, the liquid piston pump/motor (F2) pumps water into the compression/expansion chamber, compressing the air within it. Heat of compression is transferred to the porous material and to the water to maintain a near-isothermal operation. When the chamber pressure exceeds that of the storage vessel (E), the compressed air is ejected and stored in the vessel. The chamber is then refilled by releasing the water and filling it with atmospheric air for the next cycle. When retrieving energy, the compressed air is released into the expansion chamber. As the air expands, the liquid piston retreats, the liquid piston pump/motor (F2) is motored and work is derived. Heat is supplied from the porous material to maintain the temperature of the expanding air.

3. Modeling

3.1. Wind turbine

A wind turbine extracts wind’s kinetic energy from the swept area of its rotor blades. The aerodynamic torque for a given wind speed \( V_w \) and rotor speed \( \omega_r \) on the wind turbine shaft is given by [20,21]:

\[
T_w = \frac{1}{2} \rho_a \pi R^2 C_T(\beta, \lambda) \frac{V_w^3}{\omega_r}.
\]  

where \( \rho_a \) is the air density, \( R \) is the radius of turbine, \( C_T \) is the power coefficient, \( \beta \) is the collective pitch angle and \( \lambda \) is the tip speed ratio. For maximum power capture within the turbine capability (i.e. in region 2), the turbine tip speed ratio should be at its optimal setting \( \lambda_{opt} = 8.1 \) and \( \beta = 0 \) (see Fig. 3). At high (region 3) wind speeds, \( \beta \) is adjusted for power curtailment.

3.2. Nacelle (B) and down-tower (C) pump/motors

The nacelle pump (B) is directly connected to the wind turbine with speed \( \omega_s \) and the down-tower pump/motor (C) is in series with the generator and air compressor/expander with speed \( \omega_{ak} \). Both units are connected to the accumulator (E) at pressure \( P_{acc} \). Thus, the hydraulic flow \( Q \) and torque \( T \) for the nacelle (B) and down-tower (C) pump/motors are given by:

\[
Q_p = \frac{D_p(t)}{2\pi} \omega_r - L_p(D_p, \omega_r, r)
\]

\[T_p = -\frac{D_p(t)}{2\pi} P_0(r-1) - \Gamma_p(D_p, \omega_r, r)
\]

\[
Q_{pm} = \frac{D_{pm}(t)}{2\pi} \omega_g - L_{pm}(D_{pm}, \omega_g, r)
\]

\[T_{pm} = -\frac{D_{pm}(t)}{2\pi} P_0(r-1) - \Gamma_{pm}(D_{pm}, \omega_g, r)
\]

where the subscripts “p” and “pm” refer to the nacelle pump and the down-tower pump/motor respectively, \( D_{p/pm}(t) \) are the input displacements of the pump/motors and \( L_{p/pm}() \) and \( \Gamma_{p/pm}() \) are the flow and torque losses, \( r := P_{acc}/P_0 \) is the ratio of the accumulator (E) pressure \( P_{acc} \) to the ambient pressure \( P_0 \). The nacelle pump displacement is restricted to \( D_p(t) \in [0, D_{pm}^{max}] \) while the down-tower pump/motor displacement can go over center: \( D_{pm}(t) \in [-D_{pm}^{min}, D_{pm}^{max}] \). In our notation, \( D \cdot \omega > 0 \) implies that the pump/motor is pumping. The overall efficiency of a hydraulic machine is a function of its displacement, pressure and speed [22]. A sample pump efficiency for a fixed line pressure is shown in Fig. 4. Note that efficiency is reduced as displacement is low.

---

1 The water would ideally be taken from the sea or lake which are excellent heat reservoirs. However, filtering, cleanliness and corrosion issues need to be addressed.
The air compressor/expander uses mechanical energy from the rotational shaft to compress air for storage, and extracts energy from the compressed air through expansion to drive the shaft. In the liquid piston air compressor/expander, the transmission is via a water column and the liquid piston pump/motor (F2). The compression/expansion chamber (F1) is filled with a porous material for enhanced heat transfer as described in Section 2. When storing energy, (i) the compression chamber is filled with atmospheric air; (ii) water is pumped into the compression chamber, compressing the air; (iii) when the pressure reaches $P_{acc}$ of the storage vessel (E), air is injected into (E); (iv) the liquid piston retroacts, a valve opens to the atmosphere, the chamber is filled with atmospheric air and the cycle repeats. When extracting energy, (i) the chamber is filled with water; (ii) some amount of compressed air is injected from the storage vessel to chamber; (iii) air valve closes and the compressed air expands pushing the liquid piston and motoring the hydraulic pump/motor; (iv) a valve opens, ejecting the expanded air to the atmosphere as the liquid piston refills the chamber and the cycle repeats. The liquid piston pump/motor operates cyclically with a period of $\sim 1-2$ s.

The tradeoff between work input/output (hence efficiency) and compression/expansion times (hence power) is heavily dependent on the compression/expansion trajectory [23,24]. To capture this tradeoff in an overall system model, a continuous cycle-average model for the compressor/expander is developed here based on a family of Adiabatic–Isothermal–Adiabatic (AIA) compression/expansion trajectories for the liquid piston proposed in [23] (Fig. 6). This assumption is made because in the case when the $hA$ product (where $h$ is the convective heat transfer coefficient and $A$ is the heat transfer area) is constant, AIA trajectories result in optimal Pareto tradeoff between efficiency and power density [23]. In implementation, the actual optimal trajectories used would also have similar characteristics as AIA trajectories. Moreover, viscous liquid friction losses inside the chamber have been shown to be small compared to thermodynamic losses with proper implementation [25]. Leaks can be lumped into the flow loss of the liquid piston pump/motor.

