AN APPROACH TO REDUCE THE FLOW REQUIREMENT FOR A LIQUID PISTON AIR COMPRESSOR/EXANDER IN A COMPRESSED AIR ENERGY STORAGE SYSTEM (CAES)

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Abstract

A compressed air energy storage (CAES) system that uses a high pressure, isothermal air compressor/expander (C/E) has no carbon emission and is more efficient than a conventional system that uses fossil fuels. To be successful, the compressor/expander must be efficient and has high power density. However, there is a trade-off between efficiency and power density due to heat transfer. Our previous work has shown that by optimizing the compression/expansion trajectories in a liquid piston C/E, the power density can be improved by many folds without sacrificing efficiency. Yet, to achieve the optimized trajectory, this requires a large liquid piston pump/motor that often operates at low displacement, low efficiency regime. This paper proposes that by combining the liquid piston with a solid piston actuated via a hydraulic intensifier, the pump/motor size can be reduced significantly. A case study shows that with an optimal intensifier ratio, the pump/motor size is reduced by 85%, the ratio between maximum and minimum displacements is reduced by 7 folds, and the mean efficiency is increased by 2.4 times.

1 Introduction

Energy storage is recognized as key to integrating renewable energy into the electrical grid and compressed air energy storage (CAES) is a potentially cost effective and scalable means of doing so. In the past few years, our research team has been working towards a novel CAES system particularly suited for off-shore wind turbines [1, 2] that stores excess wind energy prior to electricity generation. The system uses an open accumulator architecture [3] and a near isothermal compressor / expander without the use of hydrocarbon fuel as in conventional CAES systems (Figure 1).

For this or other CAES systems to be successful, the compressor / expander must be capable of high pressure (200-300 bar) and is both efficient and power dense. In our approach, this is achieved using a liquid piston and porous media inserts (Fig. 2) to augment heat transfer, to prevent leakage of the compressed air, and to reduce dead volume [4, 5]. In addition, the compression / expansion trajectories, which are controlled by the liquid piston flow rate, are optimized so as to maximize the power at a given efficiency; or equivalently, to maximize the efficiency at a given power [6, 7, 8]. It has been shown that by merely optimizing the trajectories, power density can be increased by over many folds over ad-hoc trajectories, such as sinusoidal or linear profiles, without sacrificing efficiency (see e.g. Figure 4). This translates to a smaller system and lower capital cost. Our current compression / expansion time is between 1-2 sec. for a pressure of 210 bar.

Although optimizing the compression / expansion trajectories can improve efficiency/power-density trade-off, there is a price to be paid. Because the optimal trajectories consist of fast-slow-fast segments, the pump/motor for the liquid piston needs to be large and and must operate at both very high and very low displacements. This is costly and maintaining efficiency for both high and low displacements is challenging. This paper proposes a combined liquid piston and solid piston/hydraulic intensifier approach so one can reap the benefit of optimized trajectories without needing an excessively large pump/motor.

In section 2, the Pareto optimal compression/expansion trajectory concept and its challenges are reviewed. Section 3 presents the proposed combined solid/liquid piston approach. Section 4 uses a case study to illustrate the performance of the proposed approach during compression. Results for expansion are similar. Conclusions are given in section 5.

Figure 1: Open accumulator Compressed Air Energy Storage system coupled to a hydraulic wind turbine
2 Optimal compression/expansion trajectories

2.1 Liquid Piston Compressor/Expander

Figure 2 shows a schematic of the liquid piston compressor/expander. The upper portion of its chamber is filled with porous media that increases heat transfer area [4, 5]. As the liquid (water in our case) is pumped into the chamber, air above it is compressed. Similarly, as the compressed air expands, work is extracted via the pump/motor as liquid is withdrawn. Advantages of the liquid piston compared to a solid piston are 1) liquid can flow through porous media, 2) it forms a good seal, and 3) it eliminates potential dead volumes. Moreover, the velocity of the liquid piston, and hence compression/expansion rates can be controlled by controlling the liquid pump/motor.

