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Initial Design and Experimental Results of a Novel Near-Isothermal Compressor for Heat Pump Applications

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ABSTRACT

Efforts to increase the efficiency of residential and commercial air conditioners and heat pumps have demonstrated that the compressor has the highest electrical energy usage of the system. Therefore, the efficiency of this component should be improved to reduce its energy usage. Another challenge with heat pump design is that some compressor types have drawbacks that make modulation difficult. To address these challenges, we developed a near-isothermal liquid compressor at the US Department of Energy's Oak Ridge National Laboratory. The compressor uses propylene glycol to compress CO_2. In the compression chamber the propylene glycol can enter from the bottom to create a liquid piston for compression or through a spray nozzle at the top of the chamber. In the latter case, heat transfer from the gas to be compressed to the liquid droplets is high. This allows near isothermal operation of the compressor, which increases the efficiency by 17% to 30% compared to adiabatic compression. Furthermore, the isothermal liquid compressor enables very efficient and simple part load modulation. Experimental results demonstrating the operation of the liquid compressor using CO_2 are presented. Initial data demonstrated a temperature rise of 7 K at pressure ratios of almost 4. For comparison at the same initial pressure, temperature, and pressure ratio, adiabatic compression would result in a temperature rise of approximately 70 K. Data plotted on a pressure-enthalpy diagram demonstrated that the compression started at superheated state and ended in supercritical state. Testing was performed with repeated compressions in the superheated region of the pressure-enthalpy diagram at liquid flow rates of 2×10^{-3} m^3/min and 3×10^{-3} m^3/min to understand the limitations of the prototype for use with an actual heat pump system. This work demonstrated a novel cycle on a temperature-entropy diagram. Results from this work will be used to develop a second-generation prototype in which more rapid cycling is possible.

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1. INTRODUCTION

In the United States alone, residential and commercial space heating and cooling use 4.4 Quads of primary energy and are responsible for 148 Mt of CO_2 emissions annually (DOE, 2018). The compressor in a vapor compression cycle serves to increase the pressure and the saturation temperature of the refrigerant from the evaporator level to the level required in the condenser. Compression in conventional compressors is performed at high speed with very short residence time for the gas to exchange heat with its surroundings. This results in large viscous dissipation and high discharge temperatures. In vapor compression cooling devices, the discharge superheated refrigerant rejects its heat to the ambient in the condenser. Because of the low overall heat transfer coefficient of refrigerant-air heat exchangers, a higher approach temperature difference is required. However, increasing the approach temperature difference increases the compressor lift and lowers the efficiency of the compressor and the refrigeration cycle. Furthermore, conventional heat pump compressors cannot compress two-phase refrigerants because of damage caused by pitting due to wet vapors. This limitation lowers the thermodynamic limit of the maximum achievable coefficient of performance (COP) of a vapor compression cycle as it departs from the ideal Carnot limit, which is based on two phase compression.

A near-isothermal liquid piston compressor (LPC) works by compressing the gas inside a compression chamber using a rising liquid column. Spraying the liquid into the chamber provides a large surface heat exchange area. Because of the large heat transfer rate between the gas being compressed and the compression liquid, the compression process is almost entirely done at the temperature of the sprayed liquid, which remains at a constant temperature and is the source of the "near isothermal" in the condition. This approximates a mechanical piston slowly rising in a reciprocating compressor but with much lower friction losses while removing all the heat of compression. Because the moving boundaries inside the compression chamber are hydraulic boundaries, the LPC is not limited to compressing only the gaseous phase and can compress saturated vapors. The pumped liquid is cooled by ambient in a much smaller surface area heat exchanger than a refrigerant-air heat exchanger. The implications for vapor compression refrigerating systems are significant. The LPC can be designed to have the condensation take place inside the compression chamber, without the need for the refrigerant to reject heat to ambient air in the condenser. This advantage is especially beneficial to trans-critical CO_2 refrigeration systems. The CO_2 can be simultaneously compressed and cooled inside the LPC, and a gas cooler may no longer be needed.