The control input to the liquid piston air compressor/expander is the cycle average displacement $D_{lp}$, together with shaft speed $\omega_g$ and pressure ratio $r$, the air mass flow rate and liquid piston pump/motor torques are:

$$m = \rho_b \left[ \frac{D_{lp}(t)}{2\pi} \omega_g - L_{bp}(D_{lp}, \omega_g, r) \right]$$

where $Q_{acc}$ and $m$ are the liquid volumetric flow rate and mean air mass flow rate into the storage vessel and $\rho_b = P_b/(RT_b)$ is the air density at the ambient condition.

The energy in the compressed air with volume $V$ and pressure ratio $r$ is defined to be the maximum work achievable through an isothermal expansion at $T_0$ [23]:

$$E_{at}(r, V) = P_b V (r \ln(r) - r + 1)$$

The actual compression work and the work that can be extracted from the compressed air differ from $E_{at}(r, V)$. The compressor and expander efficiencies are respectively the ratios of the stored energy to the actual work input, and of the actual work output to stored energy.

---

2 The liquid can either be hydraulic oil or clean water. A barrier or bladder may be used to separate the liquid and compressed air if air absorption into the liquid becomes problematic.
\[ T_p = -\frac{D_p}{2\pi} (P_w - P_0) - \Gamma_p (D_p, \omega_g, r) \quad \text{(A.6)} \]

where \( P_w (D_p, \omega_g, r) \) is the cycle average in chamber pressure, \( L_p (\cdot) \) and \( \Gamma_p (\cdot) \) are the volumetric and mechanical losses of the liquid piston compressor/expander. Detail derivation of these expressions as well as the thermal efficiencies of the compressor/expander are given in Appendix A. Since both the generator shaft speed \( \omega_g \) and the accumulator pressure ratio \( r \) will be maintained at their desired values \((r = 200, \omega_g = 1800 \text{ rpm})\), the cycle-average pressure \( P_w \) and thermal efficiency of the chamber are mainly functions of cycle average displacement \( D_p \). Fig. 7 shows that efficiency decreases as \( |D_p| \) increases. Also, cycle average pressure \( P_w \) is small compared to accumulator pressure \( rP_0 \). This is the reason why air compressor/expander are much less power dense than hydraulic pump/motors.

### 3.5. Induction generator

A three-phase induction generator is assumed. Its static torque-speed relationship is given by [20]:

\[ T_g (\omega_g) = \frac{3p_Tr_L}{2(\omega_s - \omega_g)} \times \frac{V_s^2}{(r_s + \frac{r_L}{r_m - r_g})^2 + (X_s + X_r)^2} \quad \text{(10)} \]

where \( \omega_s \) is the generator speed, \( \omega_g = 2\pi f_s / p_T \) is the synchronous speed with \( f_s = 60 \text{ Hz} \) is the grid frequency, \( r_s \) and \( r_L \) are the resistances and \( X_s \) and \( X_r \) are the leakage reactivities of the rotor/stator, \( V_s \) is the voltage and \( p_T \) is the number of poles. The machine is a generator when \( \omega_s - \omega_g < 0 \) and is a motor otherwise. From Eq. (10), at a given grid frequency \( f_s \), \( \omega_g \) is the only free variable, therefore controlling \( \omega_g \) is equivalent to controlling the generator power:

\[ \text{GeneratorPower} = T_g (\omega_g) \cdot \omega_g = f (\omega_g) \quad \text{(11)} \]

### 3.6. Inertia dynamics

The inertia dynamics of the turbine rotor is:

\[ J_r \ddot{\omega}_r = T_p + T_w \quad \text{(12)} \]

where \( J_r \) is the moment of inertia of the rotor, \( T_p \) is the hydraulic pump (B) torque and \( T_w \) is the torque applied by wind (given by Eqs. (1) and (4)). The inertia dynamics of the generator shaft is given by:

\[ J_g \ddot{\omega}_g = T_p + T_{pm} + T_{d} \quad \text{(13)} \]

where \( J_g \) is the moment of inertia of the generator shaft and \( T_{pm} \), \( T_p \) and \( T_{d} \) are respectively the down-tower pump/motor (C) torque in (6), the liquid piston air compressor/expander (F2) torque in (A.6) and the generator torque (G) in (10).

### 3.7. Summary of dynamic model

In summary, the overall dynamics of the combined wind turbine and the storage system are:

\[ \dot{\omega}_r = \frac{D_p}{2\pi} T_p (r - 1) - \Gamma_p (D_p, \omega_r, r) + \frac{1}{2} \rho_0 \pi R_c^2 \rho_s L_p (\beta, \lambda) \frac{V_s^2}{\omega_s} \quad \text{(14)} \]

\[ \dot{\omega}_g = \frac{D_{pm}}{2\pi} T_p (r - 1) - \Gamma_{pm} (D_{pm}, \omega_r, r) - \Gamma_p (D_p, \omega_g, r) \quad \text{(15)} \]

\[ \dot{V}_s = \frac{1}{2\pi} (D_p \omega_r + D_p \omega_g r + D_{pm} \omega_r r - L_p (D_p, \omega_r, r) - L_{pm} (D_{pm}, \omega_r, r)) \quad \text{(16)} \]

\[ \dot{V}_g = \frac{1}{2\pi} (D_p \omega_r + D_p \omega_g r + L_p (D_p, \omega_r, r) + L_{pm} (D_{pm}, \omega_r, r)) \quad \text{(17)} \]

where “\( r \)” and “\( g \)” are subscripts standing for the turbine rotor and generator. Note that \( D_p \), \( D_{pm} \) and \( D_p \) are control inputs to the dynamic system.

