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# Development and Evaluation of Ammonia Vapor Compression Coupled to a $CO_2$ Convection Loop

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## Abstract

A novel approach to the application of zero GWP refrigerants for commercial space cooling was demonstrated using ammonia vapor compression chilling coupled to pumped carbon dioxide convection. Following the lead of emerging supermarket refrigeration systems, this approach is both modeled after and meant to substitute halocarbon-based pumped chilled water cooling. Compared to pumped chilled water, the use of $CO_2$ coupled with ammonia-based chilling could provide a system targeting multiple paths of optimization: reduced cost, increased efficiency, lower GWP potential, improved flexibility. Potential cost reduction may come from reduced piping cost, reduced installation cost, reduced physical building load and reduced pumping cost. These are all derivatives of the higher heat capacity of phase-changing $CO_2$ as compared to pumped water. Furthermore, low-charge ammonia provides a cheaper alternative to high GWP refrigerants. Efficiency gains potentially come from the use of ammonia as the vapor compression refrigerant—a fluid with inherently favorable thermodynamic properties as an HVAC refrigerant. Additionally, power may be reduced in pumping liquid $CO_2$ compared to pumping water. GWP reduction potential comes from eliminating the use of halocarbon refrigerants and from higher system efficiency. There is also potential for reduction related to embedded manufacturing cost, though this study did not evaluate those possibilities. In this first prototype system, the concept was proven to be technically feasible. The ability to operate a convection loop at a wide range of temperatures, including below water's freezing point, allows for design and operational flexibility that isn't possible with a water loop. Cooling coils may be operated colder in order to adjust the sensible heat ratio (SHR) and thus have flexibility for greater dehumidification without energy intensive re-heat. This paper summarizes the technical approach to design and shares testing results of the prototype 8-ton system.



*Keywords: Heat pump; Hydronic*

## 1. Introduction

In California 14.9% of electricity is used for cooling applications (high temperature applications, not refrigeration). California commercial building cooling energy consumption was estimated to be 10,017 GWh in 2004. The majority of the existing stationary HVAC systems use R22, R410A, R134a, or R407C as the working fluid (refrigerant). R22, a very common HCFC has ozone depleting potential (ODP) as well as GWP. HFCs such as R134a have zero ODP, but still have high GWP.[1]

Environmental concerns have drawn the attention of many regulatory bodies which have undertaken various initiatives to reduce the effect of HCFC's and CFC's. The most famous legislative action in this area is the Montreal Protocol, which mandated the phasedown first of CFCs, and more gradually of HCFCs such as R22. This action was primarily focused on preventing ozone depletion. Later legislation such as the Kyoto Protocol targets global warming, and with that, sets goals for HFC phase-down. The Montreal Protocol was signed by the US, and the Clean Air Act includes actions relating to this protocol. The US did not sign the Kyoto Protocol, but the US EPA is authorized to act through the Significant New Alternatives Policy (SNAP) Program, also under the Clean Air Act, and is doing so to phase down the usage of many high-GWP refrigerants. On a state level, California's Air Resources Board (CARB) has a Refrigerant Management Program (RMP)[2] which institutes requirements for leak checking, reporting, and logging of usage of refrigerants, first for heavy users (such as supermarkets and refrigerated warehouses), and later for smaller equipment.

Abroad, the European Union's "F-Gas" bans include aggressive requirements to phase down production, and eventually recovery and use of refrigerants with "high GWP" (150-2000, depending on application). Some European nations further have specific taxes targeting refrigerants. In Europe, these legislative actions are significant enough that alternative refrigerant adoption is considerably higher than in the US.[3]

Some promising, low GWP refrigerants may have other considerations, like toxicity or high pressure, making requiring specific protocols for safe operation. The above concerns – safety and environmental considerations – must be considered in parallel to equally important fundamental thermodynamic behavior of the fluid, which determines component sizing, operating pressures, efficiency and capacity, and so on. Combining all of these factors leads to a challenge, as no substance provides the ultimate combination of traits for all applications.