Several studies have examined the use of sprayed droplets and similar concepts for achieving isothermal compression. Gerstmann and Hill provided an analytical study of increasing the efficiency of a refrigeration process by isothermalization of compression (Gerstmann & Hill, 1986). In that work, the refrigerant was cooled by an aerosol that was externally cooled. The aerosol then became the liquid piston that served to compress the refrigerant. Gerstmann and Hill noted that for the aerosol, the droplets must be small, as well as the surface area, to transfer the heat stored in the liquid to the working gas. They also noted that an issue to be solved before practical applications can be realized is the instabilities at the liquid and gas interface, which become important with liquid pistons.

Van de Ven and Li created a computer model of a liquid piston concept (Van de Ven & Li, 2009) to improve the efficiency of gas compression and expansion. Preliminary modeling showed an increase in the total efficiency, as defined as the maximum potential energy of the gas divided by the work of compression, from 70% to more than 83%. They found that greater efficiency could be realized with a larger quantity of small diameter compression cylinders, which improves the heat transfer while minimizing the viscous flow forces. The efficiency appeared to not have a limit, though the maximum that was modeled - 1 million compression chambers - rapidly became impractical. However, the authors noted that their modeling assumptions may be oversimplifications. Although the study did not specifically involve droplet spray heat transfer, several items were noted as necessary for successful liquid piston compression. First, a portion of the gas will diffuse in the liquid, which can reduce the bulk modulus of the liquid. This can be avoided by designing the chamber to minimize splashing of the liquid, by selecting a fluid with low gas solubility, or by using a bladder to physically separate the liquid and gas. Furthermore, the possibility of the liquid leaving the compression chamber during the gas exhaust stroke should be avoided.

The concept of liquid droplet cooling of compression has also been applied to Compressed Air Energy Storage. A multiphase thermodynamic heat transfer model was developed, and the total surface area of the droplets was found to be the most important characteristic for increasing compression efficiency (Qin & Loth, 2014; Qin *et al*., 2014). The researchers found that small droplets and high mass loading, defined as the ratio of the mass of water injected into the chamber to the mass of air already in the chamber were optimal, which agrees with the conclusions by Gerstmann and Hill (Gerstmann & Hill, 1986). Computer models and experiments to study spray cooling to improve the round-trip efficiency of a hydropneumatic energy storage system that uses an LPC and direct contact heat exchanger for waste heat recovery have also been completed (Odukomaiya *et al.*, 2016). The researchers found that both spray cooling during compression and waste heat utilization during expansion provided improved efficiency. Indicated efficiency, defined by the researchers as the work output divided by the work input and only includes the thermodynamic losses, increased from 0.90 to 0.96, whereas the electrical efficiency, which includes all loses, increased from 0.66 to 0.70.

In a study of the compression of air using a water piston (Patil *et al.*, 2020), the authors used a compression ratio of approximately 2.5 at injection pressures varying from 10 to 70 psi (69 to 483 kPa) and spray angles of 60°, 90°, and 120°. They defined isothermal efficiency to express how closely the compression process follows an isothermal trajectory. They reported a maximum isothermal efficiency of 95% at the highest injection pressure. The spray angles of 60° and 90° resulted in similar performances that were both better than the performance with the spray angle of 120°. The authors attributed the reduction in efficiency of the wide spray angle to the droplets colliding with the walls of the chamber, which made them unavailable to participate in heat transfer. Therefore, the authors specifically noted the importance of spray design.

In summary, some of the issues that the literature demonstrates with using droplet heat transfer and an LPC are instabilities at the interface of the liquid and gas that disturb the compressing fluid, gas diffusion in liquid reducing the effectiveness of compression, liquid entrainment in the gas exiting from the discharge port, selection of compression fluid to minimize reaction and dissolution between the gas and liquid, and changes in inlet pressure and spray angle that affect the efficiency of compression. In this paper, we present a preliminary experimental evaluation of a CO_2 LPC. A prototype was built and installed in an experimental facility at the US Department of Energy's Oak Ridge National Laboratory. The experimental system used propylene glycol as the compression liquid. Experiments were carried out to investigate the compression speeds obtainable with commercially available spray nozzles and the impact of compression speed on the heat transfer. The LPC, experimental facility, and findings are presented.

