ABSTRACT
To achieve both accumulator pressure regulation and generator power tracking for a Compressed Air Energy Storage (CAES) system, a nonlinear controller designed based on an energy based Lyapunov function. The control inputs for the storage system are the pump/motor displacements inside the hydraulic transformer and the liquid piston air compressor/expander. While the pump/motor inside the liquid piston has a low bandwidth, the other pump/motor inside the hydraulic transformer has a relatively higher bandwidth. On the other hand, the pneumatic energy storage path of open accumulator has high energy density, whereas the hydraulic path is more power dense. The nonlinear controller is then modified based on these properties. In the proposed approach, the control effort is distributed between the two pump/motors based on their bandwidths: Hydraulic transformer reacts to high frequency events, while the liquid piston air compressor/expander performs a steady storage/regeneration task. As a result, liquid piston air compressor/expander will loosely maintain the accumulator pressure ratio, while the pump/motor in hydraulic transformer 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 due to wind gusts.

INTRODUCTION
Extracting energy from wind is perhaps one of the most attractive industries in renewable energy generation. However, the main disadvantage and constraint in providing a good performance and capacity for the wind turbines is the intermittency and mismatch between available wind power and electrical power demand. Therefore, large scale energy storage systems can be significantly useful to improve the capacity factor of wind farms by providing the steady and predictable power for grid as well as capturing maximum available wind power in normal situations. Storing energy in high pressure compressed air is attractive since increasing the pressure ratio of the compressed air can result an appreciable energy density. For example, at a pressure ratio of 350 (35 MPa), 170MJ of energy can be stored in 1m³ of volume. Other major benefits of CAES systems are their low cost and long operation life. A novel CAES system has been proposed and modeled in [1, 2] (Fig. 1). The excess energy from the wind turbine is stored in the storage vessel prior to electricity generation, while the generator power is maintained at the desired value (demand power from electrical grid). This allows to downsize the electrical components and reduce the involved power electronics. In particular, downsizing the generator will consequently improve the capacity factor of the system defined based on the generator size.

Two main challenges in the proposed CAES system are i) the low efficiency and power density of the air compressor/expander, ii) efficiency and power reduction of the compressor/expander when the pressure inside the storage vessel reduces as compressed air depletes. The first concern can be solved by deploying liquid piston air compressor/expander [3] with a chamber filled with porous materials, beside using optimal compression/expansion trajectory [4] and water spray cooling/heating method [5]. The open accumulator concept is a solution for the second issue [6]. Energy can be stored or extracted by pumping or releasing i) pressurized liquid similar to a conventional hydraulic accumulator or ii) compressed air similar to a conventional air receiver. In both cases, energy is stored in the compressed air. By coordinating the hydraulic and the pneumatic
paths, the pressure can be maintained regardless of energy content. However, this coordination for the purpose of pressure regulation can affect the generator power due to tandem shaft connection between the pump/motors connected to the pneumatic and hydraulic paths of the open accumulator as well as induction generator. Therefore, a good control algorithm is essential for simultaneous achievement of the pressure regulation and generator demand power tracking. The efficient performance of the CAES system is significantly dependent on this controller design. Note that both objectives should be satisfied in presence of supply or demand power variations.

In this paper, an energy-based controller is first developed to exponentially stabilize the system states and meet the above-mentioned control objectives. However, this controller does not take advantage of the frequency characteristics of the system as well as meeting the bandwidth requirement for the liquid piston air compressor/expander unit. Therefore, a filter is introduced in the controller implementation to channelize the high-frequency and low-frequency commands to the hydraulic pump/motor and the liquid piston compressor/expander, respectively. In this scheme, the pump/motor responds to the fast changes in the input power due to wind gusts and the liquid piston compressor acts against long-term variations in either wind speed or electrical grid demand power. Simulation results have been presented for the combined wind turbine and CAES system while the turbine torque is controlled by the standard torque controller.