A long-term objective is to provide HVAC&R solutions using natural refrigerants. Natural refrigerants are defined as compounds that are naturally occurring and have thermo-physical properties suitable to be used in a refrigeration cycle with zero ODP and zero or near-zero GWP. Natural refrigerants like ammonia ($NH_3$) and carbon dioxide ($CO_2$) which are environmentally benign are possible alternatives but introduce certain other considerations that require tradeoffs. For $NH_3$, flammability and toxicity are primary concerns whereas $CO_2$ has challenges in high-ambient cooling operation and system operating pressures. Another potentially attractive option is the use of hydrocarbons, such as propane, which have high efficiency, but are restricted in how they may be applied because of flammability.

This paper describes the construction and evaluation of a first-of-its-kind ammonia to carbon dioxide chiller and convection loop. It is designed as a potential replacement for traditional chiller/hydronic HVAC systems. The system was shown to reliably and safely operate through a range of typical operating condition. The system has some inherent adjustability, allowing flexibility in the sensible heat ratio. Efficiency was reasonable with coefficients of performance ranging from ~2 – 3, considering that components were mostly taken from other, low temperature refrigeration applications, and the system was not fully optimized for HVAC operation. There is likely room for improvement in further designs.

There is potential for reducing material and installation cost because of the smaller (and lighter) piping requirements for pumped carbon dioxide compared to traditional pumped chilled-water. Simple economic analysis shows install cost may be reduced by a factor of 2-3.

Future development in this area should focus on optimizing for HVAC temperatures to push efficiency higher, design systems that incorporate heating or dual-purpose systems and field testing prototype systems.

## 2. System Design

A system was designed and constructed that uses natural refrigerants with low to zero GWP. The system is comprised of a low-charge Ammonia ($NH_3$) based direct-exchange (DX) vapor compression air-cooled chiller coupled to a pumped Carbon Dioxide ($CO_2$) convection loop. Refrigerant charge was less than 1 pound $NH_3$ per ton. This arrangement is aligned with traditional systems for space cooling using halocarbon based chillers coupled to chilled-water pumped loops. The evaluation is a hybrid between laboratory and field setup, using the EPRI, Knoxville laboratory space as the "field" installation, but taking measures such that indoor conditions could be held reasonably fixed. Figure 1 shows a schematic for the $NH_3/CO_2$ system.

The chiller is a nominal 8-ton semi-hermetic chiller with ammonia as the working fluid. It was a beta prototype unit at the time of evaluation, designed to exchange heat on the source (cooling) side through an ammonia to carbon dioxide plate heat exchanger.

Carbon dioxide was piped in a separate convection loop, coupled to the chiller, much as a chilled water loop would be coupled in traditional systems. $CO_2$ remained in a saturated state throughout evaluation, maintained at pressures between ~3.1-3.8 MPa (450 – 550 psig). A central receiver tank acted as the repository for $CO_2$. Vapor from the top of the receiver entered the chiller heat exchanger, condensed while flowing through it and returned to the receiver through gravity—there was no pumping for the chiller-$CO_2$ heat exchange.

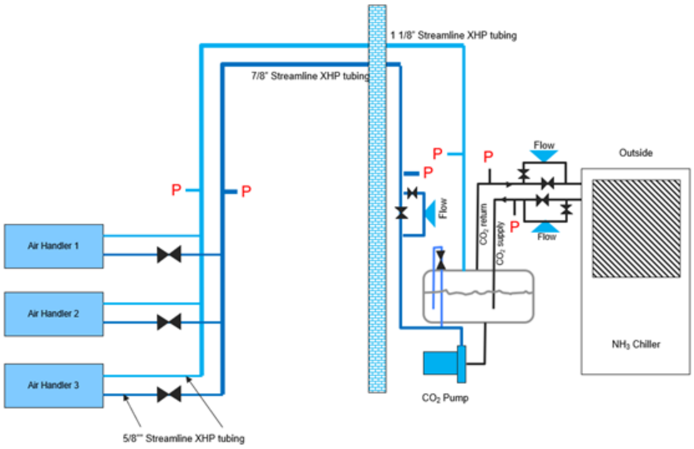


Figure 1. System Schematic

## 3. Evaluation Approach

The objective was to evaluate overall system operation, with a focus on measuring cooling capacity, power draw characteristics and system efficiency as functions of ambient temperature conditions. Other system characteristics were observed and measured as appropriate. Such other characteristics may include part-load dynamics, standby pressure conditions and cycling behavior.

