AIR COMPRESSION PERFORMANCE IMPROVEMENT VIA TRAJECTORY OPTIMIZATION - EXPERIMENTAL VALIDATION

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ABSTRACT

In an isothermal compressed air energy storage (CAES) system, it is critical that the high pressure air compressor/expander is both efficient and power dense. The fundamental trade-off between efficiency and power density is due to limitation in heat transfer capacity during the compression/expansion process. In our previous works, optimization of the compression/expansion trajectory has been proposed as a means to mitigate this trade-off. Analysis and simulations have shown that the use of optimized trajectory can increase power density significantly (2-3 fold) over ad-hoc linear or sinusoidal trajectories without sacrificing efficiency especially for high pressure ratios. This paper presents the first experimental validation of this approach in high pressure (7bar to 200bar) compression. Experiments are performed on an instrumented liquid piston compressor. Correlations for the heat transfer coefficient were obtained empirically from a set of CFD simulations under different conditions. Dynamic programming approach is used to calculate the optimal compression trajectories by minimizing the compression time for a range of desired compression efficiencies. These compression profiles (as function of compression time) are then tracked in a liquid piston air compressor testbed using a combination of feed-forward and feedback control strategy. Compared to ad-hoc constant flow rate trajectories, the optimal trajectories double the power density at 80% efficiency or improve the thermal efficiency by 5% over a range of power densities.

1 INTRODUCTION

Compressed air energy storage is a potential solution for mitigating the variable and unpredictable nature of renewable energies like wind or solar. Fig. 1 shows the Open Accumulator Isothermal Compressed Air Energy Storage (OA-ICAES) system introduced in [1] and [2] for storing excess energy for wind turbine. A key component of this system is the high pressure air compressor/expander unit which is responsible for the transformation between mechanical work and stored energy in the form of compressed air. As such it must be efficient as well as powerful enough to handle the power requirement.

For a system without any thermal storage (except the environment at the ambient temperature), the most efficient process is the isothermal compression/expansion process at the ambient temperature. However, an ideal isothermal process takes infinite amount of time and hence absorbs/produces no power since
power is work divided by time. This is because the heat transfer with the environment becomes vanishingly small as the temperature differential with the heat source/sink approaches zero. As process time decreases, power increases but the process deviates more and more from the isothermal process leading to reduced efficiency. This illustrates the trade-off between efficiency and power density where power density refers to the power normalized by the compressor/expander volume.

This efficiency-power density trade-off is mediated by heat transfer so that increasing heat transfer capability per unit volume will improve the trade-off. One approach is to inject tiny water droplets during the compression/expansion process [3,4] since the droplets present large surface area and heat capacity for heat transfer. This is especially useful for the low pressure stage compressor/expander. A second approach is to use a liquid piston compressor/expander that is filled with porous media. Since liquid can flow through the porous media, the liquid piston can compress the air above it while the porous media increases heat transfer surface area and heat capacitance. Analysis and experiment have shown that use of porous media with 70-80% porosity can increase the power density by an order of magnitude without sacrificing efficiency [5–7]. The liquid piston approach is especially attractive for the high pressure stage since the liquid also forms an effective seal for the air being compressed and serves to eliminate residual dead volume.

Yet another approach is to optimize and control the rate of compression/expansion. Optimizing the compression/expansion trajectory allows the process to better match the heat transfer capability. Analytical and numerical studies have shown that use of optimal compression/expansion trajectories can significantly increase power density (by 2 to 3 fold for high pressure) over ad-hoc linear or sinusoidal trajectories [8–11] for both simple and complex heat transfer models. However, experimental validation of the efficacy of this approach has only been done for low pressure (1bar to 10bar) [12] where the benefit is relatively minor. Since the benefit of optimal trajectory is more important for high pressure, the goal of this paper is to experimentally validate this concept in high pressure (7bar to 200bar) operation.

The rest of the paper is organized as follows. Section 2 presents the experimental setup. The heat transfer coefficient correlation obtained empirically from extensive CFD experiments is presented in Section 3. Calibration of the critical volume measurement is presented in Section 4. Design and control of the optimal trajectories are given in Section 5. Experimental results are given in Section 6. Concluding remarks are given in Section 7.