### Modeling

A short summary of the system model is followed that describes different subsystems and their function in the combined wind turbine and storage system:

A variable displacement pump (B) is directly (no gearbox) attached to the wind turbine (A) in nacelle which converts wind power to hydraulic power. Such a direct coupling requires a comparatively large displacement pump to transmit a large power (i.e., order of MW) since the wind turbine angular speed is low (≤ 20rpm). At the ground level, there is a tandem connection of a variable displacement hydraulic pump/motor (C), a near-isothermal liquid piston air compressor/expander (F) and a fixed speed induction generator (G), all driven by the pump (B). The liquid piston air compressor/expander, the main unit for storing/regenerating energy, consists of a compression/expansion chamber filled with some porous material (F₁) and a liquid piston pump/motor (F₂). The porous material is used in addition to optimal compression/expansion trajectory and water spray to enhance the heat transfer inside the chamber to improve its thermal efficiency which has a significant effect on the overall efficiency of the storage system. The open accumulator (E) which is in fact the storage vessel contains both air and liquid. Hydraulic path can be utilized to accommodate high power transient events such as wind gust or sudden power demand (from grid), whereas the pneumatic path can be reserved for steady storage/regeneration function.

![Figure 1. CAES SYSTEM ARCHITECTURE](image)

In summary, the overall dynamic of the combined wind turbine and storage system can be found as:

\[
J_\omega \dot{\omega}_r = -\frac{D_p}{2\pi} P_0 (r - 1) - \Gamma_p(\omega_r) + \frac{1}{2} D_0 \pi R^2 C_p(\beta, \lambda) \frac{V^3}{\omega_r} \tag{1}
\]

\[
J_g \dot{\omega}_g = -\frac{D_{pm}}{2\pi} P_0 (r - 1) - \frac{D_{lp}}{2\pi} P_0 \ln(r) \eta_{irm}(\bar{p}_{lp}) - \Gamma_{pm}(\omega_g) \]

\[\quad - \Gamma_{lp}(\omega_g) - T_g(\omega_g) \tag{2}\]

\[
V = \frac{D_{lp}}{2\pi} \omega_g + \frac{D_p}{2\pi} \omega_r, r + \frac{D_{pm}}{2\pi} \omega_g r - L_{lp}(\bar{p}_w) \]

\[\quad - rL_p(r) - rL_{pm}(r) \tag{3}\]

\[
V = -\left(\frac{D_p}{2\pi} \omega_r + \frac{D_{pm}}{2\pi} \omega_g\right) + L_p(r) + L_{pm}(r) \tag{4}\]

where \(r, g, p, pm \) and \( lp \) are subscripts standing for turbine rotor, generator shaft, pump in nacelle, pump/motor in hydraulic transformer and the pump/motor connected to the liquid piston air compressor/expander chamber, respectively. In these equations, \( D \) is used to show the displacement for hydraulic actuators, \( \Gamma \) is the mechanical loss and \( L \) is the volumetric loss. Moreover, \( r \) is the pressure ratio of the air inside the accumulator and \( V \) is the volume of the compressed air. Additionally, \( R, \) and \( V \) are the radius of the wind turbine rotor and the wind speed while \( C_p \) is the turbine power factor as a function of blade’s pitch angle (\( \beta \)) and tip speed ratio (\( \lambda \)). It should be noted that while the displacement of the actual liquid piston pump/motor in a cycle varies rapidly to achieve the desired compression/expansion profile [4], a cycle mean value is used here shown by \( \bar{D}_{lp} \). In the other words, cycle-by-cycle behavior is approximated by a time average model. \( \eta_{irm} \) is then the thermodynamic efficiency of the
compression/expansion chamber over one cycle. Finally, $P_0$ and $\rho_0$ are the ambient pressure and density of air and $J$ is the angular moment of inertia.

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 over the storage or regeneration mode; and iii) Providing the required power demanded by the electricity grid. Since the wind turbine shaft and the generator shaft are not coupled in the proposed architecture, it would be 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 located in the nacelle. The accumulator pressure and generator power will be controlled by the variable displacement pump/motors inside the hydraulic transformer as well as the liquid piston air compressor/expander unit.

Storage System and Generator

In conventional CAES systems, 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. However, in the open accumulator design, it is possible to maintain the pressure no matter how much compressed air is inside the vessel. So, it is important to design a controller such that it maintains the desired power on the generator shaft as well as the desired pressure ratio in the accumulator.