![Schematic of a liquid piston compressor/expander](image)

Figure 2: Schematic of a liquid piston compressor/expander filled with porous media

2.2 Optimal efficiency–power density trade-off

Figure 3 shows the pressure-volume trajectories of a compression process and an expansion process with a pressure ratio \( r \). Initially, the compressor/expander chamber is filled with air at pressure-volume of \((P_0, V_0)\) and ambient temperature \( T_0 \). As the liquid piston rises, the air is compressed, along the profile \( \xi_c \) to \((rP_0, V_c)\) and typically at an elevated temperature. The valve to the storage vessel is then opened, and the compressed air is ejected at constant pressure into the storage vessel. Inside the storage vessel, it is assumed that there is sufficient time for the compressed air to cool isobarically to \( T_0 \). During the expansion process, the liquid piston lowers to charge the expansion chamber isobarically with compressed air from the storage vessel to volume \( V_e = V_0/r \). The intake valve is then closed, and the air at \((rP_0, V_e)\) is expanded along the profile \( \xi_e \) to \((P_0, V_e)\) and typically at a diminished temperature.

For these processes, the work input \( W_{in} \) during compression and work output \( W_{out} \) during expansion are the shaded areas as illustrated in Fig. 3. Since the initial temperature and ambient temperature used for heat transfer are assumed to be \( T_0 \), the minimum work input and the maximum work output are attained if the compression profile \( \xi_c \) and expansion profile \( \xi_e \) are isothermal at \( T_0 \). We define this energy as the storage energy:

\[
E_s = P_0V_0 \ln(r) \tag{1}
\]

The compression/expansion efficiencies with profiles \( \xi_c \) and \( \xi_e \) are defined corresponding as:

\[
\eta_c(\xi_c) = \frac{E_s}{W_{in}(\xi_c)}; \quad \eta_e(\xi_e) = \frac{W_{out}(\xi_e)}{E_s} \tag{2}
\]

Storage power density and output power density can in turn be defined as:

\[
P_{\text{store}} = \frac{E_s}{V_0 \xi_c}; \quad P_{\text{out}} = \frac{W_{\text{out}}}{V_0 \xi_e} \tag{3}
\]

where \( t_c \) and \( t_e \) are the times for the compression or expansion processes and \( V_0 \) is the expanded air volume. These are governed by heat transfer since the compression/expansion times are:

\[
t_{c/e} = \int dt = -\int \frac{mc_{v}dT + pdV}{hA(T - T_0)} \tag{4}
\]

where \( m \) is the mass air, \( c_v \) is the specific heat, and \( hA \) is the instantaneous product of the heat transfer coefficient and the heat transfer surface area.

There is a fundamental trade-off between efficiency and power density since fast processes (required for high power) tend to increase \( W_{in} \) or decrease \( W_{out} \). For example, isothermal processes are 100% efficient but have no power as it takes infinite time.

A Pareto optimal trade-off problem is therefore to:

Find the optimal compression (or expansion) profile \( \xi_c \) (or \( \xi_e \)) such that for a given pressure ratio \( r \) and efficiency \( \eta_c(\xi_c) \), the power density \( P_{\text{store}} \) (or \( P_{\text{out}} \)) is maximized.

![Pressure-volume diagram illustrating work input and work output during a compression process \( \xi_c \) and an expansion process \( \xi_e \).](image)

Figure 3: Pressure-volume diagram illustrating work input and work output during a compression process \( \xi_c \) and an expansion process \( \xi_e \).