2 Experimental Setup

The schematic and picture of the liquid piston air compressor experiment setup are shown in Figs. 2 and 3. The setup was designed to study the compression/expansion processes during single shot experiments. In this system, a double-acting hydraulic cylinder (4) is coupled with a single-acting water cylinder (5) so that extension of the hydraulic piston will cause the water piston to be retracted and vice versa. The hydraulic cylinder is connected to a hydraulic power supply (at 200bar) via a solenoid-actuated servo valve. This valve is used to control the oil flow rate to the hydraulic cylinder and to regulate its extension speed at a desired value. A magnetic incremental encoder is connected to the tandem rod (between the hydraulic cylinder and water cylinder) in order to measure the displacement of water piston, which will be used to calculate the volume of water displaced into the compression chamber. The compression chamber is a vertical cylinder made of stainless steel and is connected to the water cylinder via a combination of hoses and ball valves. Retracting the water piston causes water to be pushed into the compression chamber and raises the water column level inside it. This will compress the air inside the compression chamber. A pressure transducer is located at the top of compression chamber to measure the air pressure during compression process. A transparent plastic side tube is used to estimate the initial level of liquid column in the compression chamber and calculate the initial air volume in it. By knowing the amount of water that is displaced into the compression chamber (from water cylinder), it would be possible to estimate the air volume inside the compression chamber during the compression process. A combination of ball valves and a single poppet valve (mounted on top of the compression chamber) are used to control the filling of the chamber with fresh air. While the liquid piston air compressor is considered for compressing air from 7bar to 200bar, a conventional solid-piston air compressor is used to compress air from ambient pressure to 7bar. By opening the poppet valve, the compression chamber is filled with fresh air at 7bar provided by the solid-piston air compressor. After the chamber is filled
with air, the poppet valve closes and the system becomes ready for compression process. By regulating the flow rate through the hydraulic servo-valve, it would be possible to control the extension speed of hydraulic piston which in turn defines the retraction speed of water piston and consequently the water flow rate into the compression chamber. Therefore, a previously defined flow rate (as a function of compression time) can be tracked using an appropriate closed-loop controller. More details and information regarding this experimental facility can be found in [6] where the same setup was used to study the effect of porous media. In this paper, the effect of optimal trajectories will be studied without using porous media.

3 Heat Transfer Modeling
Computing the optimal compression trajectory is sensitive to the model used for heat transfer prediction between air under
compression and compression chamber walls. Either underesti-
dating or overestimating the heat transfer between air and heat
exchanger material (in our case, the chamber’s walls since porous
media is not used) results in a wrong optimal compression pro-
file, which in turn reduces the improvement of power density (for
a fixed thermal efficiency). Therefore, the first step in calculating
the optimal compression profile is to find a reasonably accurate
heat transfer model for the chamber.

Assuming lumped properties for air (i.e. zero-dimensional
temperature and pressure), the heat transfer between air and its
surrounding environment (described by \( Q \)) can be written as:

\[
Q(t) = hA(T_{\text{air}} - T_{\text{wall}})
\]

where \( h \) is the convective heat transfer coefficient, \( A \) is the avail-
able heat transfer area, \( T_{\text{air}} \) is the air temperature and \( T_{\text{wall}} \) is the
wall temperature that is assumed to remain constant during the
compression process (\( T_{\text{wall}} = 295 K \)). While calculating the total
heat transfer area is easy (since it’s only a function of air vol-
ume at any time), evaluating heat transfer coefficient is relatively
complex since it is an instantaneous function of air properties,
piston speed and chamber geometry. To find this dependency, a
series of numerical simulations is performed in COMSOL Multi-
physics software to investigate the correlation between convec-
tive heat transfer coefficient and air properties, piston speed and
chamber geometry.

While there are many parameters that affect the heat trans-
fer coefficient, a comprehensive study is performed by changing
some parameters while keeping the rest of them constant, in or-
der to study their effect on heat transfer coefficient. It should be
emphasized that such a flexibility is only available in numerical
analysis since the experimental investigation for revealing the de-
pendency of heat transfer to different parameters is very difficult
and time consuming. According to the comprehensive numeri-
cal analysis that is done in COMSOL, a correlation between \( h \),
aspect ratio of air column \( L/D \) (ratio between length of air col-
umn \( L \) and its diameter \( D \)), piston speed \( U \), heat conductivity \( k \),
density \( \rho \) and viscosity \( \mu \) of air is suggested as:

\[
Y = c_1X^2 + c_2X + c_3
\]

where \( X \) and \( Y \) are:

\[
X = \frac{\rho}{\mu} U^a \left( \frac{\mu}{k} \right)^b
\]

\[
Y = \frac{h}{k} \left( \frac{L}{D} \right)^d
\]