Design of an appropriate nonlinear controller begins by choosing a suitable Lyapunov function. Here, the idea is to use an energy based Lyapunov function relying on the energy of generator shaft as well as compressed air inside the accumulator. Note that because the storage vessel is assumed to have enough space for the compressed air, the air volume dynamics given by Eqn. 4 will not be controlled at this level (i.e. a high level supervisory controller will control air volume). The Lyapunov function considering the generator shaft speed and accumulator pressure tracking errors is defined as:

$$E = \frac{1}{2} J g \dot{\omega}_s^2 + \int_v^V P_0 (r - r_d) \, dv$$

where the errors are defined as:

$$\dot{r} = r - r_d$$

$$\dot{\omega}_g = \omega_g - \omega_g^d$$

Note that the pressure dependent part of this Lyapunov function is in fact the required energy to compress/expand a fixed mass of air at pressure $r P_0$ and volume $V$ to the desired pressure of $r_d P_0$. In this definition, the energy level of air at desired pressure is zero. If such a compression/expansion takes place as an isothermal process, the energy level at current pressure ratio $r$ and volume $V$ is:

$$E = \frac{1}{2} J g \dot{\omega}_s^2 + P_0 V \left( r \ln \left( \frac{r}{r_d} \right) - \tilde{r} \right)$$

by taking time derivation of Eqn. 8 and substituting equivalent terms from Eqns. 2, 3 and 4, we will get:

$$E = -\dot{\omega}_s P_0 F - P_0 \left( G \tilde{r} + H \gamma(\tilde{r}) \tilde{r}^2 \right)$$

where $F$, $G$ and $H$ are:

$$F = \frac{D_{pm}}{2\pi} (r - 1) + \frac{\overline{D}_{ip}}{2\pi} \ln (r) \eta_{term}(\overline{D}_{ip})$$

$$G = -\frac{1}{2\pi} \left( \frac{\overline{D}_{ip} \omega_g}{r_d} + D_p \omega_r + D_{pm} \omega_g \right)$$

$$H = \frac{\overline{D}_{ip}}{2\pi} \omega_g - L_{ip}$$

Here, $\ln(r)$ is divided into two parts based on its Taylor series expansion around $r_d$ as:

$$\ln \left( \frac{r}{r_d} \right) = \frac{\tilde{r}^2}{r_d} - \frac{\tilde{r}^4}{3 r_d^3} + \ldots$$

Now, if $D_{pm}$ and $\overline{D}_{ip}$ 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:

$$\dot{E} = -P_0 K_1 \dot{\omega}_g^2 - P_0 \left( K_2 + H \gamma(\tilde{r}) \right) \tilde{r}^2$$

Therefore, if $K_2$ is chosen large enough, $K_2 + H \gamma(\tilde{r})$ is always positive and $\dot{E}$ is negative definite. Note that such an analogy is valid over a finite range of $\gamma(\tilde{r})$ on which the lower bound is known beforehand. Since $E$ is positive definite and radially unbounded, based on Lyapunov stability criteria, system is asymptotically stable at the desired pressure ratio and generator shaft speed [7]. According to Eqns. 10 and 11, $D_{pm}$ and $\overline{D}_{ip}$ can be
obtained as:

\[
\begin{align*}
[D_{pm}] &= 2\pi \left[ (r - 1) \ln(r) \mu_{pm}(D_{lp}) \right]^{-1} \times \left[ K_1 \ 0 \ K_2 \end{align*}
\]

\[
\begin{bmatrix}
\begin{pmatrix}
\Gamma_{pm} + \Gamma_{lp} + T_g \\
L_p + L_{pm} - \frac{D_{lp} \omega_r}{\tau_j}
\end{pmatrix}
\end{bmatrix}
\]

(15)

Although the convergence of states is guaranteed by this controller, there is a major practical drawback for this controller: High frequency terms appeared in the control command of pump/motor displacement in the liquid piston air compressor/expander unit (\(D_{lp}\)). Such a high frequency signal generated by the wind gust has bad effects on the pump/motor of the liquid piston air compressor/expander and may reduce its operation life significantly (in general, this subsystem has a relatively low bandwidth). In addition to wind gust, a fast demand power tracking can also be a source for creating high frequency command for the liquid piston pump/motor.