### 4. Controller design

The overall control objectives of the combined wind turbine and the storage system can be summarized as: (i) capturing maximum available power from wind by controlling the wind turbine angular speed; (ii) maintaining the accumulator pressure ratio during the energy storage or regeneration mode; and (iii) providing the required power demanded by the electrical grid. Since the wind turbine shaft and the generator shaft are not coupled in the proposed architecture, it is possible to achieve all these goals at the same time. Wind turbine control is performed by utilizing standard torque control approach through the hydraulic pump (B) located in the nacelle. The accumulator pressure and generator power will be controlled by the down-tower variable displacement pump/motor (C) as well as the liquid piston air compressor/expander unit (F2). Notice that while the controller relies on the generator shaft speed error, the actual error is determined by calculating the generator power (from the measurement of its phase voltage and current) instead of measuring its shaft speed.

#### 4.1. Storage system and generator

In conventional CAES systems, the pressure in the storage vessel drops as compressed air in the storage vessel depletes, making it difficult for the air compressor/expander to maintain either its efficiency or power at all energy levels. In the open accumulator design, it is possible to maintain the pressure no matter how much compressed air is inside the vessel. Therefore, the controller should maintain the desired pressure on the generator shaft and the desired pressure ratio in the accumulator. Because the storage vessel is assumed to have enough space for the compressed air, the air volume dynamics given by Eq. (17) is not controlled at this level but will be controlled by a high level supervisory controller instead.

To design the nonlinear controller, a generator shaft speed error \( \dot{\omega}_g \) and accumulator pressure ratio tracking error \( \dot{r} \) dependent Lyapunov function \( E (\dot{\omega}_g, \dot{r}) \) motivated by the kinetic and potential energies of the system defined to be:

\[ E := \frac{1}{2} J_g \dot{\omega}_g^2 + \int_{0}^{V_1} P_0 \ddot{r} dV \quad \text{(18)} \]
where $J_g$ is the generator shaft inertia, and the errors are defined as:
\[
\dot{r} = r - r_d \tag{19}
\]
\[
\dot{\omega}_g = \omega_g - \omega_g^d \tag{20}
\]
with $r_d$ and $\omega_g^d$ being the desired pressure ratio and generator speed. Note that the shaft speed error dependent part of $E$ takes the form of a kinetic energy, and the pressure dependent part is the required energy to compress/expand a fixed mass of air at pressure $p_0$ and volume $V$ to the desired pressure $r_d p_0$ isothermally. Then the Lyapunov function at the current pressure ratio $r$ and volume $V$ becomes:
\[
E = \frac{1}{2} J_g \dot{\omega}_g^2 + p_0 V \left( r \ln \left( \frac{r}{r_d} \right) - \dot{r} \right) \tag{21}
\]
By taking time derivative of Eq. (21) and substituting equivalent terms from Eqs. (15)–(17), we have:
\[
\dot{E} = -F \dot{\omega}_g - G \dot{r} - H \gamma(\dot{r}) \dot{r}^2 \tag{22}
\]
where $F$, $G$, and $H$ are defined as:
\[
F = \frac{D_{pm}}{2\pi} (r - 1) + \frac{D_p}{2\pi} \left( \frac{p_e}{p_0} - 1 \right) + \frac{\Gamma_{pm} + \Gamma_p + T_g}{p_0} \tag{23}
\]
\[
G = -\frac{1}{2\pi} \left( D_p \dot{\omega}_g + D_{pm} \dot{\omega}_g^d + \frac{D_p}{r_d} \omega_g^d \right) + \left( \frac{L_p}{r_d} + \frac{L_{pm}}{r_d} \right) \tag{24}
\]
\[
H = \frac{D_p}{2\pi} \omega_g - L_p \tag{25}
\]
\[
\text{since } \ln \left( \frac{r}{r_d} \right) \text{ is a monotonically increasing function of } \dot{r}, \ln \left( \frac{r}{r_d} \right) \text{ is replaced by:}
\]
\[
\ln \left( \frac{r}{r_d} \right) = \frac{r}{r_d} - \gamma(\dot{r}) \dot{r}^2 \tag{26}
\]
where $\gamma$ is a strictly positive function of $\dot{r}$. Now, if $D_{pm}$ and $D_p$ are controlled such that $F = K_1 \dot{\omega}_g$ and $G = K_2 \dot{r}$ where $K_1$ and $K_2$ are two positive gains, then $\dot{E}$ becomes:
\[
\dot{E} = -p_0 K_1 \dot{\omega}_g^2 - p_0 (K_2 + H \gamma(\dot{r})) \dot{r}^2 \tag{27}
\]
Therefore, if $K_2 > 0$ is chosen large enough, $K_2 + H \gamma(\dot{r})$ is always positive and $\dot{E}$ is negative definite. Note that this analogy is valid over a large range of pressure ratios since:
\[
\text{Max}(\gamma(\dot{r})) = 10^{-4}, \quad \dot{r} \in [-200, 200] \tag{28}
\]
For example, if $\min(H) = -4 \text{ m}^2/\text{s}$ then $K_2 > 4 \times 10^{-4}$ will satisfy this requirement. Since $E$ is positive definite and radially unbounded, based on Lyapunov stability criteria, system is asymptotically stable at the desired pressure ratio and generator speed [28]. Therefore, according to Eqs. (23) and (24), the displacement commands for the storage system can be found by solving the system of equations given by:
\[
(r - 1) \frac{D_{pm}}{2\pi} + \left( \frac{p_e}{p_0} - 1 \right) \frac{D_p}{2\pi} + K_1 \dot{\omega}_g - \frac{\Gamma_{pm} + \Gamma_p + T_g}{p_0} \tag{29}
\]
\[
\omega_g \frac{D_{pm}}{2\pi} \dot{\omega}_g^d \frac{D_p}{r_d} = -K_2 \dot{r} - \omega_g \frac{D_p}{2\pi} \left( L_p + L_{pm} \right) \frac{L_p}{r_d} \tag{30}
\]
\[\text{Note that } p_e, \Gamma, \text{ and } L \text{ are dependent on the control inputs } D_{pm} \text{ and } D_p. \text{ In the implementation, they are calculated using the } D_{pm} \text{ and } D_p \text{ from the previous time step to simplify the computation of the control inputs.}
\]
Although the convergence of states is guaranteed by this control, a practical drawback of this controller is that the high frequency terms appear in the control command of the compressor/expander ($D_p$). These high frequency signals, either from the wind gust or sudden variation in power demand, can adversely affect the liquid piston air compressor/expander performance and may reduce its operation life. In the case of a transient high power wind, it will be more efficient to store its energy via the hydraulic path directly into the accumulator by-passing the down-tower hydraulic pump/motor or pneumatic components. Furthermore, storing this transient high power through the hydraulic path can allow the down-tower hydraulic pump/motor and liquid piston air compressor/expander to be sized for mean wind power instead of peak power.

