A Near-Zero-GWP Heat Pump System for All-Electric Heating & Cooling in California

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ABSTRACT

Heat pumps are ideal for decarbonization of space heating end use in the California climate and will play a substantial role in reaching the state's energy goals. Unfortunately, heat pumps available today use high global warming potential (GWP) refrigerants whose leakage into the atmosphere has the potential to offset any gains made by decarbonization of space heating. While refrigerant regulations are beginning to drive the transition to lower GWP alternatives, this piecemeal approach may not be sufficient and energy efficient ultra-low-GWP natural refrigerant technologies must be considered to fully meet emissions targets. This project demonstrates a natural refrigerant based heat pump system for small commercial applications (10 to 20 tons) with a unique design: ammonia (R-717) as the primary refrigerant and carbon dioxide (CO_2) as a distribution fluid with small diameter copper piping. R-717 has superior thermodynamic properties compared to synthetic refrigerants, and CO_2 loop holds tremendous potential as a cost effective and efficient alternative to conventional hydronic loops. The challenge with R-717 stems from its toxicity and mild flammability, but risks can be minimized by utilizing a low charge packaged system with semi-hermetic compressor. The challenge with utilizing CO_2 as a distribution fluid is the necessity for a CO_2 compressor in heating mode since CO_2 will be in a supercritical state. However, this project has demonstrated the possibility to circulate supercritical CO_2 with the use of a centrifugal pump, providing a path to realize substantial energy and cost savings.

Introduction

Heat pumps are ideal for decarbonization of space heating end use in the California climate with mild winters. They are not reliant on fossil fuels unlike boilers or furnaces, and they offer higher efficiency than electric resistance heaters. Unfortunately, heat pumps available today use high GWP refrigerants whose leakage into the atmosphere has the potential to offset gains achieved by decarbonization of space heating. Conventional refrigerants in space conditioning such as R-410A and R-32 are potent greenhouse gases with global warming potential (GWP) of 2088 and 675 respectively. GWP is a metric of the contribution to global warming resulting from the emission of one unit mass of the refrigerant relative to one unit of mass of CO_2, which has a GWP of 1. Currently, there are no market available commercial scale reversible heat pumps with near-zero GWP in the U.S. at costs competitive with units using conventional refrigerants.

Conventional space conditioning systems typically utilize synthetic hydrofluorocarbon (HFC) refrigerants due to several factors: they are non-toxic, non-flammable, and have no ozone depletion potential (ODP). They are also compatible with copper, which is used in hermetic or semi hermetic compressor motors. However, HFC refrigerants tend to have very high GWP, and their usage can counteract carbon reduction policies. As a result, there has been regulatory

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actions on the use of HFCs and other high GWP gases. Historically, California has some of the most aggressive refrigerant phase down regulations within the U.S., with the California Air Resource Board (CARB) standards limiting the GWP of refrigerants used in new stationary heat pumps to 750 beginning in 2025, with variable refrigerant flow systems in 2026 (SNAP, 2018). In recent years, the U.S. saw new federal regulations on refrigerant based on their GWP with the American Innovation and Manufacturing (AIM) Act of 2020. The AIM Act calls for phasedown of high GWP refrigerants by 60% starting 2024 (EPA, 2023a). In 2023, the AIM Act final rule further restricts the use of refrigerants with GWP greater than 700 for stationary air conditioning and heat pumps. (EPA, 2023b)

Regulations on refrigerant GWP has sparked interest in the use of natural refrigerants that have favorable thermodynamic properties and low environmental impact. However, natural refrigerants tend to be flammable (propane/R-290) or toxic (ammonia/R-717) and require significant component and system redesign for deployment. There are restrictions on the use of these refrigerants depending on the application due to safety concerns, and stringent limits on refrigerant charge are often applied to mitigate hazards from potential leakages. As a result, the use of natural refrigerants has been limited to systems with very small charge amounts (e.g. refrigerators) or industrial processes with minimal risks to personnel in case of refrigerant leaks. Given the trend towards lower GWPs, facility and plant managers in commercial buildings may be faced with difficult decisions on which systems to install, since lower GWP alternatives (e.g. R-32, R-454B) may face future phase downs and few natural refrigerant options exists.