The system can be viewed with multiple control volumes:

- Control volume around the entire system, measuring overall system input (electrical power) and output (cooling capacity). This approach gives insight on the overall system efficiency for comparison to other comparable approaches (e.g. halocarbon chiller with pumped chilled water).
- Separate control volumes around the $NH_3$ chiller and the pumped $CO_2$ loop. This approach isolates each subsystem so that capacity, power draw details and efficiency can be measured for each.

Instrumentation of the setup is designed to accommodate both approaches.

### *3.1. System Control Volume Approach*

Referencing the system control volume in Figure 2, overall system capacity is determined through air-side measurement on the indoor air handlers. Capacity is calculated by:

$$\dot{Q}_{air\ capacity} = \sum_{i=1}^{4} \dot{m}_{air,i}\,[h_{air,out} - h_{air,in}]_i \qquad [1]$$

Where:
$\dot{m}_{air}$ is the mass flow rate of air discharged from each air handler. $h$ is air enthalpy.

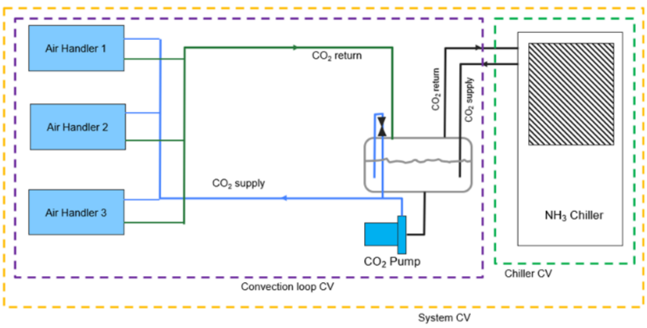


Figure 2. Control Volumes

System power is measured at five points, summing to the total power use of the system:

- Chiller unit power
- $CO_2$ pump rack and controls power
- 3-6 -> Air handler 1-3 power

$$Total\ Power = \sum_{i=1}^{6} Power_i \qquad [2]$$

Overall system efficiency, termed the Coefficient of Performance (COP) is thus defined as:

$$Overall\ system\ efficiency\ (COP) = \frac{\dot{Q}}{Power} \qquad [3]$$

These three quantities constitute the primary metrics of system performance. All three are functions of operating conditions, namely the outdoor and indoor dry bulb temperature and the indoor relative humidity.

$$\dot{Q}, Power, COP \sim f(T_{out}, T_{in}, RH_{in}) \qquad [4]$$

Air side capacity measurement is an industry accepted method for determining total delivered system capacity. It captures the heat transfer usefully delivered to the space and does not capture otherwise lost heat—losses. In a true building arrangement, there may be additional losses associated with the details of ducting. These are not captured in a laboratory style evaluation. The method of test is generally guided by AHRI 340/360 approaches and methods of measurement, though strict ambient air conditions are not maintained with psychrometric-style testing).

Air temperature is measured with type T thermocouple arrays; relative humidity is measured with capacitive RH sensors and air volume flow is measured via a pitot tube array. Air mass flow is calculated from air density, via measured static air pressure, temperature and relative humidity.

### *3.2. Subsystem Control Volume Approach*

Referencing **Error! Reference source not found.** an additional control volume is shown around the air-cooled chiller. With both the full system CV and the chiller CV, then three overall sets of output metrics (capacity, power draw and efficiency) can be calculated:

- Full system
- Air-cooled chiller
- Convection loop

Because of complexities of measurement, only the full system CV was used for analysis of the experiment. Following is a discussion of some of the associated complexities for sub-component measurement, owing primarily to the nature of 2-phase CO2 flow.

Convection loop metrics are then the difference between the full system and the chiller. For example:

$$\dot{Q}_{conv\ loop} = \dot{Q}_{system} - \dot{Q}_{chiller} \qquad [5]$$

$\dot{Q}_{conv\ loop}$ is expected to be negative, indicating losses in the piping network.

#### *3.2.1. Chiller CV Measurement*

Measuring the chiller capacity is somewhat challenging. An air side approach can be used, though accurately measuring air volume flow is difficult on the type of low-static fans that move air through the condensing coil. The act of measuring can substantially change the volume flow rate, so some method of boosting airflow, with a secondary fan, back to its non-disturbed state, is necessary. This approach is sometimes also employed for indoor unit testing, if air flow is significantly affected by the act of measuring.