Solutions to the optimal trade-off problem are given in [6, 7, 8]. In [6] and [7], analytical solutions are provided for the case where “hA” – the product of the heat transfer coefficient and the heat transfer area, is a constant or air-volume varying. In [8], numerical solutions are provided where friction losses in the liquid piston and flow limitations are also considered. In all these cases, the optimal trade-offs consist of fast-slow-fast segments. For example, when \( hA \) is a constant, the
optimal solution consists of an initial adiabatic process, followed by an isothermal process, and a final adiabatic process. The adiabatic segments are executed as fast as possible, and the isothermal process is executed at constant power. Choice of the isothermal temperature or power determines the target efficiency. Figure 4 illustrates that by merely optimizing the trajectory, the efficiency-power density trade-off can be improved dramatically. For example, at 90% efficiency, the optimized trajectory increases the power density by 500%-1500% over ad-hoc sinusoidal and linear trajectories.

Figure 4: Efficiency-power trade-offs with optimized, adiabatic- isothermal-adiabatic (AIA), sinusoidal and linear trajectories for a compression ratio of r=350 and volume varying hA(V). Taken from [7].

2.3 Challenges of optimized trajectories

To illustrate the challenges for realizing the optimal trajectories, a sample optimal compression trajectory, as shown in Figure 5, is used. It is obtained for a 5 to 200 bar (r=40) 2 stage liquid piston compression. The compressor is filled with porous media and has an air chamber volume of 3 liters. The compression time is 1.35s, thermodynamic efficiency is 90.4%, storage power is 5kW and the storage power density is 1.66MW/m³.

For this trajectory, the flow rate is 11 liters/sec at the beginning, decreases to 0.36 liters/sec and reaches 11 liters/sec again for a brief moment at the end (Figure 5 - top). The mean flow rate (time averaged) for the process is only 2.2 liters/sec. Thus, the liquid piston pump/motor must be sized 5 times larger than the average flow. Moreover, the minimum displacement is only 0.033 of the maximum, and for over 40% of the time, the displacement is less than 0.1. The turndown ratio, i.e. the ratio between the maximum and minimum displacements is 11/0.36 = 31 which is very large. Moreover, the time average pressure is only 41bar while the maximum pressure is 200 bar (Figure 5 - bottom). Thus, the high-pressure capability of the pump/motor, which is often obtained at the expense of increased friction and bulkiness, is only needed for a small fraction of time.

The problems with this situation are that:
1. Cost, physical size and weight of the pump/motors increase with maximum displacement.
2. Variable displacement pump/motors are typically more efficient when operating near its maximum displacement.

Efficiency drops off significantly at low displacements. Here the min. displacement is extremely low (0.033).

What is required is a method that allows the pump/motor to be small, operate at higher mean pressure, while achieving the optimal compression/expansion profiles.

Figure 5: Optimal compression flow rate (top) and the resulted pressure and temperature profiles (bottom).

3 Combined solid piston-liquid piston concept

To solve this problem, we propose to combine a solid piston and a liquid piston, with the solid piston actuated via a hydraulic intensifier. This will allow the optimal profile to be achieved while allowing the liquid pump/motor to be smaller and to operate at higher displacement ratio (to maximum) and at higher mean pressure.

Since the initial high flow, low pressure portion of the compression is ill suited for the liquid piston pump/motor, this phase is achieved by a solid piston (or a bellow) instead. To retain the liquid piston’s benefits of using porous media to augment heat transfer and sealing, the solid piston can be submerged in a liquid column. The movement of the solid piston in turn results movement of the liquid column. This configuration is shown in Fig. 6. When the pressure and flow requirements are suitable for the liquid piston pump/motor, liquid can be injected into or withdrawn from the chamber directly as in a regular liquid piston compressor/expander.

The solid piston can also be configured such that its movement changes the air chamber volume directly in order to avoid the need for extra liquid in (and volume of) the compression/expansion chamber. A disadvantage of this is that porous media cannot easily be placed in the volume that would be occupied by the solid piston.

The solid piston can be actuated by various means, such as mechanically, via a cam shaft, linkage, etc., or hydraulically. Figure 6 shows the case when it is actuated hydraulically, via a flow intensifier, by the same pump/motor that feeds the liquid piston. The intensifier is configured so that the piston...
area in the chamber is $A_2$ and the piston area on the side connected to the pump/motor is $A_1$. Typically, $A_2 > A_1$. The 3-way hydraulic valve switches the pump/motor flow to either the liquid piston or the solid piston.