This paper presents a first of its kind heat pump system with natural refrigerants for commerical buildings applications that mitigates the risk of using hazardous refrigerants. The heat pump utilizes R-717 as the primary refrigerant with a CO_2 distribution loop. R-717 was selected as the refrigerant due to its favorable thermodynamic properties and zero GWP. The use of a CO_2 distribution loop allows the toxic refrigerant to be contained outdoors and allows the use small diameter pipes. The following sections describe the motivation behind the selection of the working fluids, a detailed description of the system and reports on the fabrication of the prototype system.

Refrigerant Selection

The space conditioning system presented in this paper utilizes R-717 as the primary refrigerant. R-717 is one of the oldest known refrigerants and it is widely used in industrial process cooling and refrigeration. It offers three distinct advantages over HFC refrigerants. First, it is a naturally occurring substance that is widely available, environmentally friendly, and does not deplete the ozone layer or contribute to global warming. Second, R-717 has superior thermodynamic qualities, and the overall energy efficiency of the system can be substantially improved if features such as premium motor, electronic expansion valve, and variable capacity compressor are included. Third, though R-717 is a hazardous substance, its recognizable odor is a great safety asset. Unlike most other refrigerants that have no odor, R-717 leakages are not likely to escape detection. However, use of R-717 in space conditioning is hampered by limitations such as potential leaks and special handling due to its B2L refrigerant classification. ASHRAE standard 34 classifies class B2L as refrigerants with higher toxicity and lower flammability (AHSRAE, 2022).

R-717 is typically used in large capacity systems (>50 RT), and the total charge amount results in rigorous regulations even at low-charge amounts. The use of ammonia faces restrictions on the federal, state and local levels. In particular, the U.S. EPA has different regulations applying for site inventory thresholds of 500 lbs and 10,000 lbs and requires emergency release notification in the event of leaks exceeding 100 lbs in a 24-hour period (EPCRA, 2019). Similarly, Occupational Safety and Health Administration (OSHA) requirements apply to R-717 facilities, with additional requirements when exceeding the 10,000 lbs threshold (OSHA, 2012). State level regulations are common too, most notably in California, where the quantity for increased scrutiny is 500 lbs (CalARP, 2014). Inspections and reporting are required at regular intervals, and compliance audits must also be undertaken at regular intervals. Several efforts are underway to develop regulations specifically for low charge R-717 systems that can take advantage of its high efficiency while minimizing the risk of harm to due leaks, such as the ASHRAE position document on R-717 (ASHRAE, 2008).

Previous work at the Electric Power Research Institute (EPRI) investigated the potential of a packaged R-717 chiller for space cooling with promising results. A 10-ton (35.2 kW) R-717 chiller was coupled with a carbon dioxide (CO_2) distribution/secondary loop to the indoor units (EPRI, 2022). The heat pump system considered in this paper follows a similar design, and only contains ~110 lbs of R-717 charge, well below the threshold for most burdensome regulations that otherwise govern the use of. This is possible due to the use of the packaged system approach with advanced heat exchangers, this will be further detailed in the System Description section.

Secondary Fluid Selection

CO_2 was selected as the secondary fluid for this system, and it has several advantages over the use of water in typical hydronic systems. The main motivation is that CO_2 can be evaporated in the indoor air handling unit (AHU). The allows the system to leverage the heat of vaporization of CO_2 at ~500 psig (~40 ºF), approximately 99.3 Btu/lbm, is significantly higher than the heating capacity of liquid water, approximately 20 Btu/lbm with a 20 ºF ΔT. Phase changing CO_2 can provide up to 5 times the convection heat transfer per unit mass of pumped fluid, and this improvement in heat transfer can be leveraged to realize energy savings. The higher heat transfer per unit mass allows the system to use smaller diameter pipes and lower mass flow rate in the distribution loop over.