To find the best combination for \( a, b \) and \( d \), an optimization
problem is defined and solved. Here, we are looking for the best
set of parameters that results in minimum difference between the
numerical value of \( h \) (shown in Fig. 4-top) and the value calcu-
lated by the suggested correlation defined by Eqs. (2), (3) and
(4). Therefore, the optimization problem is formulated as:

\[
\{a^*, b^*, d^*\} = \min_{a,b,d} \| h_{\text{COMSOL}} - \hat{h}(k,\rho,\mu,U,L,a,b,d) \|_2
\]

where \( \hat{h} \) is the heat transfer coefficient according to the cor-
relation, and calculated as:

\[
\hat{h} = k \left( \frac{D}{L} \right)^d \times (c_1X^2 + c_2X + c_3)
\]

All the data points found from the numerical simulation (in
COMSOL) are used here to find the best set of parameters. The best
combination is found as: \( a = 0.35851 \), \( b = 0.88792 \) and
\( d = 0.35404 \). The comparison between numerical \( h \) and the cor-
relation (using the optimal coefficients) is shown in Fig. 4.

\[
\begin{align*}
Y &= 3.39210^{-5}X^2 + 1.5628X + 1025.4175 \\
\end{align*}
\]

\[
\begin{align*}
(\frac{L}{D})^d &\times (\frac{c_1X^2}{\rho})^{0.35851} \times (\frac{U}{k})^{0.88792}
\end{align*}
\]
4 Estimation of Air Volume in the Compression Chamber

Direct measurement of the air volume in the compression chamber is not available in the experimental setup. Instead, air volume is estimated from the initial air volume and the change in water volume. Because water is slightly compressible and the components such as hoses expand, the change in water volume in the chamber consists of the volume of water injected and the volume change due to pressure. Thus, the air volume in the compression chamber can be expressed as:

\[ V(t) = V_0 - V_{\text{Displaced}}(t) + V_C(P) \]  

where \( V_0 \) is the initial volume of air in the chamber at the beginning of compression (i.e. \( t = 0 \)), \( V_C \) is the pressure dependent volume adjustment due to water compressibility and system expansion, and \( V_{\text{Displaced}} \) is the volume of water pushed into the compression chamber from the water cylinder. It is important to consider \( V_C \) since it can account for 30% of the volume at the end of compression.

The pressure dependent volume adjustment term \( V_C \) is obtained by filling the chamber completely with water and compressing it. The result is shown in Fig. 5. \( V_C \) has a larger slope at lower pressures which is due to soft components such as hoses and entrained air in the water. At higher pressures (> 10bar), it has a constant slope which is slightly greater than that due pure water compressibility.

The displaced water volume term \( V_{\text{Displaced}} \) should ideally be proportional to the movement of the water hydraulic cylinder as reflected by the linear magnetic encoder measurement \( C_i \). In order to account for any slight nonlinearity, a quadratic relation is used. Eq. (7) becomes:

\[ V(t) = V_0 - K_1C_i - K_2C_i^2 + V_C(P) \]  

where \( V_0, K_1, K_2 \) are the constant parameters calibrated for each experiment. To do this, at the end of each experiment, the air in the chamber is allowed to return to ambient temperature at successive volumes (the liquid piston is withdrawn in each step). A sample pressure trace is shown in Fig. 6. Notice the step decreases in pressures at the end of the experiment. Assuming ideal gas behavior (the same approach can be done with real gas model), air volume and pressure after the air has returned to ambient temperature (i.e. at the end of each step) must satisfy:

\[ T_1 = T_2 = \ldots = T_n \]  

\[ P_1 V_1 = P_2 V_2 = \ldots = P_m V_m \]

where \( V_i, i = 1, \ldots, m \) can be expressed using (8). The coefficients \( V_0, K_1, K_2 \) are then optimized to minimize the relative error in (10), specifically,

\[ \{V_0^*, K_1^*, K_2^*\} = \min_{V_0, K_1, K_2} \text{VAR} \left( P_i(V_0 - K_1C_i - K_2C_i^2 + V_C(P_i)) \right) \]  

\[ (11) \]
where $VAR$ denotes the variance of the $m$ air pressure and volume products. The approach described above is used for each compression test since the initial air volume for each run can be slightly different than the other tests. For the sample case shown in Fig. 6 these parameters are found as follows:

\[
V_0 = 2.292 \times 10^{-3} \text{m}^3 \\
K_1 = 1.196 \times 10^{-7} \text{m}^3/\text{count}, \\
K_2 = 2.316 \times 10^{-14} \text{m}^3/\text{count}^2
\]

5 Design and Implementation of the Optimal Compression Trajectories

The heat transfer correlation found based on COMSOL simulations is used to calculate a series of optimal compression trajectories for the given chamber geometry and desired initial and final pressures. The optimization problem is formulated such that the compression time is the cost function while the compression efficiency is an equality constraint that needs to be satisfied. The flow rate must also be below the pressure dependent flow capability of the system. Dynamic Programming (DP) approach is then used to solve the optimal control problem [11].

