Experimental Study of Heat Transfer Enhancement in a Liquid Piston Compressor/Expander Using Porous Media Inserts

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Abstract

The efficiency and power density of gas compression and expansion are strongly dependent on heat transfer during the process. Since porous media inserts can significantly increase heat transfer surface area, their addition to a liquid piston compressor/expander has been hypothesized to reduce the time to complete the compression or expansion process and hence the power density for a given thermodynamic efficiency; or to increase the thermodynamic efficiency at a fixed power density. This paper presents an experimental investigation on heat transfer with porous inserts during compression for a pressure ratio of 10 and during expansion for a pressure ratio of 6. A baseline case without inserts and five cases with different porous inserts are tested in a compression experiment: 3 interrupted ABS inserts with plate spacing of 2.5, 5, and 10 millimeters and 2 aluminum foam inserts sized with 10 and 40 pores per inch. The 2.5mm and 5mm interrupted plate inserts were also tested in expansion experiments. Porous inserts are found, in compression, to increase power-density by 39 fold at 95% efficiency and to increase efficiency by 18% at 100kW/m$^3$ power density; in expansion, power density is increased three fold at 89% efficiency, and efficiency is increased by 7 % at 150kW/m$^3$. Surface area increase is found to be the predominant cause in the improvement in performance. Thus, a liquid piston compressor/expander together with a porous medium may be used in applications requiring high compression ratios, high efficiencies, and high power density such as in an open-accumulator Compressed Air Energy Storage (CAES) system or a compressor for compressed natural gas (CNG).

Highlights

- Porous medias role to improve compressor/expander performance shown experimentally
- Significant increase in efficiency increase (up to 18%) at fixed power density
- Significant increase in power density (up to 39 folds) at fixed efficiency
- Surface area is the predominant contribution to improvements

Keywords: Liquid piston, porous media, gas compression / expansion, compressed air energy storage (CAES), efficiency, power density
1. Introduction

Compressed air is a potential cost effective, power-dense, reliable and scalable means for storing energy at the utility scale compared to electric batteries, pumped-hydro and closed hydraulic accumulators [1–4]. In a conventional compressed air energy storage (CAES) system, compressed air is produced using excess electricity and stored in underground caverns. It is then combined with natural gas in a gas turbine to boost combustion efficiency. As a storage device, its efficiency is typically less than 50% [5]. In recent years, isothermal CAES have been proposed both by industry and academia as an energy storage approach that does not require fuel [6, 7]. Here, mechanical work is consumed in the compressor to compress air during the storage phase, and recuperated in the expander as the compressed air expands during the regeneration phase.

For isothermal CAES to be viable, a high pressure (e.g. 200–300bar), powerful and efficient compressor/expander is critical. Such a compressor is also needed for compressing natural gas as fuel for vehicles. Conventionally a high pressure gas compressor consists of multiple stages with inter-cooling. This creates a zig-zag pressure-volume curve consisting of successive adiabatic and constant volume cooling segments. A compressor with multiple adiabatic stages and ideal inter-cooling is the most efficient if all stages have the same compression ratio [8]. As the number of stages increases, compression approaches the most efficient isothermal compression. Whereas efficiency of the compressor/expander is governed by the pressure-volume (P-V) curve, the time it takes to trace the curve and hence the power (work/time) depends on the heat transfer rate in the compressor/expander or in the intercooling. If less time is allowed for heat transfer, the P-V curve deviates more and more from isothermal and becomes less and less efficient. Hence, there is an inherent trade-off between efficiency and power of a compressor/expander.

This trade-off can be mitigated by optimizing the compression and expansion trajectories, or by improving the heat transfer capability. In [9–11], trajectories are optimized for three cases: when \( h \cdot A \) (the product between the heat transfer coefficient, \( h \), and the heat transfer surface area, \( A \)) is a constant, when \( h \cdot A \) is volume dependent, and when \( h \cdot A \) is an arbitrary function. By matching power and availability of heat transfer capability, several-fold improvement in power densities over ad-hoc (sinusoidal or linear compression/expansion) profiles at the same efficiency can be achieved. Optimal volume-time trajectories at low pressure ratios (1bar to 7bar) have been validated experimentally in [12, 13]. The effectiveness of using tuned compression trajectories is also shown in simulation in [14].

Heat transfer capability can be enhanced by actively spraying very small liquid droplets with high heat capacity and large total heat transfer surface area [15–18]. The approach pursued in this paper is the use of a liquid piston in a compression/expansion chamber filled with porous material, as proposed in [19]. The liquid piston freely flows through the porous material which greatly increases the heat transfer area. Compared with a rigid piston, a liquid piston also forms an effective low friction seal for the air being compressed. In addition to the CAES application, liquid piston is also being pursued as a cost-effective and efficient means to compress natural gas for vehicle use [20], as well as for hydrogen [21] and supercritical \( CO_2 \) [22]. Numerical simulations of fluid flow and heat transfer were conducted for various geometries (tiny tubes, metal foams and interrupted-plate heat exchangers) in [19, 23–25]. Preliminary experimental studies of enhanced heat transfer with the use of inserts are reported in [26]. However, since the instrumentation was limited, only limited qualitative inferences could be made. Recently, a liquid piston compressor was also demonstrated in a benchtop ocean compressed air energy storage (OCAES) system [27].