![Diagram of liquid piston pump/motor](image1)

Figure 6: Combined solid/liquid piston compressor with the solid piston actuated hydraulically by the same liquid piston pump/motor via a flow intensifier

The two modes of operation of the combined solid piston-liquid piston compressor/expander is shown in Fig 7. The compression process starts with the solid piston being actuated by the pump/motor via the flow intensifier (Fig. 7-top). Since $A_2 > A_1$, to achieve the rate of volume change as determined by the desired compression profile, say $Q_1(t)$, the pump/motor needs only supply the smaller flow of $Q_2 = \frac{A_1}{A_2} Q_1$. Also, if the chamber pressure is $P_2$, the pump/motor sees the pressure of $P_1 = \frac{A_2}{A_1} P_2$, which is larger than $P_2$.

As compression progresses, both the chamber pressure $P_2$ and pump/motor pressure $P_1$ increase. At some point, the valve is switched so that the liquid piston is actuated and the solid piston is locked. One possible transition point is when the pump/motor pressure $P_1$ reaches some maximum desired operating pressure of the pump/motor. Since the pump/motor is expected to feed the liquid piston, this is likely the final compression pressure of the air to avoid over design. Other transition points are also possible. After the transition, the pump/motor sends all the flow $Q_1(t)$ as demanded by the desired compression trajectory to the liquid piston (Fig. 7-bottom). In this mode, the pump/motor pressure and the compression chamber pressure are the same. The compression process continues using the liquid piston until the final compression ratio is achieved.

To summarize, the compression process is divided into two phases: In the first phase (solid piston and flow intensifier), the pump/motor sees a reduced flow rate and an increased pressure (both by a factor of $A_2/A_1$); in the second phase (liquid piston phase), the pump/motor experiences the same pressure and flow rate as the original optimal trajectory for the compressed air. The pump/motor size can be reduced because of the lower flow rate requirement. The hydraulic pump/motor can also operate at higher displacement and more efficiently during the entire compression cycle.

One remaining issue is the final portion of the optimal trajectory that requires a high flow rate and high pressure (see Figure 5 top). To achieve this with the liquid piston, the pump/motor needs a high flow as before. To achieve this with the solid piston/intensifier and a downsized pump, the pump must be capable of a much higher pressure. Neither option is attractive. Fortunately, this final compression portion involves such a small volume change that even if the flow rate is reduced to the maximum pump flow rate in the first (solid piston) phase, the compression time is increased negligibly and efficiency is not affected. Therefore, by slightly modifying the final phase of the optimal profile, the pump/motor size can be reduced with little effect on the thermodynamic efficiency-power tradeoff.

![Diagram of solid piston pump/motor](image2)

Figure 7: Compression with the solid piston (top), and with the liquid piston (bottom).

4 Results

4.1 Effects on pump flow and mean pressure

In this section, we illustrate how the proposed approach affects the average flow rate, average pressure and maximum flow rate of the pump/motor. The $r=40$, compression time=1.35sec. scenario in Figure 5 is used. The transition from the solid piston phase to the liquid piston phase can take place at any moment before the pump pressure reaches the pressure limit of the pump/motor. To avoid over-design, a
natural limit is the maximum compressed air pressure. The transition instant is also a function of the intensifier area ratio $A_2/A_1$. As an illustration, Figure 8 shows the flow and pressure trajectories if the transition pressure is 150 bar and the intensifier ratio is $A_2/A_1=4.5$. Note that the solid piston/intensifier decreases the maximum flow requirement to $11/4.5 = 2.44$ liter/sec and raises the mean operating pressure of the liquid pump/motor.