A more direct method of measuring chiller cooling output is to measure the convection loop heat transfer. In water cooled chillers this is straightforward as the product of mass flow, specific heat of water and temperature differential:

$$\dot{Q}_{chilled\ water} = \dot{m}_{water}[h_{in} - h_{out}] \qquad [6]$$

Which can be simplified to:

$$\dot{Q}_{chilled\ water} = \dot{m}_{water}C_{p,water}[T_{in} - T_{out}] \qquad [7]$$

Since enthalpy is a direct function of temperature for liquid water (a sub-cooled liquid).

For the $NH_3$ system, using pumped $CO_2$ as the convection fluid, the method is not as simple. The $CO_2$ undergoes condensation in the heat exchange process with the chiller and is therefore usually in a 2-phase state—shown by the green line in **Error! Reference source not found.**. Though equation [6] still governs the heat exchange process experienced by the $CO_2$, there is no effective way to measure the enthalpy of 2-phase $CO_2$ with only temperature and pressure measurements.

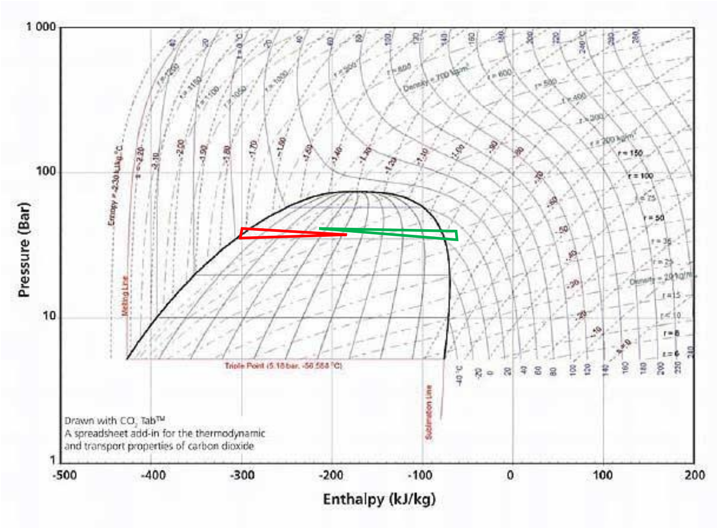


Figure 3. Illustration of Possible CO2 State Points (Green→ Chiller-to-Receiver Loop; Red → Receiver-to-AHU Loop)

For this evaluation, it was not possible to measure the enthalpy change across the chiller $CO_2$ loop. The $h_{in}$ (the enthalpy of $CO_2$ returning to the chiller) is saturated or slightly superheated vapor, because of the geometry of its draw from the top of the receiver tank, enabling enthalpy to be calculated from measured temperature and pressure. $h_{out}$ is a different matter, it can be 2-phase, saturated or sub-cooled. An enthalpy calculation is only possible if it is reliably saturated or sub-cooled (though there is no way to confirm saturation, so some small amount of sub-cooling is required).

## 4. Test Set Up

The test performed in this report was set up as a hybrid laboratory/field installation, where the laboratory space could be conditioned with reasonable control over the indoor ambient conditions. The chiller, $CO_2$ receiver, and the $CO_2$ pump were positioned outside and were subject to the outdoor ambient conditions of the summer testing period. The three indoor units were positioned in a horizontal orientation and stacked on top of each other in parallel so that temperature and humidity could be easily controlled. Supplemental heating load to the conditioned space could be adjusted to any wattage, up to ~30 kW. A variety of fans and deflectors were used to de-stratify the air and to ensure uniform entrance temperature to the three air handlers.

Humidity was supplied through manually controlled steam injection. Steam flow was adjusted to reach a rough target, return air, relative humidity of >40%. In steady-state operation of the tested system, return air conditions could be held reasonably stable (+/- 1.1C and 5%RH). Each indoor unit was assembled and setup to accommodate air-side capacity measurements.