The objective of this paper is to test experimentally the effectiveness of various porous media to increase the efficiency or power density of a liquid piston compression/expansion process. A series of compression and expansion experiments using different types of porous inserts in a low-pressure (12bar maximum pressure) compression/expansion chamber are presented. The porous inserts under study are ABS plastic interrupted-plates with 2.5mm, 5mm and 10mm spacing between plates and aluminum foams with 10 and 40 pores per inch (PPI). These were chosen to span a range of surface area augmentation types, to utilize commercially available materials (foams), and based on demonstrated advantages from previous computational studies (interrupted plates) [24]. The input/output work, heat transfer and change in internal energy of the air, and compression/expansion efficiencies are calculated during compression/expansion for cases with porous inserts and compared with a benchmark case without porous inserts. This is the first experimental study of a compressor/expander with heat transfer inserts and a liquid piston that has been conducted with precise documentation
of the volume, pressure and bulk temperature of the air during compression and expansion. The results show, indeed, that porous media inserts can significantly improve the trade off between efficiency and power density in a liquid piston compressor/expander.

The remainder of the paper is structured as follows. Section 2 provides an overview of the liquid piston compressor/expander and the governing equations that describe it. Section 3 presents the test facilities and describes the porous materials that are used. Section 4 describes the testing, data analysis and uncertainty analysis. Section 5 presents the results for the compression and expansion tests. Sections 6 and 7 contain discussion and conclusions.

2. Liquid Piston Compression/Expansion Processes

In a liquid piston air compressor/expander, air enters and exits a compression/expansion chamber from the top, and liquid (water in our case) from the bottom. The volume of the liquid in the chamber is controlled by a hydraulic pump/motor. In compression mode, the compression/expansion chamber is initially filled with air at ambient pressure. As liquid is pumped into the chamber, the volume of the air above it is reduced and the pressure is increased. In expansion mode, the chamber is initially filled with mostly liquid and a small volume of compressed air above the liquid. As liquid is allowed to exit the chamber, the volume of the compressed air increases and pressure is reduced. As such, the liquid column acts as a piston and mechanical work is applied and extracted via the hydraulic pump/motor for the liquid piston. The compression/expansion cycles can be repeated (frequently, if high power is desired) with the intake of ambient air or the exhaust of expanded air. With the liquid piston, the chamber can be filled with a porous medium since both liquid and air can flow through it. This can significantly increase the surface area for heat transfer for a given amount of initial/expanded air volume.

In this paper, air is treated as an ideal gas since pressures are relatively low (1 bar to 12 bar compression) and temperatures are modest (200 K to 450 K). This assumption allows a mass-average bulk temperature to be derived from a measured pressure and volume. An additional assumption is that the chamber and water lines are rigid and the water is incompressible. This permits the volume of air in the chamber to be determined by the difference between the initial air volume and the amount of liquid introduced.

Let $P_0$ and $T_0$ be the ambient pressure and temperature. We assume that the compressed air is to be stored in a tank at the constant pressure of $rP_0$ ($r$ is the compression ratio) and that there is sufficient dwell time for the compressed air in the storage vessel to cool down to $T_0$ before reusing. The work input for the compression process and work output from the expansion process are illustrated as the horizontally-shaded areas under the P-V curves in Fig. 1. The work input to compress $V_0$ volume of ambient air at $(P_0, T_0)$ to be stored in the tank consists of the compression work to $rP_0$ along a certain compression trajectory $(\xi_c)$ defined by the P-V (or T-V) profile, and the isobaric ejection of the compressed air into the tank:

$$W_{in}(\xi_c) = (rP_0 - P_0)V_c - \int_{V_0}^{V_c} (P - P_0)dV$$  \hspace{1cm} (1)

where the second term and $V_c$ are dependent on $\xi_c$.

Similarly, the work output from expanding a volume $V_c$ of compressed air from $(rP_0, T_0)$ to $P_0$ consists of the isobaric injection of the compressed air to the expansion chamber and the expansion work to $P_0$ along an expansion trajectory $(\xi_e)$ defined by the P-V (or T-V) profile:

$$W_{out}(\xi_e) = (rP_0 - P_0)V_c + \int_{V_c}^{V_0} (P - P_0)dV$$  \hspace{1cm} (2)
where \( V_e \) is the volume at which the expanding air reaches ambient pressure, and \( V_c \) and the second term are dependent on \( \xi_e \).

For a compressed air volume of \( V_e = V_0/r \) at pressure \( rP_0 \), we define the stored energy as the maximum work that can be extracted given an ambient environmental temperature of \( T_0 \). This can be obtained either from \( W_{in} \) or \( W_{out} \) when the compression/expansion process is isothermal at \( T_0 \). The stored energy \( E_P \) is given by:

\[
E_P = (rP_0 - P_0)V_e + \int_{V_c}^{V_0} (P - P_0)dV = P_0V_0\ln(r) \tag{3}
\]

Thus, we can define the compression and expansion efficiencies respectively as:

\[
\eta_c := \frac{E_P}{W_{in}} \tag{4}
\]
\[
\eta_e := \frac{W_{out}}{E_P} \tag{5}
\]

In addition to efficiency, power density is also an important figure of merit for a compressor/expander. The storage power of a compressor is defined as the ratio of the storage energy \( (E_P) \) to the compression time \( t_c \). The output power of an expander is defined as the ratio of the output work \( (W_{out}) \) to the expansion time \( t_e \). The compression/expansion power densities are the respective powers normalized by the compressor/expander volume, which consists of the expanded air volume \( V_0 \) and the solid volume of the porous insert \( V_{ins} \):

\[
\rho_c := \frac{E_P}{t_c(V_0 + V_{ins})} \tag{6}
\]
\[
\rho_e := \frac{W_{out}}{t_e(V_0 + V_{ins})} \tag{7}
\]

A compressor/expander with a larger power density is advantageous as smaller or fewer units are needed for a given power requirement, so that system is more compact and the capital cost is lower.