The CO_2 loop has advantages over conventional water/glycol loops when considering upfront costs in new installations. Chilled water loop is typically piped in welded steel or groove connected steel pipe. For the 20 tons capacity considered in this paper, a hydronic loop will have a flow rate of ~24 gpm assuming a typical temperature differential of 20 ºF. This requires 2” piping according to AHSRAE recommendations (ASHRAE, 2006). This is a widely available material, with raw materials at ~$11.25/ft and installation cost at ~$10.60/ft. Assuming a 40% markup for overhead and profit, the total installed cost results in $30.59/ft. On the other hand, the CO2 loop can achieve the same capacity with smaller 7/8” high-pressure copper alloy piping, a variant of standard ACR type K copper refrigeration tubing. This is a standard alloy available for CO_2 refrigeration installations and can be brazed with standard practices and materials. The cost of this copper piping is ~$4.79/ft with installation at ~$2.83/ft, resulting in a total installed cost of ~$10.67/ft at similar markups. The savings are substantial when comparing any significant pipe length, ~$19.92/ft of installed pipe.

However, the use of CO_2 can be challenging due to high system pressures and the need for high pressure rated components, specifically new piping and heat exchangers for retrofit applications. Due to the high operating pressures of CO_2, the risks of leakage into the indoor space must be managed using leakage detection systems, which may be an additional burden on equipment costs. There is an additional challenge with utilizing CO_2 as a secondary fluid during heating mode operation. The critical temperature of CO_2 (87.98 °F) is lower than the temperature typically delivered to heating coils. Therefore, using CO_2 as a space heating distribution fluid generally results in circulation of supercritical CO_2. Commonly available CO_2 compressors are designed for transcritical refrigeration cycles with low evaporating temperatures/pressures and a high-pressure lift. However, these optimized parameters match poorly with the space heating operating conditions considered in this paper, specially, the evaporating temperature will be much higher with a lower pressure lift. Therefore, using a CO_2 compressor for the secondary loop in space heating may significantly limit the efficiency for the system.

Supercritical CO_2 has relatively high density and this characteristic motivated previous work at EPRI that investigated and verified the feasibility of *pumping* supercritical CO_2 with a high-pressure CO_2 pump (Robinson et al, 2023). The feasibility study and previously described work on R-717 chiller motivated the development of the heat pump system described in this paper. The use of high-pressure CO_2 pumps for circulating liquid CO_2 in cooling mode and supercritical CO_2 in heating mode instead of an additional compressor allows for a much simpler system design and mitigates the efficiency penalties associated with the CO_2 compressor.

System Description

The R-717 heat pump system is a packaged R-717 heat pump with a CO_2 distribution loop connecting the heat pump with the indoor AHUs through a R-717 to CO_2 heat exchanger, a schematic of the prototype is shown in Figure 1. The designed capacity of the prototype is 20-tons, with 10-tons for each AHU. A CO_2 compressor is also included on this prototype to compare the heating capacity and efficiency against the pumps. A packaged approach is used for the R-717 side, which isolates the refrigerant outdoors and reduces the charge required for the system. This also allows the system to be factory welded and assembled for easy deployment and future scale up. The CO_2 secondary loop will need to be brazed in the field. The system can operate in four distinct modes:

- Heating Mode with CO_2 Pump: R-717 heat pump is in heating mode and CO_2 pump is used to circulate heat transfer fluid.
- Heating Mode with CO_2 Compressor: R-717 heat pump is in heating mode and CO_2 compressor is operating as a heat pump in heating mode. Cascade cycle.
- Cooling Mode with CO_2 Pump: R-717 heat pump is in cooling mode and CO_2 pump is used to circulate heat transfer fluid.
- Cooling Mode with CO_2 Compressor: R-717 heat pump is in cooling mode and CO_2 compressor is operating as a heat pump in cooling mode. Cascade cycle in opposite direction of 2 above.