## 5. Testing Results

In addition to some general break-in and operational familiarity testing, five general tests were performed on the system:

- Low Ambient (~21-26°C)
- Moderate Ambient (~28-31°C)
- High Ambient (~32+°C)
- Standby Operation
- Power Failure

### *5.1. Low Ambient*

Low ambient testing was performed with an outdoor ambient range of 21°C to 26°C as the outdoor temperature increased over the course of the test. The system was initially started with all three air handlers on and was then allowed to equilibrate. Air handlers 1 and 2 were successively turned off which is represented in Figure 4 as the test stepped down from three, to two, to one IDUs operating. Figure 4 shows cooling capacity versus time, and shows that with two and three IDUs operating, the chiller runs continuously, whereas with only one IDU operating, the chiller cycles. Figure 5, a graph of the chiller power consumption as a function of time, also shows the cycling where the power drops significantly during the cycling period and drops to zero when an IDU is turned off (transition effect). The data collected below is an average across a pseudo steady state temperature during that portion of the test. This low ambient test produced an operating COP range from 2.45-3.14.

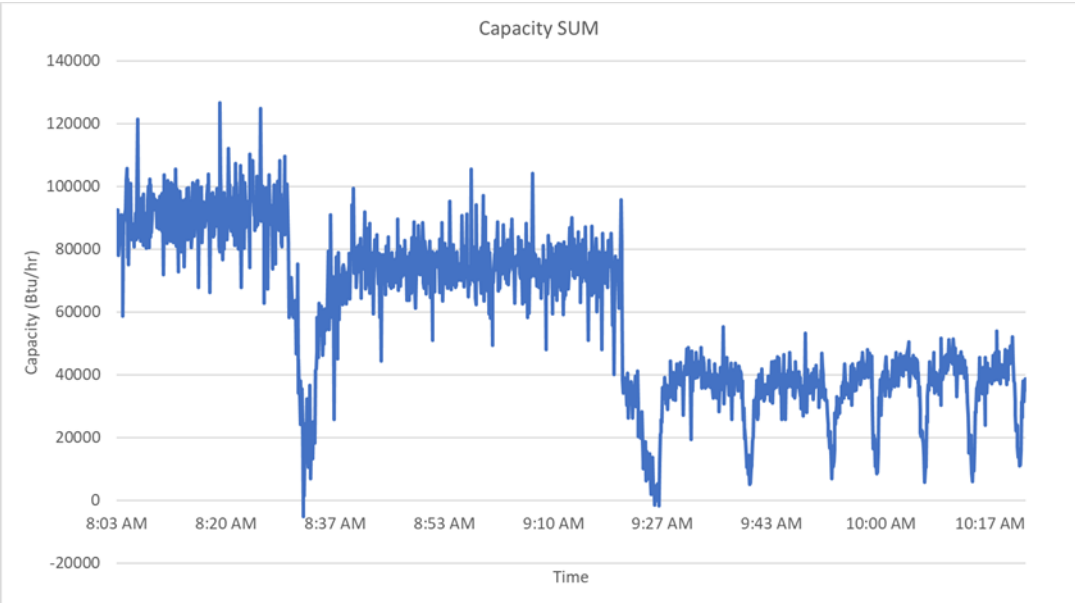


Figure 4. Cooling capacity vs time (low ambient test)

### *5.2. Moderate Ambient Test*

Medium ambient testing was performed with a steady-state outdoor ambient temperature of approximately 30°C. The system was initially started with all three IDUs turned on and was then allowed to equilibrate. The indoor temperature was held constant at approximately 27°C as the system reached equilibrium. After collecting the data in the steady-state, IDUs 1 and 2 were successively turned off similarly to the Low Ambient test. The moderate ambient test produced lower cooling capacity while increasing the ODU power. This moderate ambient test produced an operating COP range from 2.01-2.55 as shown in Table 1.