The temperature dynamics of a fixed mass \( (m) \) of air is given by the first law of thermodynamics:

\[
mc_v \frac{dT(t)}{dt} = -P(t)\dot{V}(t) - \dot{Q}(t) \tag{8}
\]

where \( c_v \) is the constant volume specific heat capacity, \( V(t) \) is the instantaneous air volume, and \( \dot{Q}(t) \) is the heat transfer rate out of the air (to the environment assumed to be at \( T_0 \)). To achieve the isothermal condition, it is necessary that the heat transfer equals input power:

\[
\dot{Q}(t) = -P(t)\dot{V}(t) = \text{Power}_{input}(t) \tag{9}
\]

Assuming Newton’s law of cooling:

\[
\dot{Q} = hA(T - T_0) \tag{10}
\]

where \( hA \) is the product of the heat transfer coefficient and surface area, then we have

\[
mc_v dT = -(PdV - hA(T - T_0)dt) \tag{11}
\]

Rearranging, we have

\[
\frac{-mc_v dT + PdV}{(T - T_0)} = hA \cdot dt \tag{12}
\]

Equation (12) shows that if \( hA \) can be increased uniformly at each air volume \( V \) such that \( hA \rightarrow ahA \), then the solution to (12) will be scaled as \( P(t) \rightarrow P(at) \), \( T(t) \rightarrow T(at) \) and \( V(t) \rightarrow V(at) \). Hence, the same P-V and T-V profiles can be accomplished faster by a factor of \( a \), i.e. \( t_{ce} \rightarrow t_{ce}/a \). This is achieved by scaling the liquid piston flow rate by \( (\alpha (-V(t) \rightarrow -\alpha V(at)) \).

With the decrease in cycle times, \( t_c \) or \( t_e \), from (6)-(7), the power densities will also be increased by a factor of \( \alpha \).

This is the basis for our hypothesis that by introducing porous media which increase the heat transfer surface area \( A \), the compression/expansion times will be reduced and the power densities will be increased while maintaining the same efficiency. Similarly, at the same power-density, it is expected that increasing \( A \) will increase efficiency. The objective of this paper is to test this hypothesis experimentally.

3. Experimental Setup

Figures 2 and 3 show a schematic and picture of the experimental setup. The setup is designed to allow the liquid flow in and out of the compression/expansion chamber to be controlled and measured. Flow rate control permits the compression/expansion trajectories to be controlled so that operation over a range of power densities can be explored. Instantaneous air volume is calculated by subtracting/adding the amount of water that has entered/exited the chamber from the initial air volume. Air pressure is measured directly with a pressure transducer. With air volume and pressure, bulk air temperature can be calculated from the ideal gas law. This avoids the use of thermocouples that can measure only local temperature and because of the porous media, are restricted in their placements. Compression / expansion works \( (W_{in} \) and \( W_{out} \) can also be calculated from air volume and pressure trajectories.

The compression/expansion chamber (1) is a 353mm long, 50.8mm internal diameter transparent polycarbonate cylinder. It can accommodate various porous media inserts. The empty volume of the chamber is 715
1) Liquid Piston C/E  
2) Tank  
3) Pump  
4) Upstream Pressure Transducer  
5) Relief Valve  
6) Solenoid Control Valve  
7) Turbine Flowmeter  
8) Inlet/Outlet Ball Valves  
9) Needle Valve  
10) Load Cell  
11) C/E Chamber Pressure Transducer  
12) Charge Air Ball Valve  
13) Regulated Compressed Air Supply

Figure 2: Schematic of the liquid piston compression/expansion experimental setup

Figure 3: Picture of the compression/expansion chamber

cc. The aspect ratio and volume were determined based on commercially available sizes and flow capabilities of the pump/valve to reach reasonable power densities (related to compression/expansion times). Graduations on the tube permit initial and final water levels to be read. Aluminum end caps at the top and bottom of the compression/expansion chamber serve as manifolds for making various connections. The top cap holds a Kulite ETM-375 pressure transducer (11) for air pressure measurement and a port (12) for charging the chamber with compressed air prior to expansion experiments. The bottom cap provides ports for liquid to enter and exit the chamber. Elastomeric O-rings are used at the end caps for sealing and tie rods are used to hold the cylinder against the two end caps.

When operating in compression mode, a Wanner Hydra-cell positive displacement diaphragm pump (3) supplies water to the circuit. The pump is capable of supplying a maximum flow rate of 275 cc/s. A relief valve (5) is used to divert a portion of this flow back to tank. This fraction is set by controlling the pressure upstream of the solenoid control valve (6). When that pressure is held constant, the flow through the relief valve to the tank and the flow into the circuit are constant. A proportional-integral controller is used to regulate this upstream pressure, measured with pressure sensor (4), to a specified level. When operating in compression mode, the flow rate is adjustable and held constant for a given experiment.

When operating in expansion mode, flow control is more limited. A needle valve (9) is placed downstream of the compressor/expander so that the flow rate is determined by the pressure drop across the valve and the orifice area. By adjusting the orifice area from run to run, the average flow rate during expansion is adjusted, but the flow rate is not constant.