Control Effort Distribution (based on subsystems’ bandwidth)

The frequency concern can be solved by using a unique feature of the storage system. Investigating the pressure dynamic and generator shaft dynamic from Eqns. 2 and 3 reveals that while the generator shaft speed has a fast dynamic in response to the actuator’s displacements, the accumulator pressure dynamic is relatively slow (except for the case when the air volume inside the storage vessel is too small). Such a property is used here to modify the nonlinear control commands that relaxes 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:

\[
\dot{D}_{lp} = \frac{1}{\tau} (D_{lp} - \hat{D}_{lp})
\]

(16)

where \(\tau\) depends on available bandwidth of the pump/motor in liquid piston air compressor/expander. The effect of filtering (on generator shaft speed) will be compensated then by the pump/motor displacement command (inside the hydraulic transformer):

\[
\dot{D}_{pm} = -\frac{\ln(r)}{r - 1} \mu_{pm}(D_{lp}) \hat{D}_{lp} - \frac{2\pi}{P_0 (r - 1)} (\Gamma_{pm} + \Gamma_{lp} + T_g - K_3 \dot{\omega}_g)
\]

(17)

Note that the convergence of both states (pressure and speed) can be proved even by such a modification [8]. In this way, the liquid piston pump/motor 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 will precisely control the generator shaft speed to maintain the desired generation power (Fig. 2). Note that a precise control of generator shaft speed is important since generator power is a sensitive function of its speed. The control effort (to achieve both pressure regulation and precise shaft speed tracking) will be shared between two pump/motors inside the hydraulic transformer and liquid piston air compressor/expander based on their available bandwidths.

Such a control scheme has an other attractive outcome for the storage system: the open accumulator will play the role of a big damper for the system by absorbing all the high frequency fluctuations in the flow coming from the pump in nacelle. These high frequency power flows will be stored in the accumulator through the hydraulic path which has a high power density and is suitable for this task. As mentioned earlier, these high frequency components are mainly generated by the gusts in wind speed.

Wind Turbine Torque Control

Nacelle pump is used to control the turbine speed via the standard torque controller to maximize power capture [9]:

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

(18)

\[
K_s = \frac{1}{2} \rho_0 \pi R_p^5 \frac{C_{max}}{\lambda^*}
\]

(19)

Pump displacement \(D_p(t)\) is then set according to Eqn. (1). When the measurements and turbine model are perfect, the rotor speed converges to the optimal value (Fig. 3 and Eqn. 20). In region 2 of wind speed, \(\beta = 0\) in order to capture maximum wind power. In this case, \(\lambda^* = 8.1\) gives the maximum power factor.
\( C_p^{\text{max}} = 0.48 \).

\[
\dot{\omega}_r = \frac{1}{2\tau} \sum_{n} R e \omega_0 \varepsilon_0 \left( C_p \left( \frac{\omega_{\text{r}}}{\lambda} \right)^3 - C_p^{\text{max}} \left( \frac{\omega_{\text{r}}}{\lambda} \right)^3 \right)
\]

\[
C_p < \frac{C_p^{\text{max}}}{\lambda^3} \Rightarrow \dot{\omega}_r < 0
\]

\[
C_p > \frac{C_p^{\text{max}}}{\lambda^3} \Rightarrow \dot{\omega}_r > 0
\]

(20)

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

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

where \( \dot{\omega}_r = \omega_r - \omega^d \) 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.

![Figure 3. POWER FACTOR VERSUS TIP SPEED RATIO FOR ZERO PITCH ANGLE](image)

Simulation Results

A 65-hour simulation has been run to show how the combined wind turbine and storage/regeneration system works using the proposed nonlinear controller. The wind profile is generated by superimposing a 10-minute average wind speed profile and the corresponding turbulent wind (Fig. 4). The mean wind speed is a recorded series of data at 50m elevation (provided by Renewable Energy Research Laboratory (RERL)) while the turbulent wind speed generated by Turbsim software based on the mean wind speed profile [10]. It is assumed that the storage system is designed for a 3MW wind turbine while the demand power is set at 700kW and the electricity line frequency is 60Hz (over the whole time period). The total storage size is 500m³. The rest of the constant parameters used in this simulation are given in Table 1. Note that \( \Omega \) is the bandwidth considered for each actuator in the combined system. More details regarding these parameters can be found in [2].