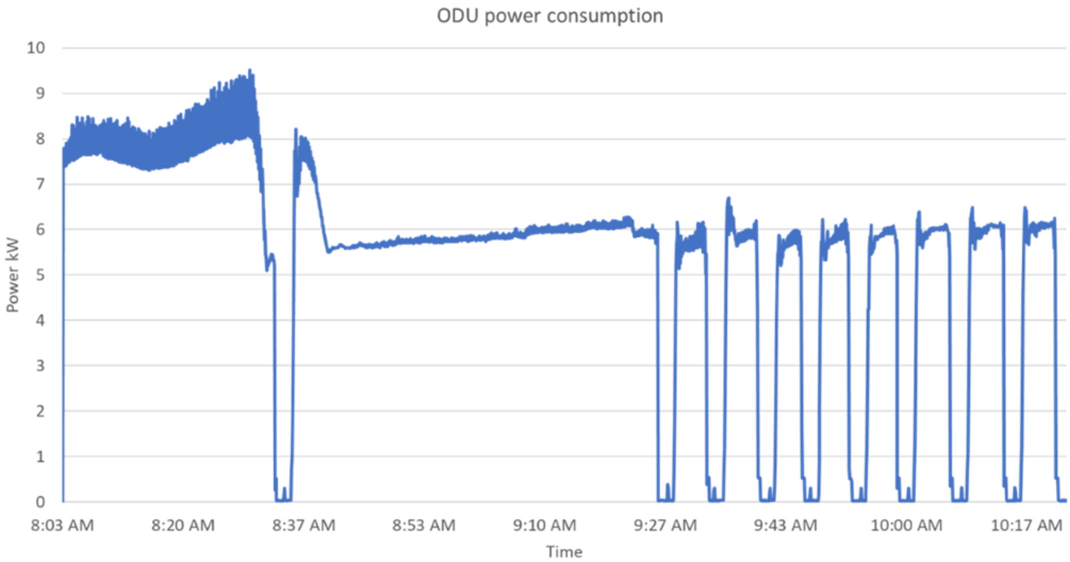

Figure 5. Chiller power vs. time (low ambient test)

Table 1. Steady-state operation for moderate ambient testing

| **Number of IDU ON** | | 1 | 2 | 3 |
|---|---|---|---|---|
| Outdoor Temp | °C | 30 | 30 | 30 |
| $CO_2$ Pressure | MPa | 3.5 | 3.4 | 3.6 |
| Capacity | kW | 8.44 | 19.27 | 24.44 |
| COP | - | 2.01 | 2.51 | 2.55 |
| IDU Power | Watts | 327 | 643 | 949 |
| Pump Power | Watts | 184 | 200 | 200 |
| ODU Power | Watts | 3,692 | 6,846 | 8,432 |
| Total Power | Watts | 4,202 | 7,689 | 9,581 |

### *5.3. High Ambient Test*

High ambient testing was performed by adding supplemental electric heat to the condenser inlet in addition to the ~29°C ambient outdoor air at the time of testing. This additional heat boosted the condenser inlet air temperature to approximately 33°C. All three IDUs were in operation throughout the entire test. The high ambient temperature with three IDUs running consumed total power around 12.6 kW. Total air-side cooling capacity was 28.84 kW (~8.2 tons), which operated at a COP of 2.14 (Table 2). Figure 6 shows the relationship between the ambient temperature with additional resistive heat and the cooling capacity of the system. The data observed was taken from operating temperatures of ~91°F which are featured in Figure 6. $CO_2$ tank pressure remained stable between ~3.4-3.6 MPa (500-525 psig).

Table 2. Steady-state operation for high ambient testing

| **Number of IDU ON** | | 3 |
|---|---|---|
| Outdoor Temp | °C | 33 |
| $CO_2$ Saturation Temp | °C | 4 |
| Capacity | kW | 28.85 |
| COP | - | 2.14 |
| IDU Power | Watts | 943.7 |
| Pump Power | Watts | 192 |
| ODU Power | Watts | 11,543.7 |
| Total Power | Watts | 12,679.4 |