A combined feedback and feedforward controller is used to track the optimal flow trajectory in the compression chamber. According to (8), the air volume rate of change can be calculated as:

\[
\dot{V} = -F(t) = -K_1 \dot{C} - 2K_2 \dot{C} \dot{\dot{C}} + \frac{dV_C}{dP_r}. \tag{12}
\]

An open loop calibration test is first performed on the system (in terms of different voltages on hydraulic servo-valve) to evaluate the required command signal for a given flow rate at a given pressure. This map is found as shown in Fig. 7. By inverting the results, it would be possible to find the required servo valve voltage for a desired piston speed at a given pressure. This map is used in the feedforward controller. The feedback part of the controller is simply a PI controller on air volume error (difference between the actual air volume and the desired air volume calculated by time integral of optimal flow rate). The controller block diagram used for this experiment is shown in Fig. 8.

6 Experimental Results

The experimental results of applying the optimal trajectories are shown in Fig. 9. The optimal compression profile starts with the maximum available flow rate ($Q_{max} = 800 \text{cc/s}$), which is followed by a much lower flow that continues for nearly the rest of the compression process. A short fast compression concludes the process and achieves the final desired pressure (200bar) at the end. Such fast-slow-fast trajectories are consistent with optimal trajectories from our previous studies [8–11].
Higher Efficiency

FIGURE 9: Optimal compression flow rate results; Top: optimal flow rate versus time ratio ($t/t_{end}$ where $t_{end}$ is the total compression time); Middle: air pressure versus air volume ratio ($V/V_0$ where $V_0$ is the initial air volume); Bottom: air pressure versus compression time

note that the integration for $E$ in Eq. (14) is taken over an isothermal compression process\(^1\) which starts at $(P_0, V_0)$ and ends at $(V_{iso}^f, P_{iso}^f)$ (see [2,11] for more details). The compression power density is defined as the ratio between the storage power and the total volume of compression chamber:

$$\text{Power Density} = PD = \frac{E}{t_{end}V_0}$$

(16)

Compression efficiency and power density for each test are calculated based on Eqs. (13), (14), (15) and (16). Results in terms of efficiency versus compression time and efficiency versus power density are shown in Fig. 11. In general, the optimal compression flow rate results in a smaller compression time, therefore a larger storage power density for the same thermal efficiency. According to the experimental results, for the given chamber geometry and initial and final pressures, this improvement can be as high as 100% for thermal efficiencies around 80% (from 55kW/m\(^3\) to 120kW/m\(^3\)). Hence, a compression chamber that uses constant flow rate to compress air can be downsized to its half size and maintains its performance (compression power and thermal efficiency) if it uses the optimal compression rate to compress air. The performance improvement can be also interpreted as a higher thermal efficiency for the same storage power

\(^1\)Real gas model is used instead of ideal gas model for better accuracy at high pressures [13]
density. According to the results, this raise in thermal efficiency can be as high as 5% for storage powers around 100kW/m$^3$ (from 75% to 80%).

![Graph showing thermal efficiency vs. storage power density](image)

**FIGURE 11**: Comparison between the optimal flow rate and non-optimal (constant) flow rate compression; Top: thermal efficiency versus compression time; Bottom: thermal efficiency versus storage power density.

7 Conclusions

Previous theoretical and numerical studies have shown that applying optimal compression/expansion trajectory is an effective approach to improve the performance of an air compressor/expander machine by optimizing the trade-off between efficiency and power density. It is also known that an accurate heat transfer model for the compression/expansion chamber is critical in order to design optimal flow profiles and improve the system performance. In this work, a systematic approach was used to find a correlation that models the convective heat transfer coefficient between air and compression chamber wall. This correlation is found by numerical simulation performed in COMSOL. This correlation is then used to calculate the optimal compression trajectories that minimize compression time for a given (desired) compression efficiency. Dynamic programming approach was applied to determine a family of optimal compression flow profiles. The optimal performance of the system is then compared with non-optimal performance that is generated by using ad-hoc compression trajectories (here constant flow rate compression). According to the results, a 5% thermal efficiency improvement is achievable at 100kW/m$^3$ storage power density. Likewise, the storage power can be doubled at 80% efficiency if the constant flow rate is replaced by the corresponding optimal compression trajectory.

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