An advantage of conducting experiments with constant flow rates (as in compression) is that between any two experiments, the air volume trajectories \( V_1(t) \) and \( V_2(t) \) are related by a time scaling, i.e. there exists a scalar \( \alpha \) such that

\[
V_1(t) = V_2(\alpha t), \quad \dot{V}_1(t) = \alpha \dot{V}_2(\alpha t)
\]

As was noted earlier, the solution in Eq.(12) has this time scaling property if the scaling \( hA \rightarrow \alpha hA \) is applied uniformly. The orifice determined flow profiles (as in expansion), however, do not enjoy this property so that the scaling analysis is not as exact. For example, the air volume trajectories \( V_1(t) \) and \( V_2(t) \) for a 6 sec and 3 sec expansion times are not related by \( V_1(t) = 2V_2(2t) \).

An Omega FTB-1412 turbine flowmeter (7), with a measurement range of 46.7 cc/s to 466.7 cc/s, and a load...
cell (10), which measures the weight of water that has left the expansion chamber, are used to measure the flow entering or exiting the compression/expansion chamber. At high flow rates, the turbine meter is used (the load cell measurement is inaccurate in this case due to fluid inertial force); at low flow rates, the load cell is used (the flow rates are out of range of the turbine meter).

The compression/expansion chamber can be safely pressurized up to 12 bar. For compression tests, air is compressed from ambient pressure to a pressure ratio of 10. For expansion tests, compressed air at approximately 12 bar is expanded to a final pressure of approximately 2 bar. A pressure ratio of 6 is used for all expansion tests. Thus, $P_0 = 1$ bar for compression, and $P_0 = 2$ bar for expansion. This setup is capable of compression rates from 46.7 cc/s to 275 cc/s and (peak) expansion rates up to 275 cc/s.

The compression/expansion chamber (1) can accommodate various porous media inserts. Five different porous inserts are considered in these tests: two metal foams and three interrupted-plate styles, as shown in Figure 4. Table 1 compares some of the insert properties. The metal foams used in these studies are open-cell aluminum foams with 40 pores per inch (PPI) and 10 PPI (Duocel foams from ERG Aerospace Corp. Oakland, CA). These foams have a nominal porosity (i.e. the proportion of void volume to total volume) of 93%. The interrupted-plates are modeled after the design proposed in [24] and are fabricated in ABS plastic using a 3D printing process. The interrupted plates disrupt and restart the boundary layers so that heat transfer is enhanced. Their open structures enable low liquid flow drag and low tendency to trap water. The three interrupted-plate designs have a nominal plate thickness of 0.8 mm and a nominal plate length of 7.5 mm. The inter-plate spacings of the three inserts are 2.5 mm, 5.0 mm and 10.0 mm. Figure 5 compares the total surface area found in the compressor/expander when the piston is at bottom dead center. Since the inserts are to be placed in the chamber uniformly, except for minor effects of the end cap areas, the different inserts would have the effect of scaling the heat transfer surface areas nearly uniformly.

4. Methods

4.1. Testing Methods

To test the effect of porous media inserts on the efficiency and power density in liquid piston compression and expansion, the following tests were performed. The ambient and initial temperature are at $T_0 \approx 300K$.
For compression, constant flow rate compression experiments from 1 bar pressure and $T_0$ temperature to 10 bar were conducted at various compression times and with different inserts. For expansion, expansion experiments from 12 bar and $T_0$ to 2 bar are conducted for a range of expansion times and with different inserts. Since expansion flow rates cannot be actively controlled, expansion times are varied by adjusting the downstream orifice for each run. The orifice opening area is constant during each run, however.

Baseline tests with no porous media inserts were conducted first. For compression, 16 experimental runs at different flow rates, and hence power densities, were performed. Ambient pressure and temperature were used as initial conditions. The initial air volume was typically 715 cc. Exceptions were the shortest time, highest power density cases when a smaller initial volume (410 cc) was used. This is due to limited pump flow capability. Effects of differences in initial volume are factored out by considering power density instead of power. The data taken when the pressure is above 10 bar were discarded. Power densities ranging from 3 kW/m$^3$ to 180 kW/m$^3$ were tested. For expansion, 6 experimental runs at different expansion times were performed by adjusting the downstream orifice for each run. Initial pressure of 12 bar, ambient temperature and initial air volume of $\approx$ 100 cc were used in all cases. The data taken when the pressure is below 2 bar were discarded. Power densities ranging from 29 kW/m$^3$ to 223 kW/m$^3$ were tested.

Testing with porous media inserts proceeded similarly as for the baseline cases and over a similar range of power densities. For compression, all five porous media inserts shown in Figure 4 and Table 1 were tested. The entire chamber was filled with porous media. Accounting for the insert volumes, the initial air volumes were 600-670 cc, except for the highest power density cases, which were 350-460cc. For expansion, only the interrupted-plate inserts with 2.5 mm and 5.0 mm plate spacing were tested. Also, a small gap was left at the top of the chamber so that the initial air volume could be read from the graduations on the chamber. Initial air volumes were $\approx$ 100 cc. Only data taken at pressure below 10 bar (for compression) and above 2 bar (for expansion) are used for data processing.

Further details of the experimental setup and data processing can be found in [28].