2. EXPERIMENTAL SETUP

Experiments were carried out using a compression cylinder of approximately 1.4×10^{-3} m^3 volume. The chamber without its cover is shown in Figure 1. The compression chamber was designed to allow the entry of CO_2 from the top, and then the entry of the compression fluid (propylene glycol) from ports in the top or the bottom. The port on the bottom is shown in an axial location on the bottom of the chamber in the foreground of the photo. The radially located ports were originally designed for level switches and level visualization. However, visual indication of the level of the propylene glycol was impractical because the hardware used for this fitting was highly prone to leakage. Therefore, one of the ports was used for a level switch, and the other two were used for temperature measurement.

Figure 1: Fabricated compression chamber

The compression chamber was then integrated into the test prototype as shown in the Piping and Instrument Diagram (P&ID) in Figure 2. The P&ID shows the components necessary to compress CO_2 using the LPC. Carbon dioxide is supplied from a cylinder. A chiller and heat exchanger allow the flowing CO_2 to be conditioned to the desired temperature. At the same time, the propylene glycol is stored in the oil reservoir. The bath temperature is controlled by a second chiller. The propylene glycol can then be pumped to the compression chamber. As noted, it can flow into the compression chamber from the bottom to supply the liquid piston, or it can be sprayed into the top of the chamber through a nozzle that creates a fine spray of droplets. These droplets then fall to the liquid piston surface while exchanging heat with the CO_2 to be compressed. At the conclusion of the process, the compressed CO_2 can be exhausted, and the propylene glycol returns to the oil reservoir.

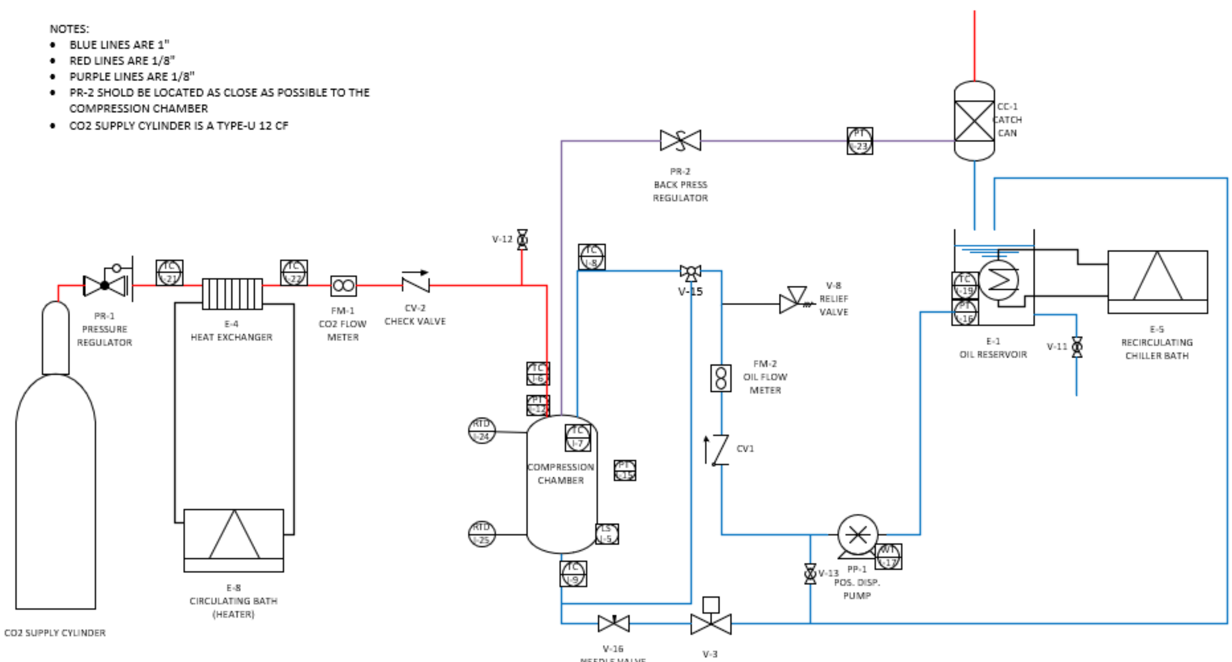

Figure 2: P&ID for testing the isothermal liquid compressor

Operation of the near isothermal LPC is demonstrated in Figure 3. It consists of four phases as follows.