4.2. Control effort distribution based on subsystems' bandwidths

The bandwidth related concern above can be solved by using a unique feature of the storage system. The pressure and the generator shaft dynamics in Eqs. (15) and (16) show that although the generator shaft speed has a fast dynamics in response to the actuator displacements, the accumulator pressure dynamics is relatively slow (except for the case when the air volume inside the storage vessel is too small). This property is used here to modify the nonlinear control commands to relax the high frequency concern. The idea is to use a low pass filter to remove the high frequency components of the liquid piston pump/motor displacement command signal. Let $\bar{D}_{p}$ be the solution to Eqs. (29) and (30). Then the dynamics for the low-pass filtered liquid piston pump/motor displacement control input are:
\[
\bar{D}_p = \frac{1}{\tau} (\bar{D}_p - D_{p}) \tag{31}
\]
where $\tau$ is a time constant that depends on the available bandwidth of the pump/motor in the liquid piston air compressor/expander. To ensure that generator speed control is not compromised, $D_{pm}$ is obtained by solving (29) again with $D_{p}$ given by the solution to Eq. (31).

The stability of both states (pressure and speed) can be proved even by such a modification [27]. In this way, the liquid piston pump/motor ($F_2$) will loosely control the accumulator pressure ratio around its desired value during the storage or regeneration phase. On the other hand, the hydraulic pump/motor ($C$) will precisely control the generator shaft speed to maintain the desired generation power (Fig. 8).

This control scheme has an additional advantage in that the accumulator acts as a damper by absorbing the high frequency flow fluctuations from the pump in the nacelle. These high frequency fluctuations...
power flows that are mainly caused by the wind gust will be stored in the accumulator through the power-dense hydraulic path.

The nonlinear control law given in Eqs. (29) and (30) uses pressure and generator shaft speed as feedback. A small error in generator speed can lead to large error in estimated generator power. In practice, it is more reasonable and realistic to measure the instantaneous generator power based on its phase voltage and current. A small error in generator speed can lead to large error in estimated generator power. In this case, \( \omega_s - \omega \) gives the maximum power factor \( C_p > C_{p_{\text{max}}} \).

\[ C_p \geq C_{p_{\text{max}}} \left( \frac{\lambda}{\lambda_r} \right)^3 \]

4.3. Wind turbine torque control

The nacelle pump (B) is used to control the turbine speed via the standard torque controller to maximize the power capturing [28]:

\[ T_p = -K_s \omega_s^2 \]

\[ K_s = \frac{1}{2} \rho_0 \pi R_t^4 \frac{C_{p_{\text{max}}}}{\lambda_r^3} \]

\( T_p \) refers to the generation mode. The generator is assumed to operate below its maximum speed. If the generator is assumed to operate below its maximum speed, this condition is satisfied. The control design presented above is an improvement over our preliminary control design in [16] in several ways. The energy based Lyapunov approach here makes use of the structural property of the system and is expected to be more robust than the feedback linearization based approach in [16]. Furthermore, the current approach makes direct use of the bandwidth properties of the hydraulic and pneumatic power paths for the distribution of control efforts. In contrast, an ad hoc approach was taken in [16] which is to simply reduce the pressure control gain.