### 4.2. Data Reduction Methods

Liquid flow rates and pressures are obtained using the Matlab/xPC Target Real-Time System (Mathworks, MA) and a 16-bit data acquisition card at a sampling rate of 1 kHz. Air volume is calculated from:

$$V(t) = V_0 - \int_0^t \text{Flow}(\tau) d\tau$$  \hspace{1cm} (13)

where $V_0$ is the initial air volume and $\text{Flow}(\tau)$ is the liquid flow rate. Bulk air temperature is obtained from the ideal gas law:

$$\frac{T(t)}{T_0} = \frac{P(t) V(t)}{P_0 V_0}$$  \hspace{1cm} (14)

The work input, $W_{in}$, work output, $W_{out}$, efficiencies, $\eta_c$ and $\eta_e$ and power-densities, $\rho_c$ and $\rho_e$, are obtained from numerical integration of the P-V trajectory, and from Eqs.(1)-(7).

### 4.3. Uncertainty Analysis

The uncertainties of the measured variables are summarized in Table 2. Pressure, volume and temperature are essential in the calculation of efficiency and power. Of these, pressure is known with the most certainty, as it is measured directly. Standard error propagation analysis shows that air volume and temperature are known with somewhat less certainty. For example, the worst case pressure error is at the initial (ambient) pressure (1.7%, ±0.017bar) and the worst case volume error is at the end of compression (6.1%, ±4.3cc).

---

**Table 1: Detailed Properties of Porous inserts.** Cell size corresponds to hydraulic diameter. Specific area is the ratio of surface area to volume. Data for the metal foams are provided by the manufacturer (ERG Aerospace, Oakland CA) and data for the interrupted plates are computed from CAD models.

<table>
<thead>
<tr>
<th>Media</th>
<th>Insert Type</th>
<th>Material</th>
<th>Cell Size (mm)</th>
<th>Specific Area (m$^{-1}$)</th>
<th>Porosity (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>40PPI</td>
<td>Metal Foam</td>
<td>Aluminum</td>
<td>0.5</td>
<td>1732</td>
<td>92</td>
</tr>
<tr>
<td>10PPI</td>
<td>Metal Foam</td>
<td>Aluminum</td>
<td>2.0</td>
<td>709</td>
<td>93</td>
</tr>
<tr>
<td>2.5mm</td>
<td>Interrupted-Plate</td>
<td>ABS</td>
<td>2.5</td>
<td>655</td>
<td>81</td>
</tr>
<tr>
<td>5 mm</td>
<td>Interrupted-Plate</td>
<td>ABS</td>
<td>5.0</td>
<td>377</td>
<td>88</td>
</tr>
<tr>
<td>10 mm</td>
<td>Interrupted-Plate</td>
<td>ABS</td>
<td>10.0</td>
<td>203</td>
<td>86</td>
</tr>
<tr>
<td>baseline</td>
<td>no-insert</td>
<td>Polycarbonate</td>
<td>50.8</td>
<td>78.7</td>
<td>100</td>
</tr>
</tbody>
</table>
Table 2: Uncertainty in Experimental Measurements

<table>
<thead>
<tr>
<th>Measurement</th>
<th>Instrument</th>
<th>Measuring Range</th>
<th>Measurement Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure</td>
<td>ETM-375</td>
<td>0-17 bar</td>
<td>±0.1% of Full Scale</td>
</tr>
<tr>
<td>Flow</td>
<td>FTB-1412</td>
<td>140-467 cc/s</td>
<td>±0.6% of Reading</td>
</tr>
<tr>
<td>Flow</td>
<td>FTB-1412</td>
<td>47-140 cc/s</td>
<td>±1.0% of Reading</td>
</tr>
<tr>
<td>Flow</td>
<td>Load Cell</td>
<td>0-100 g/s</td>
<td>±0.5% of Reading at Steady-State</td>
</tr>
<tr>
<td>Initial Air Volume</td>
<td>Graduated Cylinder</td>
<td>N/A</td>
<td>± 2 cm³ (Baseline)</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>± 2.8 cc (with Inserts)</td>
</tr>
</tbody>
</table>

Table 3: Uncertainty in Efficiency for Theoretical Compression and Expansion Trajectories

<table>
<thead>
<tr>
<th></th>
<th>Baseline</th>
<th>2.5 mm</th>
<th>5.0 mm</th>
<th>10.0 mm</th>
<th>10 PPI</th>
<th>40 PPI</th>
</tr>
</thead>
<tbody>
<tr>
<td>Error in Isothermal Compression Efficiency</td>
<td>±1.3%</td>
<td>±1.7%</td>
<td>±1.6%</td>
<td>±1.6%</td>
<td>±1.6%</td>
<td>±1.6%</td>
</tr>
<tr>
<td>Error in Adiabatic Compression Efficiency</td>
<td>±0.9%</td>
<td>±1.0%</td>
<td>±0.9%</td>
<td>±0.9%</td>
<td>±0.9%</td>
<td>±0.9%</td>
</tr>
<tr>
<td>Error in Isothermal Expansion Efficiency</td>
<td>±1.2%</td>
<td>±1.2%</td>
<td>±1.2%</td>
<td>N/A</td>
<td>N/A</td>
<td>N/A</td>
</tr>
<tr>
<td>Error in Adiabatic Expansion Efficiency</td>
<td>±0.4%</td>
<td>±0.4%</td>
<td>±0.4%</td>
<td>N/A</td>
<td>N/A</td>
<td>N/A</td>
</tr>
</tbody>
</table>

To estimate the impact of measurement uncertainties on the estimate of efficiencies, adiabatic and isothermal compression and expansion processes are considered. These represent the slowest and fastest cases with efficiencies that can be calculated analytically, and thus present good bounds on the possible uncertainties. By perturbing these trajectories by the estimated errors in pressure, volume and temperature, uncertainties in efficiency estimates are obtained. Results are summarized in Table 3. Uncertainty values in efficiency are highest when the process is close to isothermal (when flow rate is low), but uncertainty values are roughly the same for the different inserts.