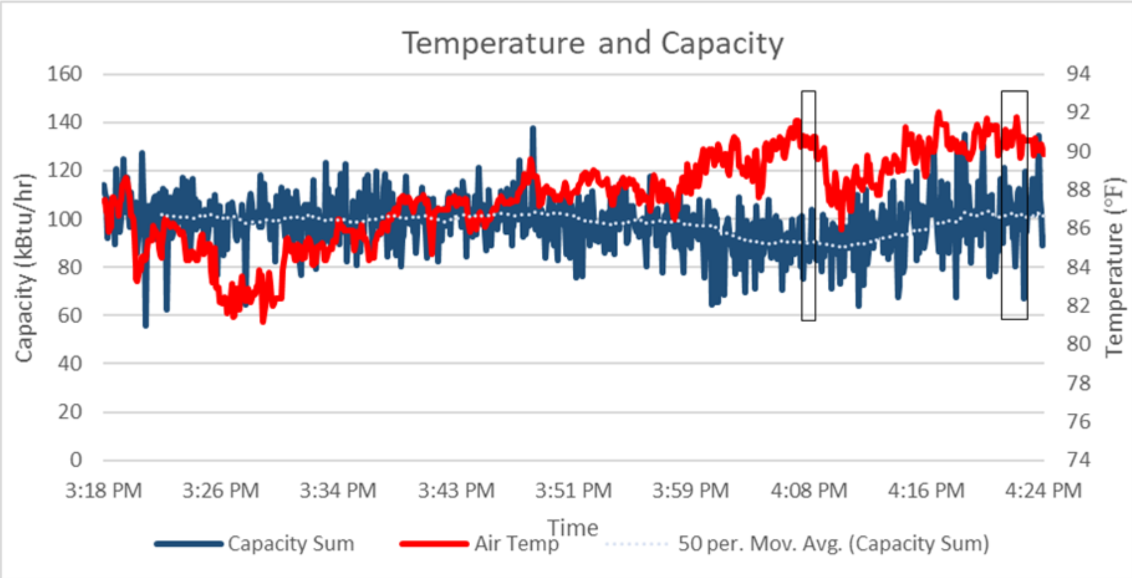


Figure 6. Cooling capacity and outdoor temperature versus time (high ambient test)

### *5.4. Standby Operation*

A standby test was performed to observe how the chiller operates when there is no cooling load imposed on the system. The chiller periodically cycles on to cool the CO2 receiver tank as shown in Figure 7—as $CO_2$ tank pressure and Chiller power as a function of time. The pressure in the tank oscillates between ~3.27-3.59 MPa (475 – 520 psig) over a period of approximately 20 minutes. The time between intervals decreases slightly as the ambient outdoor temperature increases over the course of the test.

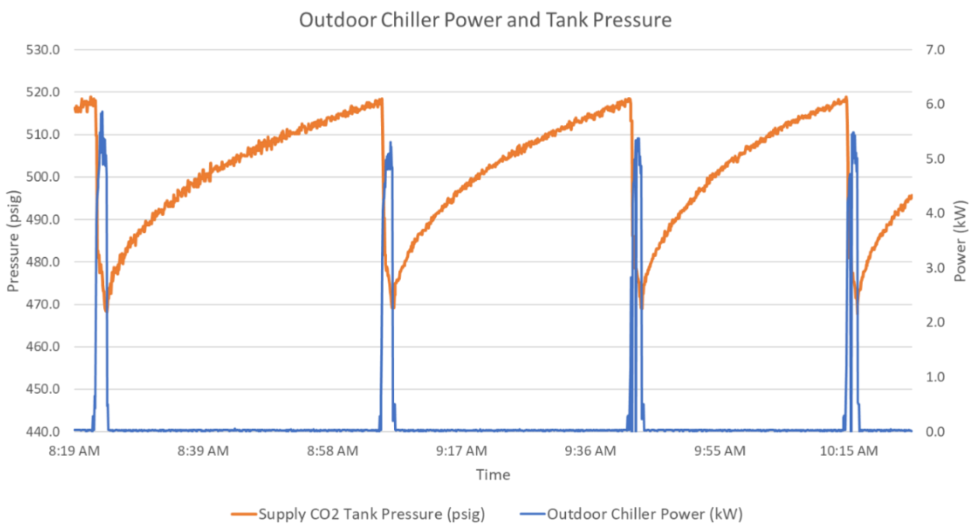


Figure 4. Outdoor chiller and tank pressure versus time (standby operation test)

### *5.5. Simulated Power Failure*

This test was performed to ensure that the system would properly re-energize and reset after a power outage or power failure. The system was initially fully turned on with all three IDUs in operation. All power was then cut from the system by de-energizing the main circuit breakers. After a 10-minute standby, the power was re-applied to the system and observations regarding the test were as follows:

- When power was off, $CO_2$ tank pressure naturally drifted up ~0.007 MPa/min (1 psi/minute)
- When power was re-applied, the chiller began its restart sequence.
- After 5-minute built-in delay, the compressor and ancillary systems restarted.
- $CO_2$ tank temperature dropped accordingly, and the system began cycling normally.

The test setup required the $CO_2$ pump and the three IDUs to be manually restarted after a power failure to restart the cooling process. This was a design choice for this particular experimental setup and is not necessarily indicative of how a system would actually be set up in a field deployment. Automatic restart of pumping and cooling could be easily accommodated with proper control strategies.