- Intake: During this phase, low-pressure CO_2 is drawn into the chamber from a supply cylinder. The gas can be heated to create the appropriate gas condition. The gas flow is metered. A check valve prevents flow back to the supply cylinder from the compression chamber.
- Compression: This phase can take place in one of two modes. Liquid can be forced to the bottom of the chamber, which will move the liquid piston up and increase the pressure of the CO_2. Alternatively, for near isothermal operation, the liquid can be sprayed into the chamber through a nozzle that will produce small droplets. The droplets then fall to the surface of the LPC. The droplets absorb heat from the gas, which occurs as the pressure of the gas is increasing. The temperature of the liquid is controlled. Flow of the liquid and pump power are both measured. The level of the compression fluid is indicated by a level switch.
- High pressure gas discharge to the exhaust line: This phase occurs when the gas pressure exceeds the setpoint of the back pressure regulator. The exhaust line also includes a catch can to capture any liquid that is unintentionally captured by the discharging gas. This excess liquid then flows to the liquid storage container via gravity. The remaining liquid in the chamber is returned to the oil reservoir by opening a solenoid valve.
- Second intake: This phase occurs when the liquid level in the chamber is allowed to fall. A solenoid valve is opened to allow low pressure CO_2 to enter the chamber to begin the process again.

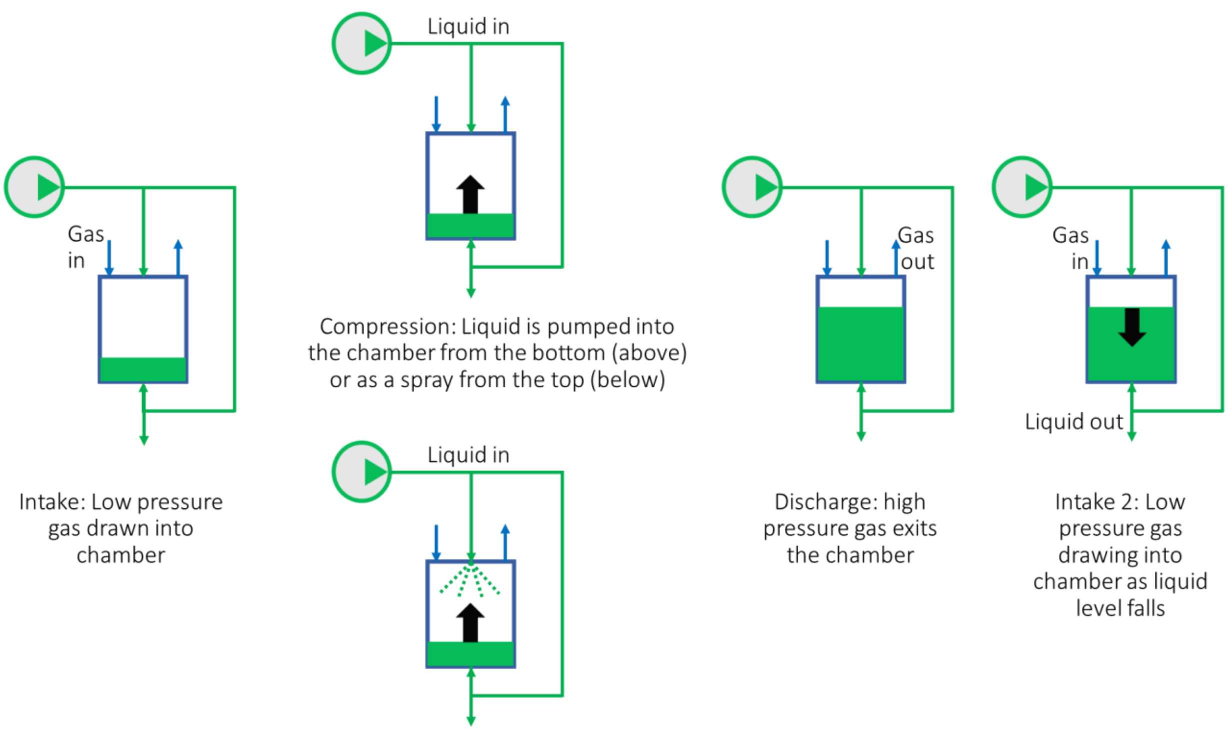

Figure 3: Operation of the near-isothermal LPC

3. RESULTS AND DISCUSSION

Initially, an experiment was performed to demonstrate the differences between pumping the compression liquid into the bottom of the chamber and spraying the compression liquid into the top of the chamber. The average CO_2 temperature and pressure and the compression liquid temperature of one cycle in each mode are shown in Figure 4. The temperature of the compressed gas at the end of the first cycle rose by approximately 7 K, and at the end of the second cycle, it rose by 2.5 K. This suggests that the heat was being removed adequately and much more significantly when compressed with a spray nozzle. Careful observation of the temperature and pressure profiles at the conclusion of the compression phase of the process demonstrates some significant differences. As noted, discharge is a two-step process. The first step occurs when the pressure in the chamber exceeds the setpoint of the back pressure regulator, which discharges most of the high-pressure CO_2. The second step occurs when the solenoid valve is manually operated to release the pressure of the compression fluid. These two steps are shown in Figure 4. The second discharge starts at approximately 80 s. with the opening of the back pressure regulator. This corresponds to the maximum pressure in the chamber. The pressure then decreases until the solenoid valve is opened to release the compression liquid at approximately 170 s. The chamber pressure then decreases rapidly and goes below the original steady state pressure of 2 MPa. Adding compression fluid back to the chamber brings the pressure back up to the steady state pressure of 2 MPa. A similar pattern can be observed for the second compression process, with the back pressure regulator opening at 690 s. and the solenoid valve opening at 790 s. The solenoid valve was only left open for approximately 10 s. after which the pressure decrease became more gradual.