\[ \dot{\omega}_r = \frac{1}{2 J_r} \rho_0 \pi R_t^4 \frac{C_p}{\lambda_r^3} \left( C_p^\text{max} - \frac{C_p}{\lambda_r^3} \right) \]

\[ \Rightarrow \begin{cases} C_p \leq \frac{C_p^\text{max}}{\lambda_r^3} : \dot{\omega}_r < 0 \\ C_p > \frac{C_p^\text{max}}{\lambda_r^3} : \dot{\omega}_r > 0 \end{cases} \]

Speed regulation of rotor at higher wind speeds (region 3) is typically achieved using a PID controller:

\[ \beta(t) = K_p \tilde{\omega}_r(t) + K_i \int_0^t \tilde{\omega}_r(\tau) d\tau + K_d \frac{d\tilde{\omega}_r(t)}{dt} \]

where \( \tilde{\omega}_r = \omega_r - \omega \) is the rotor speed error, the difference between the desired rotor speed and the measured rotor speed. In this region, the primary objective is to limit the turbine power so that safe mechanical loads are not exceeded. Power limitation can be achieved by pitching the blades or by yawing the turbine out of the wind (not considered here), both will reduce the aerodynamic torque below what is theoretically available from an increase in wind speed [28].

Please cite this article in press as: Saadat M et al. Modeling and control of an open accumulator Compressed Air Energy Storage (CAES) system for wind turbines. Appl Energy (2014), http://dx.doi.org/10.1016/j.apenergy.2014.09.085
5. System sizing study

As a case study, the storage system is designed for an offshore wind turbine with a rotor radius of 45 m (power rated at 3 MW for wind speed rated at 12 m/s). The wind profile is generated by superimposing a 10-min average wind speed profile and the corresponding turbulent wind (Fig. 11). Suppose a 600 kW constant power is demanded by the grid. Then, the storage system must be able to store about 16 MW h of energy. At a pressure ratio of 200, from Eq. (9), the storage vessel size of $670 \, \text{m}^3$ is required. This size is quite compatible with turbines of this capacity and is smaller than the volume in the tower. To capture the maximum power at the rated wind speed, if directly coupled to the rotor, at pressure ratio of 200, a nacelle hydraulic pump (B) displacement $D_{\text{rated}} = 440 \, \text{l/rev}$ is needed at maximum speed of 20.5 rpm. Correspondingly, the hydraulic pump/motor (C) rotating at the generator speed (~1800 rpm) will need a displacement of $D_{\text{rated}} = 5 \, \text{l/rev}$.

To size the liquid piston air compressor/expander, we prescribe the maximum power to be 2 MW and the minimum thermal efficiency to be 90%. At this rated power, atmospheric air flow rate...
of 3.75 m³/s is needed. At 1 Hz, this corresponds to an air compressor/expander displacement of 3.75 m³. From (A.2) and (A.8), a heat transfer capability of hA = 59 kW/K would be sufficient. Since the cycle-average water pressure of the compression/expansion chamber is around 6.3 bar, the rated cycle-average displacement for the pump/motor connected to the chamber (F2) will be about 125 l/rev (at 1800 rpm). Note that the instantaneous flow rate and displacement for the liquid piston pump/motor would be larger to meet the requirement of the optimal compression/expansion profile. The bandwidth (Ω) of each actuator is also considered in the simulation. The rest of parameters and values used for this simulation are given in Table 1.

6. Simulation results

6.1. Long-term simulation: Overall system behavior

A long-term simulation (72 h) has been done to show how the overall storage system works and interacts with the wind turbine. The mean wind speed is a recorded series of data at 60 m elevation [29], while the turbulent wind speed is generated by Turbsim software based on the mean wind speed profile [30]. Fig. 11 shows the captured wind power through the pump located in the nacelle as well as accumulator energy level. Energy is stored during high wind speed conditions, and regenerated during low wind speed conditions. The desired electrical power is generated and the storage vessel pressure is maintained at a constant value by adding/removing liquid to/from the accumulator. In particular, notice that wind power as high as 3 MW is being captured while the generator is only sized at 600 kW (i.e. generator is downsized to 20% of wind turbine capacity, while all the available wind energy is captured and delivered to the electrical grid). In this scenario, the generator is operating at a capacity factor of almost unity.

As shown in Fig. 12 (top), the accumulator pressure ratio is always close to its desired value (200) no matter how much energy is stored. Since the accumulator plays the role of a damper for the storage system, the pressure deviation is directly related to the fluctuations in the turbine speed which affects the pump displacement (in the nacelle). Fig. 12 (bottom) shows the angular speeds of the turbine and generator. While the turbine speed changes to track the optimal tip speed ratio, generator shaft speed is maintained in order to produce a constant amount of power.

Fig. 13 (top) shows the displacements corresponding to the down-tower hydraulic pump/motor (Dpm) and the liquid piston air compressor/expander unit (Dlp). Dlp is responsible for controlling the energy storage/regeneration in longer time scales (i.e. 10-min) and Dpm is utilized to compensate for the low-pass filtering effect and to track the generator power demand (via controlling \( \omega_k \)). In particular, notice that the down-tower pump/motor works mostly as a motor, and at occasions when works as a pump, it operates at low displacement. This condition may cause low efficiency in pumping mode. This suggests that by dividing the pump/motor into a large motor and a relatively smaller pump/motor (also with a higher bandwidth), better performance can be obtained in both pumping and motoring modes. Fig. 13 (bottom) shows the air and liquid flows to the accumulator (negative values mean flow is from the accumulator). While being in opposite direction, the air mass flow rate and liquid flow rate have about the same ratio almost all the time. This shows how the controller maintains the pressure inside the vessel by adding/subtracting enough volume of liquid through the hydraulic path.