5. Results

5.1. Typical Flow Profiles

The typical flow profiles for compression and expansion are shown in Fig. 6. As expected, the flow rates are nearly constant for the compression tests, as maintained by active control. For expansion, because a fixed orifice is used for each test, the flow rate decreases as pressure decreases. Ideally, the maximum flow should occur at the beginning. However, because the on/off valve takes some time to fully open, the flow rate increases continuously from zero to a maximum flow instead.

5.2. Pressure-volume and Temperature-volume Profiles

5.2.1. Compression P-V / T-V

The pressure-volume and temperature-volume profiles for the baseline compression tests with different compression times are shown in Figs. 7-8. As compression time decreases, the P-V and T-V profiles tend...
Figure 7: Pressure-volume profiles for baseline compression cases without inserts at various compression times. Isothermal and adiabatic profiles are added for comparison.

Figure 8: Temperature-volume profiles for baseline compression cases without inserts at various compression times.

Figure 9: Pressure-volume profiles for compression cases with different inserts at similar compression times of 2 s.

Figure 10: Temperature-volume profiles for compression cases with different inserts at similar compression times of 2 s.

from being close to the isothermal profiles towards the adiabatic profiles.

Typical effects of different inserts on the pressure-volume and temperature-volume profiles for the compression tests over similar times are shown in Figs. 9-10. Curves with other compression times have similar trends. These show that the introduction of inserts shifts the profiles towards the isothermal profiles. It is apparent that the 10mm and 5mm interrupted-plates are the least effective inserts.

At compression time of 2 sec., inserts reduce peak temperature by 90-120K (57%-75% of the temperature rise in the baseline empty cylinder case). Comparison between Figs. 8 and 10 shows that to limit temperature rise by $T_{0}/10$ without any inserts, a compression time of 60sec. is needed. However, with the use of inserts (40PPI, 10PPI foams, and 2.5mm interrupted-plates), compression times can be reduced 30 times to 2 sec., hence significantly improving compactness. While the general trend of the T-V profiles is that temperature increases monotonically as volume decreases, at small volume there are some small deviations. These deviations are likely due to the slight drop in flow at the end of the compression cycle (for the no-insert 2s case in Fig. 8). The increase in heat transfer surface area to volume ratio near the end of compression, since the cap and liquid piston surface areas are unchanged when the volumes are small, and the effect of water evaporating as temperature increases are also possible reasons. These effects become more apparent when input work is relatively small (low compression rates or good heat transfer).

It is interesting to note from Figs. 7 and 9 that the P-V curves for all compression times, with and without inserts, are close to straight lines on a log-log scale. This indicates that compression processes with constant flow rates follow, nearly, a polytropic process:

$$P(t)V^\lambda(t) = \text{constant}$$

where the $-\lambda$ is the slope. For isothermal and adiabatic
processes, \( \lambda = 1 \) and the ratio of the specific heats. Since input work \( W_{in} \) and efficiency \( \eta \) become functions only of the polytropic index, this simplification allows us to compare, according to (12), different compression processes with different compression times and different inserts cases that have the same efficiencies.

5.2.2. Expansion P-V/T-V

The pressure-volume and temperature-volume profiles for the expansion tests are shown in Figs. 11-14. In all cases, as evidenced in both the P-V and T-V curves, the initial portion of expansion is close to the adiabatic process, whereas the latter portion tends toward the isothermal process. This shape is a consequence of flow rates decreasing as pressure decreases (Fig. 6-bottom). For such flow profiles, expansion work is large in the beginning and much smaller towards the latter portion of the expansion. From (8) and (9), high heat transfer rates are necessary to maintain the air temperature, initially. It is easier to do so towards the end of expansion due to the lower power and larger exposed surface area for heat transfer.
Figures 11-14 also show that for slower expansions (longer expansion times) and with the introduction of inserts, transition to the isothermal process occurs at smaller volumes. It is also apparent that the 2.5mm interrupted-plate insert is more effective than the 5mm interrupted-plate inserts. Comparison between Figs. 12 and 14 shows that to limit the minimum temperature to \(0.85 \cdot T_0\) without any inserts, an expansion time of 8 sec. is needed. With the use of inserts, expansion times can be reduced more than 3 times to less than 2.5 sec.

Note that for the expansion process with the orifice determined flow profiles, the P-V and T-V data do not fit a polytropic model, as in the expansion case with constant flow profiles.

5.3. Efficiencies and power densities

Efficiencies and power densities are calculated according to Eqs. (4)-(5) and are plotted against each other in Figs. 15 and 16 for the compression and expansion tests, respectively. Error bars (estimated from Section 4.3) indicate levels of uncertainties.

Figures 15-16 show that the insert cases are more power dense at the same efficiency, and more efficient at the same power density, than the baseline (no insert) cases. Specifically, for a compression efficiency of 95%, power density is increased by a factor of 10 to 37 (with 10mm plates and 40 PPI foam) at the same efficiency. Similarly, for a fixed expansion efficiency of 89%, expansion power density is increased by a factor of 3 with the use of inserts. The inserts can also be seen as improving efficiency for a given power density.