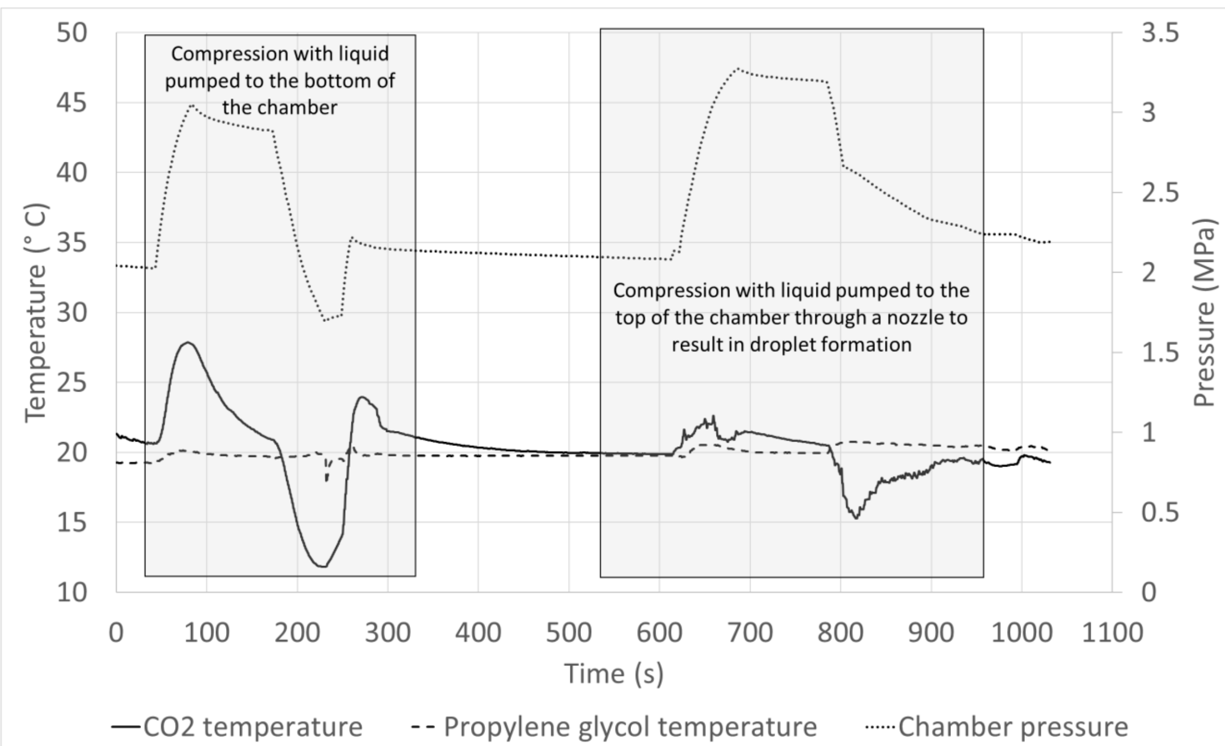

Figure 4: Propylene glycol and CO2 temperature and pressure during compression and discharge cycles of the isothermal compressor system (left) without spray and (right) with spray. One compression cycle with liquid only was followed by one compression cycle with spray cooling

The temperature trend in Figure 4 shows rapid fluctuations during the spray compression process. The fluctuations are likely caused by the sprayed droplets. With the spray nozzle angle of 170°, the spray reached the temperature sensor and cooled the sensor. The same observation was reported in (Patil *et al.*, 2020)

In further work considering applications of this technology to transcritical CO_2 refrigeration, we performed compression of CO_2 from the superheated vapor state to the supercritical state. Two improvements over the previous experiments were implemented. First, the propylene glycol was circulated through the chamber at 60°C before the compression began to warm the thermal mass of the chamber walls. Second, the CO_2 gas was heated and admitted to the