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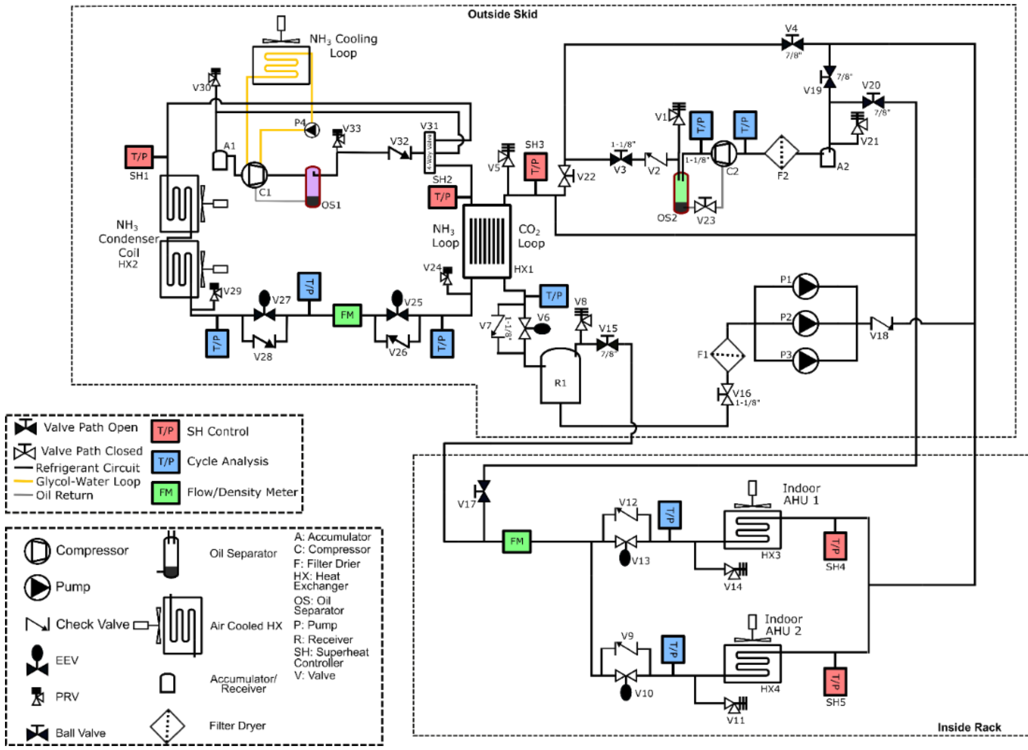

Figure 1. System schematic of ammonia heat pump with carbon dioxide distribution loop. *Source*: EPRI.

Figure 2 shows the R-717 heat pump loop with the state points over a pressure-enthalpy diagram. States 1 (compressor suction) and 2 (compressor discharge) can be measured using the superheat controllers (red-shaded), while states 3 and 4 are measured using additional temperature sensors and pressure gauges (blue shaded).