At the power density of 100 kW/m\(^3\), the compression efficiency was increased from 78% to 96% with the addition of porous media (40 PPI Al foam). At a power density of 150 kW/m\(^3\), the expansion efficiency was increased from 83% to 90% with the addition of porous media (2.5 mm interrupted-plates). These improvements are expected because the inserts provide more surface area for heat transfer. The order of decreasing effectiveness of the inserts to improve efficiency/power density in compression are: 1) 40PPI-foam, 2) 10PPI-foam or 2.5mm plates, 3) 5mm plates, 4) 10mm plates, and 6) no-inserts. For expansion, the order is: 1) 2.5mm plates, 2) 5mm plates, 3) no-inserts. These orders correlate exactly with the surface areas provided by these inserts in Fig. 5.

5.4. Efficiencies and power densities normalized by specific surface area

To further investigate the role of surface area increase in performance improvement, efficiencies are plotted
against power densities normalized by the volume specific surface area (i.e. ratio of surface area to total volume) for the compression and expansion cases in Figs. 17 and 18, respectively. This normalized power density is also the mean power per surface area. By using the total volume to define the specific surface area, the negative effect of the inserts occupying more volume (which can affect power density by 8-20%, depending on porosity) is also factored out. While the areas of the top cap and the liquid piston surface are included, their contributions are relatively small (except when the volume has become small). The difference in specific surface areas between compression and expansion cases is due to the gap between the top cap and the inserts in the expansion cases.

In the case of compression, Fig. 17 shows that after surface area is accounted for, the cases with porous inserts collapse to approximately the same curve. Particularly, at 95% efficiency, the normalized power densities lie between 0.1 $kW/m^2$ and 0.16 $kW/m^2$, within the range of data scatter. On the other hand, the baseline no-insert case is noticeably lower at 0.045$kW/m^2$. The difference is even more pronounced at lower efficiencies.

In the expansion case, Fig. 18 shows that the efficiency vs. normalized power density curves for all cases (including the no-insert cases) also collapse to approximately the same curve. However, the difference between insert and no-insert cases does not seem to be significant.

Because the compression processes (with the constant flow rates) are nearly polytropic, each experiment with the same efficiency will have similar P-V / T-V curves. From (12), this normalization allows other factors affecting heat transfer beyond surface area, such as heat transfer coefficient, $h$, and heat exchanger temperature, $T_0$, to be compared. Because the expansion processes (with orifice flow control) are not polytropic, the comparison is more approximate, as the same efficiency does not necessarily imply the same P-V / T-V curves, and hence, the same LHS of (12).

These results suggest that surface area is indeed the most important factor for increasing efficiency and power density. The fact that the normalized curves collapse to a single curve implies that all the inserts have similar effects, if any, on factors other than area that may influence heat transfer. As the no-insert case has a lower normalized power density than the insert cases at the same efficiency, the insert cases show greater positive influence on these other factors than no-insert case in compression. A possible explanation is that the porous media provide many small features to interrupt the flow.
and encourage new boundary layers to develop. The small hydraulic diameters in the porous media also induce higher local velocities. Both can cause heat transfer coefficients, $h$, to increase. Another possibility is that the heat transfer surfaces may have heated more in the no-insert case than in the insert case, thus increasing the $T_0$ on the LHS of (12) and decreasing the temperature gradient.

6. Discussion

The results described above show clearly that porous media can significantly improve the trade-off between efficiency and power-density of a liquid piston compressor/expander. The experiments also show that the improvement can be explained predominantly by the increase in heat transfer surface area introduced by the porous media. However, the pressure ratios in these tests are relatively low compared to the applications of compressed air energy storage or compressed natural gas for which pressures of 200-300 bar are expected. Thus, future experiments must consider higher pressure ratios.

The use of families of constant flow rates (for compression) and orifice flows (for expansion) in the experiments simplifies the experimental conditions to better isolate the effects of the porous media within the flow control constraint of the experimental setup. As shown in [9–11] analytically, and [12, 13] experimentally, an optimized flow trajectory can significantly improve performance of compression/expansion processes. Judicious choice of trajectories can also mitigate the effect of liquid pressure drop while maximizing overall performance [11]. An optimized profile typically consists of segments of high flows at the beginning and at the end of the process, and a slow (nearly isothermal) portion in the middle. In this regard, the orifice flow profiles in expansion resemble optimized profiles more so than the constant-flow profiles in compression (see Fig. 6). This may explain partially why the baseline efficiency in expansion is higher than the efficiency in compression at similar power densities, and that the subsequent improvement due to adding porous media is less dramatic in expansion than in compression. Further experiments are needed to clarify the combined benefits of porous media inserts and trajectory optimization, however, since the experiments in [12, 13] were performed without inserts.

While the experiments in this paper highlight the heat transfer benefits, especially the increased surface area provided by the porous media inserts, other aspects of the porous media need to be considered when choosing a porous medium. In [19], long narrow cylinders were analyzed for liquid piston applications. Preliminary experiments in [26] have shown that while these provide good heat transfer, differences in drag inside and between the cylinders induce unwelcome water jets. The open cell foams and interrupted-plates in this study avoid this issue. The metal foams were selected in this study for their isotropic properties and high porosity. The interrupted-plates were selected because of their interrupted boundary layer features, which offered promising flow features for heat exchangers and lower drag.