chamber at a low flow rate to bring the gas to the temperature of the chamber. Thus, the temperature of the CO_2 was kept above its condensing temperature in this compression stroke. The compression process was superimposed on the pressure-enthalpy and temperature-entropy diagrams for CO_2, as shown in Figure 5. The figure shows isothermal compression taking place from the superheated vapor state to the supercritical state. The initial conditions of the CO_2 were a temperature of 28.6 °C and a pressure of 1.87 MPa. The final conditions were temperature of 38.8 °C and a pressure of 8.58 MPa. The maximum temperature of the gas was 40.2 °C, which occurred at a pressure of 7.43 MPa. The abrupt change from increasing temperature to decreasing temperature occurred when the flow rate of liquid was decreased in preparation for the conclusion of the test. The flow of liquid continued, but at a decreasing rate.

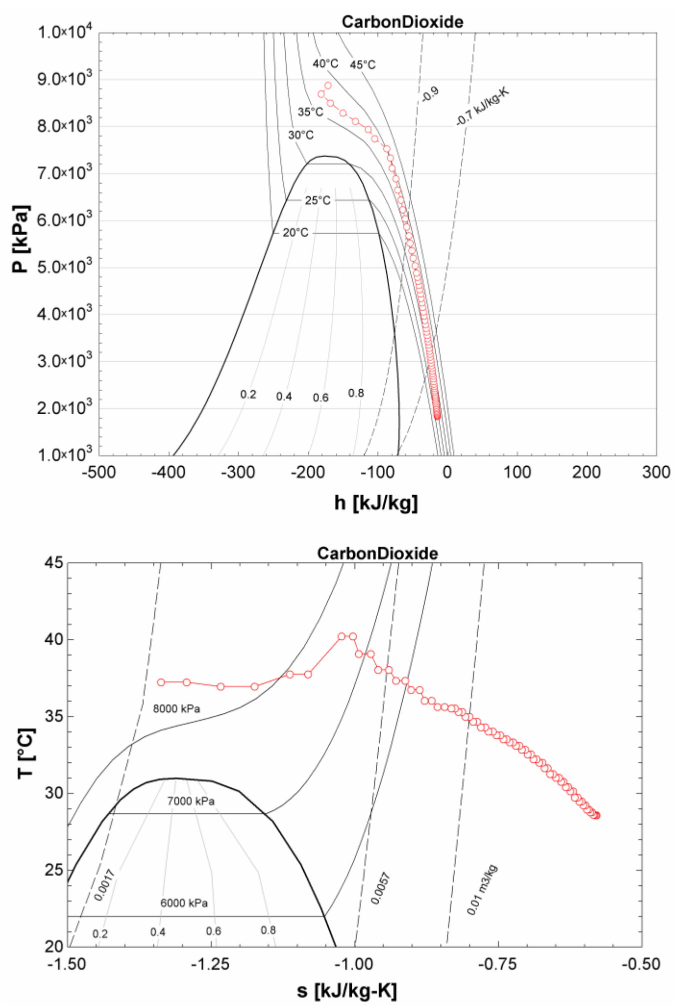

Figure 5: (Above) Pressure-enthalpy and (below) temperature-entropy charts of the compression of CO_2 showing near-isothermal compression.