## 6. Economic Analysis of Aspects of Installation

There is potential installation cost savings from using phase-change $CO_2$ as the convection fluid instead of water. $CO_2$ is used in some refrigeration systems because it can provide low-temperature (sub-freezing) convection. For HVAC applications where the convection temperature can remain above freezing, CO2 still may offer an advantage of smaller pipe size and, in turn, lower installation cost. The following is an example per-unit calculation of relative material and installation cost for $CO_2$ convection. This analysis is focused on the $CO_2$ convection loop, not the ammonia chiller. It is assumed that capital cost per unit of installed capacity of an ammonia or halocarbon chiller is roughly equivalent, leaving any difference in cost to the convection loop only.

The boiling heat capacity of $CO_2$ at ~3.44 MPa (500 psig), is approximately 416 kJ/kgK (99.3 Btu/lbm℉), while for water, the heat capacity is ~4.19 kJ/kgK (1 Btu/lbm℉). If typical chilled water applications undergo a 5.5°C-temperature change, then phase-changing $CO_2$ can provide up to 10 times the convection heat transfer per unit mass of pumped fluid. This 10x heat transfer is the key point of exploration for potential installation cost reduction. The full 10x may not be realized in actual systems because full phase transition may not occur as systems may be designed to return a partially boiled mixture of liquid and vapor $_{CO2}$ from the evaporator.

Chilled water is typically piped in welded steel or grooved connected steel pipe. CO2 for HVAC applications can be piped in braised high-pressure copper alloy which is a variant of standard ACR type K copper refrigerant tubing. The 8-ton system under test used 7/8" (22mm) O.D. braised copper/iron tubing for the $CO_2$ supply line and 1-1/8" (28.5mm) for the return. A similarly sized chilled water convection loop would use ~1.5"-4.0" (38-102mm) steel pipe. The $CO_2$ system with its respective hardware would accommodate ~7.95 lpm (2.1 gpm) of $CO_2$ flow compared ~72.7lpm (19.2 gpm) of water flow for the similar-sized chilled water convection loop. For an 8-ton system, as described in this report, the $CO_2$ copper piping would cost 2.67 times less to install than the chilled water steel pipe. Table 30 shows a comparison of installation costs of variously sized ACR copper and welded steel pipe in price per linear foot.

Table 3. Comparison of installation cost of various sizes of ACR copper and welded steel pipe

| Tubing/Pipe Size | Raw Material | Raw Labor | Total Installed (with O&P) |
|---|---|---|---|
| Type K Copper (ACR) | Cost ($) per linear foot | | |
| 7/8" | $4.97 | $2.83 | $10.10 |
| 1 1/8" | $7.00 | $3.2 | $12.95 |
| 1 3/8" | $9.35 | $3.68 | $16.35 |
| Schedule 40 Welded Steel | Cost ($) per linear foot | | |
| 1 ½" | $5.70 | $8.50 | $21.06 |
| 2" | $11.25 | $10.60 | $30.76 |
| 3" | $16.55 | $15.05 | $44.14 |
| 4" | $17.00 | $17.45 | $48.87 |

## 7. Summary and Conclusions

A first-of-its-kind, natural refrigerant based HVAC system was constructed and evaluated. The system uses ammonia ($NH_3$) in a low-charge packaged chiller as the primary refrigerant for vapor compression, and carbon dioxide ($CO_2$) as the convection fluid to distribute cooling to remote indoor air handlers. This combination takes advantage of the high efficiency of $NH_3$ and the high heat capacity of $CO_2$ to provide an environmentally friendly option for medium to large HVAC applications. The technology builds on advances made in the supermarket refrigeration sector where refrigerant management is highly important and the shift toward zero GWP refrigerants is already underway.

The general workability of the system approach was demonstrated successfully. Cooling capacity and efficiency were reasonable, with limited effort to optimize the component selection and system controls, likely leaving room for substantial improvement in future designs.

This system was only designed and tested for cooling operation. The concept can be extended to include heating with additional consideration and componentry to accommodate high-pressure CO2 for heat convection.

The smaller size of piping required for CO2 convection (compared to hydronic convection) offers potential for material and installation cost savings, that could help to offset any capital premium for this new type of equipment.

## Acknowledgments

The authors would like to thank Southern California Edison (SCE) and Mayekawa (MYCOM) for their participation in this project.

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