Porous media materials also have different thermal conductivities and thermal capacitances. The thermal conductivity of the Aluminium foam (167W/m-K) is 1000 times higher than that of the ABS plastic interrupted plates (0.19W/m-K). Results in this paper suggest that at the relatively low pressure ratios of 6-10, these differences in conductivities and capacitances do not have significant effects. For example, the ABS 2.5mm interrupted plates and the Al 10 PPI foam have similar surface areas (table 1) but very different thermal conductivities. Taking porosity into account, the heat capacitance per unit air volume of the ABS plates is 2.3 times that of the the Al foam. Yet, their performances in terms of efficiency/power density trade-off are not discernable from the data scatter in Figs. 15 and 17. To understand why performance is insensitive to thermal conductivity, consider the (worst) cases with the fastest flow rates (2sec). Using the heat transfer correlations for metal foam investigated in [25], and for the interrupted plates in [24], it is found that the Biot number is of the order of $10^{-4}$ for the Al foam and ranges from 0.03 to 0.2 for the ABS interrupted plates. Thus, the Al foam can reasonably be considered as a lumped thermal mass but the validity of this assumption is not as clear for the interrupted plates. To further investigate this issue, zero-dimensional, two-energy-equations (both the air and the porous medium exposed to air are considered as lumped masses) simulations are performed for the 2.5mm interrupted plate case using a slightly modified heat transfer correlation proposed in [24] and the volume profiles from the experiment. The predicted temperature profile matches the experiment well with the predicted efficiency (92.98 %) matches that of the experiment ($\approx 93\%$). The predicted rise in porous medium temperature is only 0.45K. When the plates are assumed

\[ Nu = 2 + 0.0876 \cdot Re^{0.792} \cdot Pr^{0.17} \]

The correlation used is $Nu = 2 + 0.0876 \cdot Re^{0.792} \cdot Pr^{0.17}$. The bias term in [24] is 9.7 instead of 2 and theoretically should be 1. The modification fits the data much better.
to be 1/4 of their original thickness of 0.8mm so that the
Biot number would be comfortably less than 0.1, the
temperature rise becomes 1.8K. This change in temper-
Cation ratio decreases efficiency negligibly (from 92.98%
to 92.89%). These small media temperature increases
are consistent with the 2D CFD simulations for metal
foams and ABS interrupted plates in [25, 29] respect-
Cally. This analysis shows that because the thermal ca-
pacitance is sufficiently large compared to the thermal
energy and the heat transfer coefficients in these low
pressure ratio experiments, the effect of thermal con-
ductivities is insignificant. For larger pressure ratios,
for example for our 2nd stage setup (7bar to 210bar),
the effect of conductivity both in the axial and the radial
directions would be much more significant as the ther-
mal energy and the heat transfer coefficients will be at
least 1 order of magnitude higher.

Another aspect to be considered is the likelihood for
the porous media to trap water or air. This reduces the
volumetric efficiency of the compressor/expander
as it prevents the ejection of the compressed air and
the intake of fresh ambient air. In this respect, despite
their thermal characteristics, the 40PPI and 10PPI metal
foams both trap significant amount of water and pock-
et of air which would make them impractical for the
actual application. Trapped water and air pockets repres-
ent unused dead volumes that decrease the volumetric
efficiency of the compressor/expander. Air pockets that
remain in the chamber undergo repeated compression
and expansion leading to energy losses. Indeed, the rea-
son why the metals foams were not tested in the expa-
sion cases is because of the large amount of water that
remains trapped in the pores after an expansion process.
Since the amount of trapped air/water seems to be ran-
dom from case to case, it is difficult to keep uncertainty
to reasonable levels.

7. Conclusions

Experiments were conducted to test the hypothesis
that porous media can improve efficiency/power den-
sity trade-off in a liquid piston compressor/expander.
Compression performances were compared between 5
different inserts and a baseline, at a pressure ratio of
10 rising from atmospheric pressure, and power den-
sities that ranged from 3.8kW/m$^3$ to 182kW/m$^3$ using
constant flow profiles. Expansion performances were
compared with 2 inserts and a baseline, at an expan-
sion ratio of 6, starting from 12 bar, and power den-
sities that range from 29kW/m$^3$ to 223kW/m$^3$ using ori-
Cice expansion flow profiles. In compression, porous
media increase power density by as much as 39 times
at a constant efficiency; and efficiency by as much as
18% for typical constant power densities. In expan-
sion, porous media increase power density by as much
as 3 times for a fixed efficiency; and efficiency by as
much as 7% for typical power densities. The benefit of
porous media is more significant for higher power den-
sities. The improvement has been attributed mainly to
the increase in surface area from the porous media. Liq-
uid piston compressor/expander in combination with a
suitable porous medium can enable the high compres-
sion ratios, high efficiencies, and high power densities,
needed in applications such as Compressed Air Energy
Storage (CAES) system and compressors of natural gas
(CNG) for vehicles.

Experimental investigation of the improvement that
porous inserts provide at higher pressures, higher com-
pression ratios, and higher power densities is near com-
pletion (see [30] for preliminary findings). Further stud-
ies will also focus on optimized insert geometries, how
best to distribute the surface area within a compres-
sion/expansion chamber [31] as well as the combina-
tion of optimal compression/expansion trajectories with
porous media inserts. Combined effects of porous in-
serts with other efficiency enhancing measures such as
optimized compression chamber geometry and droplet
sprays are also worthwhile topics. In addition, for liquid
piston compressor/expander to be practicable, proper
ejection of compressed air and draining of the liquid pis-
ton through the medium must also be studied.

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