Fig. 14 shows the performance of the standard torque controller applied to the wind turbine shaft through the hydraulic pump in the nacelle. As shown, the controller tracks the desired wind turbine power vs. wind speed curve in region 2 of wind speed (\( V_{wind} \leq 12 \text{ m/s} \)), while the blade pitch angle is deviated from zero to cut the extra wind power in region 3 of wind speed (\( V_{wind} > 12 \text{ m/s} \)). The overall efficiency for the entire system (wind turbine and storage system) over the 72-h simulation is found to be 74.8%. A breakdown of the various pump/motors as well as the thermal loss inside the compression/expansion chamber is shown in Fig. 15.

Fig. 16 shows a sample power flow over different components inside the combined wind turbine and storage system. Two different time snapshots are used here to show how the system works over high and low wind power conditions. As it can be seen, for the case of high wind power, the accumulator is discharging with the rate of 1.24 MW, while in the low wind power situation, the accumulator is charging with the rate of 0.73 MW. From this figure, it is also possible to find the instantaneous efficiency of pump/motors as well as compression/expansion chamber.

* Reflects the amount of energy delivered to the grid by the generator relative to the amount of energy captured by the wind turbine.

Please cite this article in press as: Saadat M et al. Modeling and control of an open accumulator Compressed Air Energy Storage (CAES) system for wind turbines. Appl Energy (2014), http://dx.doi.org/10.1016/j.apenergy.2014.09.085
Fig. 17. Short-term simulation results. 1. Wind speed and desired generation power. 2. Displacements of down-tower pump/motor and the liquid piston air compressor/expander. 3. Accumulator air pressure ratio and air volume. 4. Generator power and wind turbine tip speed ratio. 5. Liquid and air flow rates to the accumulator.
6.2. Short-term simulation: Control task distribution

As discussed earlier, accumulator pressure regulation and generator power tracking tasks are shared between the hydraulic pump/motor and the liquid piston air compressor/expander based on their bandwidth capabilities. To see how it works, consider a wind speed and generation power demand profile as shown in Fig. 17-1 with rising and falling step changes (at times “a” and “c”) in power demand (e.g. from unexpected generator or transmission line failure). Moreover, a transient high speed wind (at “b”) that causes a large hydraulic flow from the nacelle to the CAES system is also assumed. The pump/motor bandwidths are

![Figure 18](image1.png)

Fig. 18. February 16–23, 2012 optimal revenue results for a 1.5 MW wind turbine. Electricity price and available power (top). Captured power, generated power and state of energy storage when the generator can optimally buy and sell electricity – case 4 (bottom). Price data is for N.E. USA taken from PJM [32] and wind data is for the Blandford, MA site taken from [29].

![Figure 19](image2.png)

Fig. 19. July 5–12, 2012 optimal revenue results for a 1.5 MW wind turbine. Electricity price and available power (top). Captured power, generated power and state of energy storage when the generator can optimally buy and sell electricity – case 4 (bottom). Price data is for the N.E. USA taken from PJM [32] and wind data is for the Blandford, MA site taken from [29].

Please cite this article in press as: Saadat M et al. Modeling and control of an open accumulator Compressed Air Energy Storage (CAES) system for wind turbines. Appl Energy (2014), http://dx.doi.org/10.1016/j.apenergy.2014.09.085
considered to be 0.5 Hz (in the nacelle and down-tower) while the maximum allowable frequency of the liquid piston air compressor/expander actuation command is 0.005 Hz.

In response to the step changes in the power demand, the pump/motor displacement changes quickly to inject/absorb the required power to the generator shaft (Fig. 17-2). This action is done through the power-dense hydraulic path of the accumulator which causes the air pressure to drop/increase quickly (Fig. 17-3). The mean pressure variation in the accumulator is however compensated by the air compressor/expander. The precise generator power tracking is robust to wind power fluctuations. In fact, the high frequency part of the captured power (in the form of fluctuating hydraulic flow) is stored in the accumulator through the power-dense and efficient hydraulic path, while the low frequency part is stored through the energy-dense pneumatic path after transforming into compressed air (Fig. 17-5). During all these events, the wind turbine is operating close to its optimal condition with the tip speed ratio of 8.1 which results in maximum power capturing (Fig. 17-4).

6.3. Profit comparison with and without CAES system

Price of electricity varies hourly during the day. They tend to peak in the morning (6–9 AM) and afternoon (5–8 PM) when demand is high. Price can vary during the day by several folds as illustrated in Figs. 18 (top) and 19 (top) which show price variations over 7 day periods between 18–90 $/MW h in February 2012 and between 30–270 $/MW h in July 2012. Occasionally, electricity price can peak as high as 2500 $/MW h on a hot day in Texas! A wind power plant can use large scale storage to shift the time of energy production to increase profit. Wind energy is stored during off-peak hours, energy is released to generate electricity when the price is high.