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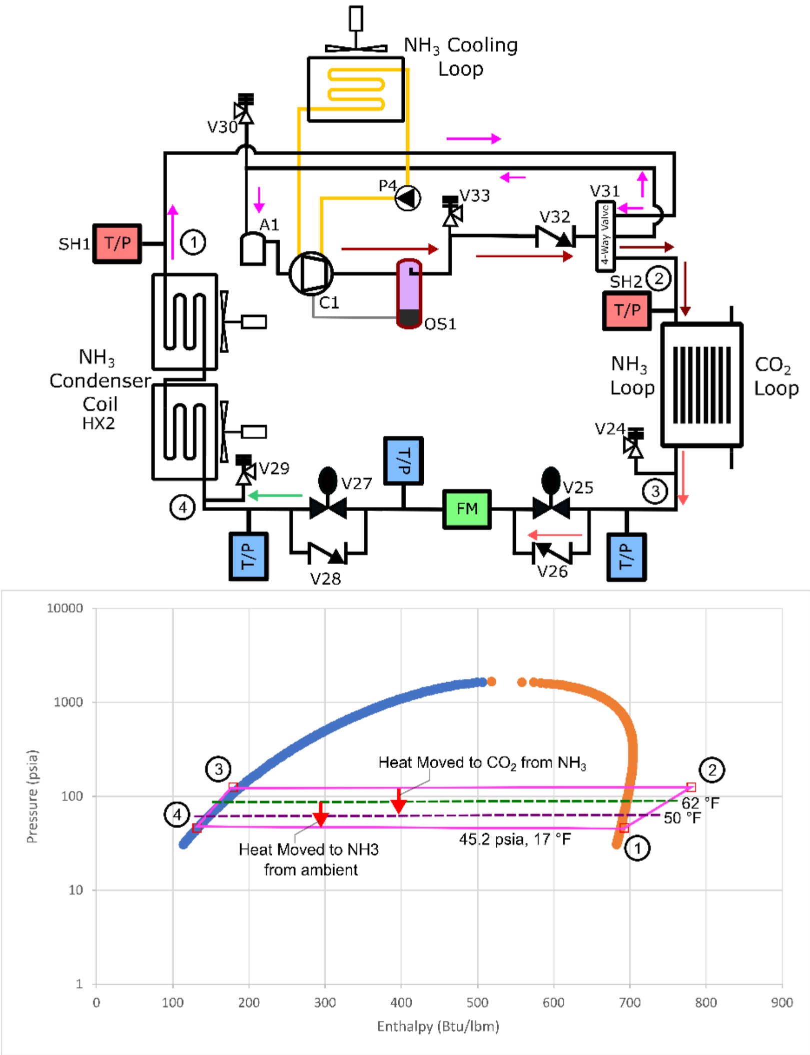

Figure 2. Top - R-717 heat pump in heating mode with salient thermodynamic states (circled). Bottom - Pressure and enthalpy diagram of R-717 with operating thermodynamic states which correspond to the above figure. *Source*: EPRI.

Similarly, Figure 3 shows the CO_2 loop (compressor mode) with the state points over a pressure-enthalpy diagram. Since the CO_2 loop has long line sets that run from the outdoor unit to the indoor unit, two additional locations are instrumented to accurately measure the CO_2 loop capacity to account for the potential pressure and temperature drops. The first is at outlet of the intermediate R-717 to CO_2 heat exchanger (State 1"), which is compared to the suction of the CO_2 compressor (State 1). The second is at the inlet of the indoor CO_2 to air heat exchangers (State 2"), which is compared to the discharge of the CO_2 compressor (State 2).