A higher speed of compression was then investigated. Figure 6 shows the temperature rise against the pressure normalized by the starting pressure each cycle for seven cycles. The first five cycles, shown in the figure by dotted lines, were performed at a relatively lower liquid flow rate of approximately 2 $\times 10^{-3}$ m^3/min while the last two cycles, shown in the Figure by solid lines, were performed at a relatively higher liquid flow rate of approximately 3

$\times 10^{-3}$ m^3/min. All seven cycles had similar behavior regarding the temperature change of the CO_2 during compression.

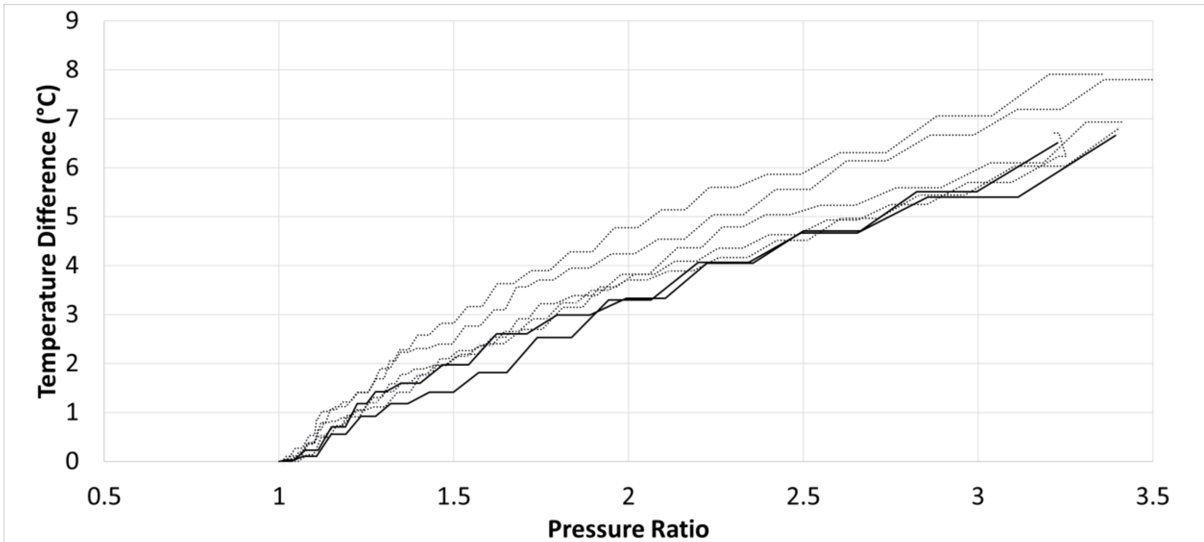

Figure 6. Seven compression cycles of the near isothermal liquid compressor. Dotted lines are at a liquid flow rate of approximately 2×10^{-3} m^3 per minute, and solid lines are at a flow rate of approximately 3×10^{-3} m^3 per minute.

During this testing, qualitative observations were made that indicate the current limit of the liquid flow rate for the experimental prototype. The sixth cycle, which was the first at the higher flow rate, resulted in some entrainment of liquid in the CO_2. This was evident as during the discharge of cycle seven, also using the higher flow rate, oil discharged to the catch can, which is shown in the upper right portion of the P&ID in Figure 2.

Follow up testing was then performed to investigate the optimal design of the nozzle that creates the spray. Because the spray pattern could not easily be observed in the prototype, the tests were performed with water spraying into a clear polycarbonate tube that is approximately the same diameter of the compression chamber using the same nozzle as was used in the previous experiments. The flow rate of water was maintained at 3×10^{-3} m^3/min, the same maximum flow rate for the experimental prototype compressor. The nozzle supplier literature was consulted to determine that the appropriate pressure drop across the nozzle to achieve this flow rate was 83 kPa. The appropriate flow rate was determined by observing the upstream water pressure on the water flow and changing the flow using a gate valve.