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>( R_r )</td>
<td>45</td>
<td>m</td>
</tr>
<tr>
<td>( \beta_{\text{max}} )</td>
<td>40</td>
<td>degree</td>
</tr>
<tr>
<td>( \omega^d )</td>
<td>20.5</td>
<td>rpm</td>
</tr>
<tr>
<td>( f_s )</td>
<td>60</td>
<td>Hz</td>
</tr>
<tr>
<td>( V_r )</td>
<td>415</td>
<td>volt</td>
</tr>
<tr>
<td>( D^p_{\text{max}} )</td>
<td>630</td>
<td>lit/rev</td>
</tr>
<tr>
<td>( D^p_{\text{min}} )</td>
<td>6</td>
<td>lit/rev</td>
</tr>
<tr>
<td>( J_g )</td>
<td>20</td>
<td>Kg.m²</td>
</tr>
<tr>
<td>( \Omega_{\text{uu}} )</td>
<td>2</td>
<td>Hz</td>
</tr>
<tr>
<td>( \Omega_{\text{pp}} )</td>
<td>2</td>
<td>Hz</td>
</tr>
</tbody>
</table>

![Figure 4. MEAN AND TURBULENT WIND SPEED USED FOR SIMULATION](image)
turbine rotor speed is controlled such that the optimal tip speed ratio is maintained in the second region of wind speed. Note that the hydraulic pump in Nacelle has a relatively large displacement (rated at 630 lit/rpm) since the wind turbine speed is small and no gearbox is used. Fig. 6 (middle) shows the air mass flow rate and liquid volume flow rate to/from the accumulator. While liquid and air flows are in opposite directions, they have more or less the same ratio all the time. This shows how the controller tries to maintain the pressure inside the vessel by adding/subtracting enough volume of liquid through the hydraulic path. Fig. 6 (bottom) shows the displacements corresponding to the pump/motor located inside the hydraulic transformer and the pump/motor located in the liquid piston air compressor/expander unit. While the latter is responsible to control the energy storage/regeneration in longer time scales (i.e. 10-minutes), the former is utilized to compensate the low-pass effect and maintain the generator power (or shaft speed). Particularly, it is important to notice that the pump/motor in the hydraulic transformer works as a motor most of the time while in the pumping mode it just uses a small amount of its displacement. In order to solve possible practical issues regarding this fact, one can divide this pump/motor into a large motor and a relatively smaller pump/motor (with probably a higher bandwidth). In this way, the small pump/motor can be utilized at higher displacements which will increases its overall efficiency.

The performance of the standard torque controller applied on the wind turbine shaft through the hydraulic pump in nacelle is shown in Fig. 7. As shown, the controller tracks the desired wind turbine power versus wind speed curve in region 2 of wind speed ($V_{\text{wind}} \leq 12m/s$) while the blade’s pitch angles is deviated from zero to cut the extra wind power in region 3 of wind speed ($V_{\text{wind}} > 12m/s$). The overall efficiency for the entire system (wind turbine and storage/regeneration system) over the 65-hour simulation is found 72%. The loss stream regarding each pump/motor as well as the thermal loss inside the compression/expansion chamber is shown in Fig. 8.

**Conclusion**

A nonlinear controller designed for a CAES system that has been proposed and modeled in earlier works. An energy based Lyapunov function is derived for the purpose of controller design. A modification is then performed by utilizing a low-pass filter in one channel while compensating the filter effect by other control channel to solve the frequency issues for the actuators and meet their bandwidth constraints. Therefore, the liquid piston air compressor/expander which is naturally slow (low bandwidth) is used to store/regenerate energy in long time scales (steady). The pump/motor inside the hydraulic transformer that has a relatively higher bandwidth will then compensate the filtering effect to maintain the generator output power. In this way,
both short term and long term control objectives are satisfied by the same controller. An important consequence of such a control architecture is the useful role of the storage vessel (open accumulator) for the combined system: absorbing all the high frequency power fluctuations through the hydraulic path which has a high power density. The controller design methodology used in this work is highly matched with both the physical constraints and benefits of different subsystems in the proposed CEAS system.

Figure 7. WIND TURBINE CAPTURED POWER VS. WIND SPEED

Figure 8. LOSS STREAM FOR THE COMBINED SYSTEM BASED ON PERCENT OF INPUT ENERGY

REFERENCES


[3] Li, P., Van de Ven, J., and Sancken, C., Open Accumulator Concept for Compact Fluid Power Energy Storage, Pro-