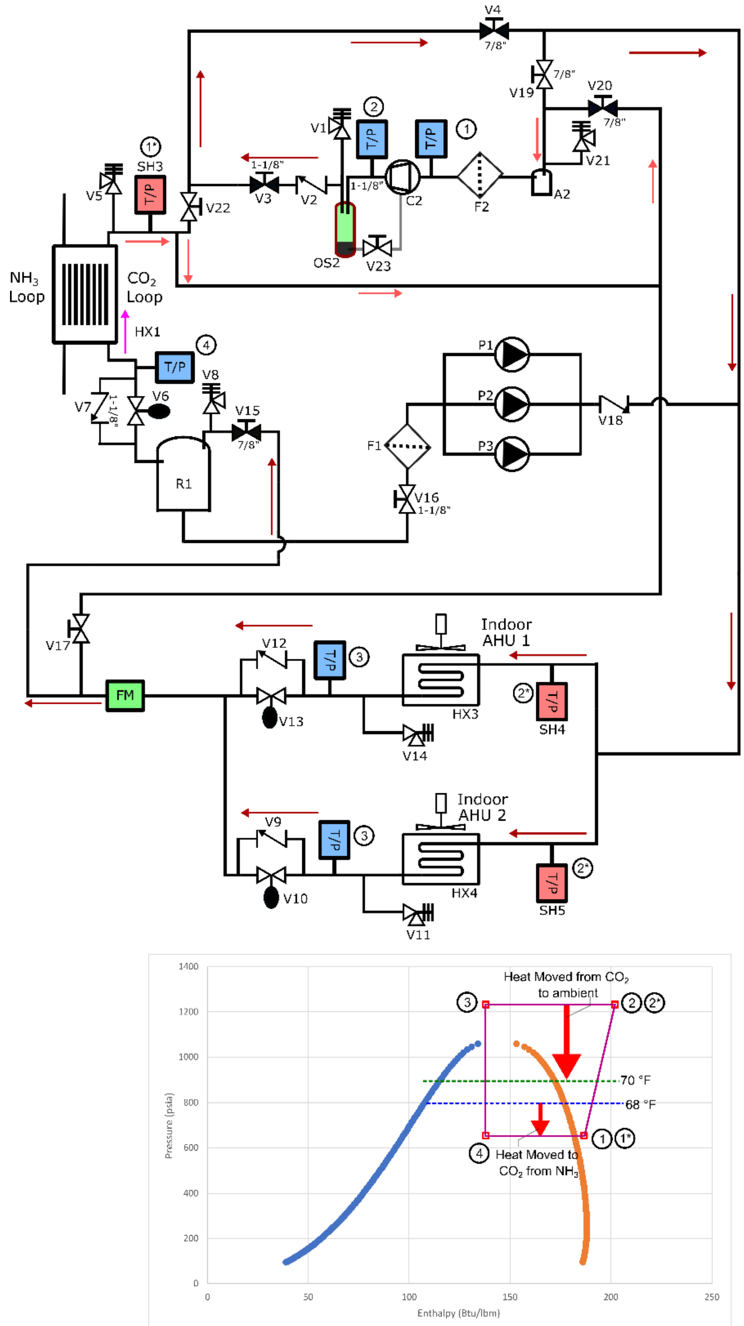

Figure 3. Top - CO_2 heat pump in heating mode with salient thermodynamic states located (circled). Refer to Figure 1 for valve open and closed indication. (Bottom) Pressure and enthalpy diagram of CO_2 with operating thermodynamic states which correspond to the above figure. *Source*: EPRI.

Modeled Performance

The heat pump performance at ambient operating conditions were modeled using available component data from manufacturers. This provided a straightforward approach to evaluate performance, though it did not reveal real-world annual performance where systems operate for many hours at lower partial-load conditions. These simulations were valuable for evaluating system performance and contextualizing it amongst competing products on the market. Heating performance was modeled with only the CO_2 compressor operating, since the use of the high pressure CO_2 pump for circulating supercritical CO_2 lacks a comprehensive performance map.

Modeling was accomplished by harmonizing vendor-supplied heat exchanger and compressor performance data into a simple model. The indoor CO_2 and outdoor R-717 tube-fin heat exchangers were modeled and tuned minimally to match vendor performance data. Polynomial expressions for compressor isentropic efficiency as a function of pressure ratio were developed through regression of supplier data. By iteratively solving heat exchanger models with compressor maps, the saturation temperatures were determined, and compressor powers were calculated. These results are shown in Table 1.

Table 1. Estimated refrigerant charge based on component and line set volumes.