The results of this testing are shown in Figure 7. The figure shows that the current nozzle creates a strong jet that penetrates the surface of the water. The interface between the gas and liquid then becomes a mixture of the two fluids. If this phenomenon occurs with the compressor prototype, this mixture of the two fluids likely creates the entrainment of the compression liquid into the compressed gas, which results in the catch can filling with propylene glycol. This entrainment is a limiting factor to increasing the flow rate of liquid to sufficient rates to better mimic an actual CO_2 compressor.

Two additional nozzle designs were tested with the same apparatus to compare their spray patterns to the one used in the previous compressor testing. Each of these nozzles created a finer spray resulting in less penetration of the jet into the water surface. This reduction in spray penetration with water in the plastic tube will likely translate well into a reduction of the entrainment of liquid into the compressed CO_2 in the experimental compressor prototype. In our future work, we will use the new nozzles with the next prototype of the near isothermal LPC.

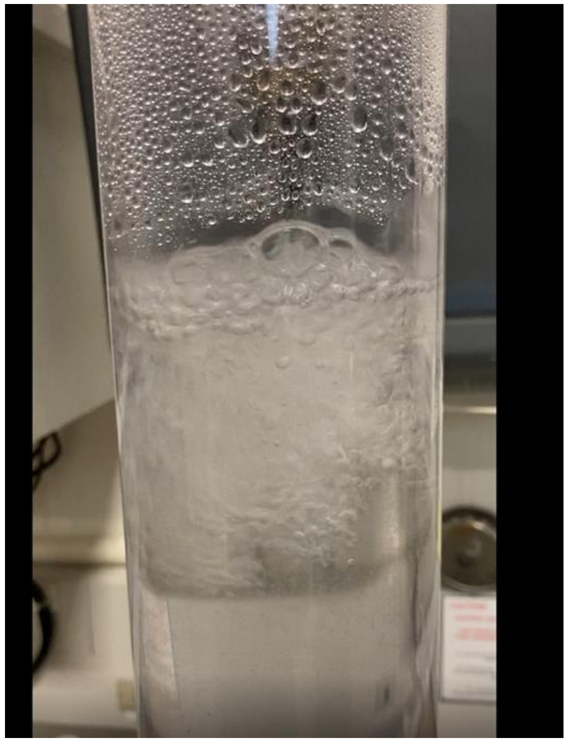

Figure 7. Spray pattern of the spiral design nozzle with water.

4. CONCLUSIONS

We developed a prototype of a near-isothermal LPC that can compress CO_2 at various pressure regimes using a spray of droplets of propylene glycol. Initial experiments demonstrated near-isothermal operation with a temperature rise of only 2.5 K in the compressed gas. These results were compared with a compression process without spray with a temperature rise of 7 K in the compressed gas. We demonstrated compression at 2.0×10^{-3} m^3/min and 3.0×10^{-3} m^3/min in compression of a gas to a supercritical fluid condition. Carryover of the compression liquid into the compressed gas led us to consider the nozzle used for creating the liquid droplets. Observation of the droplets outside of the compression chamber allowed us to select new nozzle designs for future work that will decrease the likelihood of liquid carryover and obtain higher rates of liquid flow.

In future work, we will develop a new prototype to perform testing at higher rates of liquid flow that will result in a continuous flow of compressed gas. We will expand to two chambers to enable cycling operation. The new prototype will allow independent operation of the compression process from the atmosphere using pistons and actuators that will move the liquid from one chamber to the other. Furthermore, our work with the single compression chamber has shown that it may be appropriate to use spray cooling of the compressed gas only to a certain point after which the spray nozzle will be turned off and the liquid piston will only be fed by the flow of liquid from the bottom of the compression chamber. In this way, carryover of the compression liquid to the compressed gas will likely be avoided.

NOMENCLATURE

LPC	Liquid piston compressor	(–)
P&ID	Piping and instrument diagram	(–)

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