A case study using wind and electricity price data over two 7 day periods in July 2012 and February 2012 was conducted to illustrate this time shifting. Assuming a 1.5 MW rated wind turbine, the revenues are optimized using a deterministic optimal control methodology (details given in [31]) by selling the optimal amount of electricity at the real-time price. Four scenarios are considered and compared: (1) a conventional (90% efficient) gear box and (95% efficient) generator setup; (2) the proposed system in Fig. 2 but with no storage (equivalent to a hydrostatic wind turbine); (3) the proposed system in Fig. 2 with a 125 m3, 3 MW h (@ 200 bar nominal pressure) accumulator and the generator restricted to generating electricity; (4) the same system as in (3) but the generator can motor so that the system can buy and store electricity from the grid as well. For cases 3 and 4, since pressure is allowed to exceed the nominal pressure (up to 300 bar), energy stored can be larger than the rated 3 MW h.

The results are summarized in Table 2. The optimal operations for case 4 are shown in Figs. 18 (bottom) and 19 (bottom) for both periods, hydrostatic wind-turbine (with no storage) generates less revenue than the conventional wind turbine due to a lower system efficiency. This is due to the down-tower pump/motor (C) operating at low displacements and low efficiencies. Note that for both periods, the mean powers are quite low (286 kW for February 2012 period, and 178 kW for July 2012 period) compared to the rated power. With storage, more revenues are generated relative to the conventional wind turbine despite a 5–15% decrease in efficiency and a smaller amount of net electricity sold. As expected, case 4 provides more arbitrage opportunity to increase revenue by buying cheap electricity and selling it when price is high. The time shifting effect can be seen in Figs. 18 (bottom) and 19 where electricity generation is highly correlated with price (case 3, not shown, is similar except that electricity generation cannot be negative). However, the improvement in revenue is highly dependent on the price profiles. For the July 2012 period, revenue is increased by +124% over conventional case; whereas for the February 2012 period, which has a much more modest price variation, the improvement is only +15%.

7. Conclusions

A compressed air energy storage system for offshore wind turbines and its system model are presented. A continuous cycle-average model for a liquid piston air compressor/expander was extracted from the cycle-by-cycle operation with an optimal compression or expansion profile. A nonlinear controller was designed for the storage system according to an energy based Lyapunov function to meet power demand, regulate storage vessel pressure and to maximize wind energy capture. A modification of the control allows the hydraulic and pneumatic elements to operate advantageously according to their bandwidth and power density characteristics. Therefore, the slower air compressor/expander is sized for steady power and used to store/regenerate energy in longer time scales (steady), and the faster hydraulic components are sized and used to accommodate high transient power supply/demand. The accumulator also damps out capturing all the power fluctuations coming from the wind turbine pump (due to wind gusts). Agility of the smaller down-tower pump/motor is used to track the desired generator power precisely. The controller design methodology used in this work matches both the architecture of the storage system and the physical constraints of different subsystems in the proposed CAES system. Case studies demonstrated that the storage system allows the electrical component to be downsized to 1/5 of the turbine's capacity while capturing all the available wind energy and delivering a constant mean electrical power. The CAES can also aid grid integration and load matching, and to increase profit by selling electricity at higher prices. This can lead to wind energy becoming a dispatchable base power source to replace fossil-fuel.

Table 2

<table>
<thead>
<tr>
<th>Case</th>
<th>Wind captured (MW h)</th>
<th>Electricity sold (MW h)</th>
<th>Efficiency (%)</th>
<th>Revenue ($)/% increase over case 1</th>
<th>Wind captured (MW h)</th>
<th>Electricity sold (MW h)</th>
<th>Efficiency (%)</th>
<th>Revenue ($)/% increase over case 1</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Conventional wind turbine</td>
<td>48.1</td>
<td>41</td>
<td>85.5</td>
<td>$1140</td>
<td>30</td>
<td>25.6</td>
<td>85.5</td>
<td>$1560</td>
</tr>
<tr>
<td>2. Hydrostatic wind turbine w/no storage</td>
<td>48.1</td>
<td>35</td>
<td>72.7</td>
<td>$965.5/−15%</td>
<td>30</td>
<td>19.1</td>
<td>63.5</td>
<td>$1172/−25%</td>
</tr>
<tr>
<td>3. Proposed system w/3 MW h storage (sell)</td>
<td>48.1</td>
<td>36</td>
<td>75</td>
<td>$1213/+45%</td>
<td>30</td>
<td>21</td>
<td>70</td>
<td>$2303/+47%</td>
</tr>
<tr>
<td>4. Proposed system w/3 MW h storage (buy/sell)</td>
<td>48.1</td>
<td>52.3</td>
<td>80</td>
<td>$1310/−15%</td>
<td>30</td>
<td>48.6</td>
<td>79.5</td>
<td>$3510/+124%</td>
</tr>
</tbody>
</table>

* 17 MW h electricity is bought.
* 31.3 MW h electricity is bought.
Acknowledgements

This work is supported by the National Science Foundation under Grant EFRI 1038294 and the Institute for Renewable Energy and Environments (IREE) at the University of Minnesota under Grant RM-0027-11.

Appendix A. Cycle-average model for liquid piston air compressor/expander unit

In this appendix, we derive the cycle average compressed air mass flow rate \( \dot{m} \) and liquid piston pump/motor \((F_z)\) torque \( T_p \). Both quantities are averaged over a compression/expansion cycle \((\sim 1–2\) s\) and defined at a given common shaft speed \( \omega_k \), storage vessel pressure ratio \( r = P_{in}/P_o \), and cycle average displacement \( D_{lp} \) of the liquid piston pump/motor \( F_z \) which is the control input.