Condition	Temperature (outdoor/ indoor) [°F]	R-717 Compressor Power [kW]	CO_2 Compressor /Pump Power [kW]	Fan Power [kW]	EER [Btu/hr-W] / COPh [W/W]
Cooling – *Compressor*	95 / 80	13.4	6.1	6.5	**9.2**
Cooling – *pumped*	95/ 80	17.5	0.2	6.5	**9.9**
Heating – *Compressor*	47 / 70	5.7	11.7	6.5	**2.9**

The modeled results show cooling energy efficiency ratio (EER) of 9.2 for cooling with the CO_2 compressor and an improved EER 9.9 with the CO_2 pump. While the modeled data may not include some variables in real world conditions, the increase in efficiency at the same capacity and operating temperatures show the potential of the CO_2 distribution loop. On the other hand, the heating coefficient of performance is 2.9 at 47 ºF and 2.4 at 17 ºF. The efficiency of CO_2 pump may be less in heating mode than cooling, but this will be offset by the inefficient operation of the CO_2 compressor due to the unfavorable operating conditions. Additionally, there are benefits from simplifying the design if a CO_2 compressor is not required. As with most modeling approaches, the project team anticipates the results shown above is likely higher than the real-world achievable efficiencies. However, there may still be potential efficiency gains when compared against most installed HVAC equipment at minimum efficiency standards, and there remains opportunities for component and controls optimization.

Prototype Fabrication

A prototype heat pump is constructed and will be evaluated in future studies in a laboratory setting. Components were sourced from multiple manufacturers and assembled into the prototype instead of procuring a packaged system directly from a manufacturer since no such product is currently available on the market. R-717 required stainless steel components for material compatibility. A semi-hermitic compressor with no mechanical seal between the compressor and the compressor motor was chosen. This was particularly important for a R-717 system since mechanical seal is one of the most vulnerable points and the seal can wear out over time and become prone to leakage. This resulted in a system that is practically leak free since there is no route for the refrigerant to escape in the compressor, which is highly desirable for a space conditioning application. The compressor is rated at 40 tons due to limited product selection at the time of design. The only two commercially available semi-hermitic R-717 compressors were the selected 40 ton product, and a 8 ton compressor that was only available as part of a packaged chiller system. As a result, the 40 ton compressor was loaded to 50% to match the needs of the prototype. The recommended oils for the R-717 compressor were either mineral oil or polyalkylene glycol (PAG). The former was selected for the prototype with a recommended amount of 2.4 gal. An oil separator was also installed to remove oil from the refrigerant and route the oil back to the compressor. Pressure relief valves were installed at the oil separator, the outdoor coils, intermediate heat exchanger, and accumulator with a setting of 350 psig.

The intermediate plate heat exchanger was made of stainless steel and rated for 110 bar (1595 psi) since it must be suited for both working fluids. The CO_2 distribution loop utilized high pressure rated components and copper iron piping. The heat pump system utilized the same high pressure rated CO_2 pump from the supercritical testing in previous work. Based on the previous test data, a total of 3 pumps were installed to ensure adequate heating capacity. The CO_2 compressor required lubricating polyolester oil (POE) with a manufacturer recommended amount of 2.4 gal and an oil separator. A receiver was used to manage CO_2 charge since the charge needed for cooling and heating operation were different. Pressure relief valves were installed at the oil separator, the indoor AHU coils, intermediate heat exchanger, and receiver with a setting of 1400 psig.

The use and availability of copper-iron as the primary tubing in the CO_2 circuit simplified the fabrication process by allowing technicians to use standard brazing practices (oxy-acetylene torch and silver solder). However, the use of CO_2 as a refrigerant was not as common as other refrigerants, and this was reflected when integrating components together. Any component integrated into the copper-iron tubing required fittings that can adapt to accommodate different diameter tubing stubs on equipment and threads, and some direct fittings were not available. For example, Figure 4 shows the adaption from the 7/8" system tubing to a ½" filter drier. The filter drier had *solder cups* (typically denoted with a C) that allowed for ½" tubing to be set in the cup and brazed. Unfortunately, a copper iron 7/8" cup (C) to ½" fitting (F) was not available. In its place, two fittings were used to make the transition: 7/8" coupling (C-C) and 7/8" to ½" reducer (F-F).