Other choices of $A_2/A_1$ and transition pressures can also be used. Figure 9 shows how these choices affect the maximum and average flow rate of the liquid pump/motor. Notice that for each intensifier ratio $A_2/A_1$, both the maximum and average flow rates are minimized when the transition pressure is at the highest allowable value. Choosing this pressure (200bar) as the transition pressure (at the pump side), both the average and maximum flow rates are minimized when the intensifier ratio $A_2/A_1$ is 7.

![Figure 8: A sample flow and pressure time history of the pump using the solid piston/intensifier approach (A2/A1=4.5 and transition pressure of 150 bar)](image)

At this optimal intensifier ratio, the maximum flow rate is reduced from 11 lit/sec to 1.57 lit/sec so that the pump/motor size can be reduced by 85%. The time average pump flow rate is also reduced from 2.2 lit/sec to 0.8 lit/sec. Note that the maximum flow rate is even less than the mean flow rate without the solid piston/intensifier. The ratio between the maximum and minimum displacements (the turndown ratio) is decreased from 31 to 4.4. This would allow the pump/motor to operate much more efficiently even during the low flow rate portion of the cycle.

![Figure 9: Time average (left) and maximum (right) pump/motor flow rate with solid piston/intensifier](image)

![Figure 10: Time average pump/motor pressure with solid piston/intensifier](image)

4.2 Effect on pump/motor efficiency

To evaluate the effect on the pump/motor efficiency, a case study with an intensifier area ratio of $A_2/A_1=7$ and a transition pressure of 200 bar is run. The mechanical and volumetric efficiency maps for a typical variable displacement axial piston pump/motor as shown in Figure 11 are used in this study. A constant speed of 1800 rpm is assumed as dictated by the open accumulator system architecture in Figure 1. Note that volumetric efficiency increases monotonically with displacement, and peak mechanical efficiencies occur at high displacements (above 80%). Hence, in general, the pump/motor is more efficient when operating at high displacement. Note also that mechanical efficiency increases as pressure increases while volumetric efficiency drops with higher pressure due to larger leakages. So, it would be desirable to shift the process to a higher displacement-higher pressure area in order to improve the total efficiency of the hydraulic pump.

Figure 12 shows the displacement versus pressure history of the liquid pump/motor over the compression process while performing the optimal trajectory. When not using the solid piston/intensifier, a majority of the process is performed in the poor efficiency region. The overall pump efficiency is only 27%. When the solid piston/intensifier with an area ratio of 7 is used, the pump operates in more efficient regions. In this case, the overall pump efficiency is improved to 65%.
6 Conclusions

A potential issue with a liquid piston compressor/expander is the need for a relatively large liquid pump/motor that operates at relatively low mean pressure. This is exacerbated if optimal compression/expansion trajectories are used to optimize the thermodynamic efficiency-power trade-off. This causes the pump/motor to operate at low displacement and low efficiency regimes for significant portion of the cycle while requiring large flows at initial and final instants. By allowing a part of the process to be carried out using a solid piston and an intensifier as proposed, the pump/motor size can be significantly reduced and the pump/motor can be allowed to operate more efficiently at higher displacement ratios and higher mean pressures. The case study considered in this paper shows that both the pump/motor size and turndown ratio are reduced by 7 times, mean pressure is increased by 2 times, and efficiency is increased by 2.4 times.

In this paper, a thermodynamically optimal compression/expansion trajectory is selected a priori and the liquid piston flow must achieve that trajectory. Further improvement of the overall system can be obtained if the operational losses of the pump/motor are optimized together with the compression/expansion process. This will generate an optimal combination of pump/motor size, compression/expansion trajectory, intensifier ratio and transition pressure that optimizes the overall system efficiency.

While the description of the solid piston/intensifier approach is motivated from the high flow/low pressure portion of the optimal compression/expansion trajectory, the approach can be used to trade-off pressure for flow requirements in general. For example, a solid piston/intensifier can be used to reduce the pump/motor's pressure requirement in the high pressure, low flow region of the compression/expansion trajectory.

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