Within the compression/expansion cycle, the liquid piston flow rate varies to achieve an Adiabatic–Isothermal–Adiabatic (AIA) trajectory as in Fig. 6. The family of AIA trajectories is parameterized by \( T_1 \), the temperature of the isothermal segment. For compression, \( T_1 > T_0 \), and for expansion \( T_1 < T_0 \). The final temperature is given by \( T_1 = T_f^2/{T_0} \). Different choices of \( T_1 \) result in different power and efficiency as \( T_1 \) deviates from the ambient temperature \( T_0 \). Power increases but efficiency decreases. However, for a given heat transfer capability of the compressor/expander, specified by \( hA \) [W/K] (product of heat transfer coefficient and heat transfer area), the achieved efficiency (or power) is optimal at that power (or efficiency).

The time it takes to compress/expand a given mass of air with an optimal AIA trajectory specified by \( T_1 \) can be calculated as [23]:

\[
\dot{m} = \frac{hA|1 - T_0/T_1|}{R\left( \frac{\omega_k}{\pi} \ln \left( \frac{\omega_k}{\pi} \right) + \ln(r) \right)}
\]  \hspace{1cm} (A.1)

where \( r \) is the pressure ratio, and +/- signs correspond to compression \((T_1 > T_0)\) and expansion \((T_1 < T_0)\). Let \( Q_{lp} \) be the cycle average liquid piston flow rate. Since

\[
Q_{lp} = \frac{hA(1 - T_0/T_1)}{\rho_0 R\left( \frac{\omega_k}{\pi} \ln \left( \frac{\omega_k}{\pi} \right) + \ln(r) \right)}
\]  \hspace{1cm} (A.2)

On the other hand, we can express \( Q_{lp} \) in terms of the cycle-average flow rate of the liquid piston pump/motor and leakage as:

\[
Q_{lp} = \frac{D_{lp}(t)}{2\pi} \omega_k - L_{lp}(T_1, \omega_k, r)
\]  \hspace{1cm} (A.3)

where \( D_{lp} \) is the cycle average displacement of the liquid piston pump/motor \((F_z)\), \( L_{lp}(T_1, \omega_k, r) \) represents the volumetric losses in \((F_z)\) and in the compression/expansion chamber \((F_1)\). By equating (A.2) and (A.3), we can solve for \( T_1 \) in terms of \( (D_{lp}, \omega_k, r) \). Thus, hence forth, we shall consider the liquid piston pump/motor displacement \( D_{lp} \) as the control input the compressor/expander and the corresponding AIA trajectory with isothermal temperature \( T_1(D_{lp}, \omega_k, r) \) will be used within the cycle.

The P-V work required to compress/expand a unit mass of air to/from pressure ratio \( r \) with an AIA trajectory specified by \( T_1 \) including the isochoric final ejection/injection is given by [23]:

\[
P_w = \rho_0 E_p(T_1, r) + P_o
\]  \hspace{1cm} (A.5)

Generally \( P_w < r \cdot P_o \) since a large portion of the compression/expansion trajectory is at low pressure. For example, when \( r = 200, P_w < 7.5 \) in Fig. 7. This is the main reason air compressor/expander are much less power dense than hydraulic pumps/motors.

Given \((D_{lp}, \omega_k, r)\), the cycle average liquid piston pump/motor torque \( T_p \) and air mass flow rate are given as:

\[
T_p = -\frac{D_{lp}(t)}{2\pi} (P_w - P_o) - \Gamma_p(D_{lp}, \omega_k, r)
\]  \hspace{1cm} (A.6)

\[
\dot{m} = \rho_0 \left( \frac{D_{lp}(t)}{2\pi} - L_p(D_{lp}, \omega_k, r) \right)
\]  \hspace{1cm} (A.7)

where \( \Gamma_p(D_{lp}, \omega_k, r) \) captures mechanical losses in the pump/motor and in the liquid piston, and \( P_w(D_{lp}, \omega_k, r) \) is given by Eqs. (A.5) and \( L_p(D_{lp}, \omega_k, r) = L_p(T_1, \omega_k, r) \) in Eq.(A.3) with \( T_1 \) corresponding to \((D_{lp}, \omega_k, r)\). The thermal efficiency of the compression/expansion process is defined as the ratio between the stored energy in the air (including the ejection/injection work) and the total compression/expansion work \( (E_p) \):

\[
\eta_{lm} = \frac{\int \frac{\pi}{\omega_k} \left( \frac{\omega_k}{\pi} - 1 - \frac{T_1}{T_0} \ln \left( \frac{\omega_k}{\pi} \right) \right) + \ln(r)}{\int \frac{\pi}{\omega_k} \left( \frac{\omega_k}{\pi} - 1 - \frac{T_1}{T_0} \ln \left( \frac{\omega_k}{\pi} \right) \right) + \ln(r)}
\]  \hspace{1cm} (A.8)

where +/- signs correspond to compression and expansion modes.

References


Spera DA. Wind turbine technology – fundamental concepts. 2nd ed. ASME Press; 2009.


NREL. TurbSim user’s guide for version 1.40; September 2008.