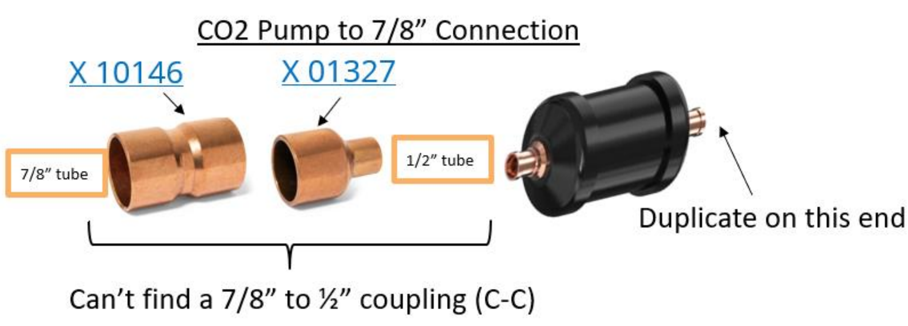

Figure 4. Utilization of multiple fittings used to accommodate the 1/2" filter drier to the 7/8" copper tubing on the CO_2 distribution loop . *Source*: EPRI.

The amount of refrigerant needed for the prototype heat pump was estimated based on the overall system volumes with assumed densities. The outdoor R-717 heat exchanger coils,

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intermediate plate heat exchanger, R-717 accumulator, R-717 oil separator, R-717 filter dryer, and 10 ft of line set were used as the volume for R-717. The density used for R-717 was the saturated liquid density at 90 °F. The indoor CO_2 coils, intermediate plate heat exchanger, CO_2 accumulator, CO_2 oil separator, CO_2 filter dryer, and 200 ft of line set were used as the volume for CO_2. The density used for CO_2 was the saturated liquid density at 40 °F. The resulting refrigerant amounts are given below in Table 2. It should be noted these estimates do not take into account the actual refrigerant line set volumes and assume the full heat exchanger volume is filled with liquid refrigerant, which is unlikely to occur in a real system. These calculations provided an estimate of how much refrigerant is required, which helped to identify any relevant safety regulations for R-717. R-717 ordered was to a purity of 99.995%, and CO_2 was ordered (refrigerant grade) for the CO_2 to ensure a low moisture content in the gas.

Table 2. Estimated refrigerant charge based on component and line set volumes.

Refrigerant	Volume (ft^3)	Density (lb/ft^3)	Charge (lb)
R-717	2.93	36.94	108.2
CO_2	2.0	56.16	112.3

The refrigerant cylinders were connected to charging equipment via regulators which serve as adapters. The R-717 regulator was compatible with R-717 (no copper/copper alloys) and connects via a CGA-705 fitting and provides a 1/4” MNPT connection for the charging equipment, and non-copper charging equipment was procured for the R-717 heat pump. The CO_2 regulator is specified for high pressure and provides a CGA-320 connection to the cylinder and a ¼” MNPT to allow connection to the system.

Conclusion

Natural refrigerant based heat pumps present tremendous potential in both energy efficiency gains and environmental impact mitigation. These systems can play a key role in the upcoming industry scale transitions as the HVAC community reacts to refrigerant GWP regulations and utilities react to carbon reduction targets. There are currently very few natural refrigerant based heat pump systems on the market that is designed for space conditioning. While deployments in Asia and Europe are increasing, the U.S. is just beginning to transition away from legacy HFC systems. This paper presents a prototype heat pump system with ammonia (R-717) as the primary refrigerant and a CO_2 based distribution loop, utilizing innovative approaches in circulating CO_2 in heating mode. The system design minimizes the R-717 charge using a packaged approach and overcomes traditionally barriers on charge limits for R-717 and other hazardous refrigerants. While the prototype’s performance is limited by the availability of suitable components, it demonstrates an achievable avenue in fabricating safe and effective natural refrigerant based heat pump systems. As more manufacturers offer components compatible with natural refrigerants, this prototype heat pump system can be more readily scaled for utility programs and wide deployments.

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