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FINAL PROJECT REPORT

A Zero GWP Heat Pump and Distribution System for All-electric Heating and Cooling in California

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PREFACE

The California Energy Commission's (CEC) Energy Research and Development Division supports energy research and development programs to spur innovation in energy efficiency, renewable energy and advanced clean generation, energy-related environmental protection, energy transmission, and distribution and transportation.

In 2012, the Electric Program Investment Charge (EPIC) was established by the California Public Utilities Commission to fund public investments in research to create and advance new energy solutions, foster regional innovation, and bring ideas from the lab to the marketplace. The EPIC program is funded by California utility customers under the auspices of the California Public Utilities Commission. The CEC and the state's three largest investor-owned utilities — Pacific Gas and Electric Company, San Diego Gas & Electric Company, and Southern California Edison — were selected to administer the EPIC funds and advance novel technologies, tools, and strategies that provide benefits to their electric ratepayers.

The CEC is committed to ensuring public participation in its research and development programs that promote greater reliability, lower costs, and increase safety for the California electric ratepayer and include:

- Providing societal benefits.
- Reducing greenhouse gas emissions in the electricity sector at the lowest possible cost.
- Supporting California's loading order to meet energy needs, first with energy efficiency and demand response, next with renewable energy (distributed generation and utility scale), and finally with clean, conventional electricity supply.
- Supporting low-emission vehicles and transportation.
- Providing economic development.
- Using ratepayer funds efficiently.

A Zero GWP Heat Pump and Distribution System for All-electric Heating and Cooling in California is the final report for EPC-19-014 conducted by Electric Power Research Institute. The information from this project contributes to the Energy Research and Development Division's EPIC Program.

For more information about the Energy Research and Development Division, please visit the CEC's research website (www.energy.ca.gov/research/) or contact the Energy Research and Development Division at ERDD@energy.ca.gov.

ABSTRACT

Heat pumps are ideal for decarbonizing space heating in California's moderate climate. Unlike boilers and furnaces, they are not reliant on fossil fuels, and they offer higher efficiency than electric resistance heaters. However, these systems depend on high global warming potential refrigerants that can be expensive to manufacture and are subject to regulations based on their environmental impact. These conditions present an avenue for deploying energy efficient natural refrigerant-based technologies in the effort to reach emission reduction goals, as well as a potential area for decarbonization through electrification. This project evaluated a first-of-its-kind ammonia-based heat pump with a carbon dioxide distribution system for multi-family or small commercial applications with 10- to 20-ton capacity. The design allows ammonia charge to be minimized while exploiting the attractive carbon dioxide heat transfer characteristics. The design and model efforts using commercially available components showed promising results with efficiency exceeding that of conventional products. However, the laboratory prototype evaluation revealed several technical challenges, which caused unstable system operation and low heating and cooling capacity. Several factors can be responsible for transience and underperformance, including control of an oversized ammonia compressor, compressor and pump speed on both ammonia and carbon dioxide sides, expansion valve positions, indoor unit fan speed, and system refrigerant charge. The combination of these factors made it difficult to ascertain which factor primarily caused the system issues. Even though the project will not be able to fulfill the originally planned field demonstrations, the laboratory evaluation accomplished a significant milestone for the industry by demonstrating supercritical carbon dioxide circulation with a high-pressure carbon dioxide pump for space heating. This demonstration is a substantial development for electrification and decarbonization efforts as the industry transitions to hazardous refrigerants that require secondary loops.

Keywords: Heat pump, global warming potential, natural refrigerants

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Executive Summary

Heat pumps are ideal for the decarbonization of space heating end use in California. Unlike boilers or furnaces, they are not reliant on fossil fuels, and they offer higher efficiency than electric resistance heaters. Unfortunately, heat pumps available today use high global warming potential refrigerants whose leakage into the atmosphere has the potential to offset any gains achieved by space heating decarbonization. Global warming potential is a metric of the contribution to global warming resulting from the emission of one unit of mass of the refrigerant relative to one unit of mass of carbon dioxide (CO_2), which has a global warming potential of 1. Conventional refrigerants in space conditioning such as R-410A and R-32 are potent greenhouse gases with global warming potential of 2088 and 675, respectively. Currently, there are no commercially available heat pumps with near-zero global warming potential at costs competitive with units using conventional refrigerants. These technologies will support the achievements of California's energy goals as listed in the 2022 Scoping Plan from the California Air Resources Board (CARB, 2024), specifically those related to emissions:

- Build on success in meeting the 2020 target to achieve the state's 2030 greenhouse gas reduction mandate of 40 percent as a stepping stone to carbon neutrality by 2045.
- Reduce greenhouse gas emissions to zero in every sector by 2045 and complement with carbon sequestration.

Project Purpose and Approach

The project purpose is to develop, test, and demonstrate an advanced system for multi-family or small commercial building applications, in the range of 10 to 20 tons of refrigerating capacity. The primary refrigerant is ammonia, which has excellent thermodynamic properties and zero global warming or ozone depletion potential. Ammonia is toxic and slightly flammable; therefore, it is desirable to minimize the refrigerant charge for safety reasons. This can be achieved by reducing the ammonia piping length and using a CO_2 based distribution loop into the conditioned space. This proposed design exploits the superior performance of CO_2 as a cooling fluid due to its phase change, providing up to five times more heat capacity than chilled water per unit mass. The CO_2 loop can also use smaller diameter copper piping compared to conventional welded steel pipes for water circulation loops, providing additional cost savings for installation at an estimated $19 per foot of installed piping.

The project carried out a design and modeling effort using commercially available components. The system performance was determined using industry standard assumptions and optimization of heat exchangers and was designed for California climate zones 7 and 10. The optimization process involved working with several heat exchanger manufacturers on fine-tuning the specific operating conditions required for efficient operation. A bill of materials was created based on the design effort and a prototype system constructed at Electric Power Research Institute's Knoxville laboratories. The prototype was evaluated for both heating and cooling performances using the general lab space as a pseudo indoor thermal chamber. The main purpose of the lab test was to understand the prototype operation sequence and prepare for the eventual field demonstration phase. The prototype heat pump was planned to be deployed in two California climate zones with a minimum of five production units. These deployments were planned to provide operational data from real world environments. The installation, commissioning, and maintenance processes were intended to provide key insights from the industry workforce and outline key barriers for wide adoption.

Two key partners were identified throughout the project as demonstration host sites: University of San Diego (USD) and Southern California Edison. California State University Maritime Academy was also involved as a host site but had to drop due to personnel reduction. USD's Sports Center was identified as a suitable host site with collaboration from San Diego Gas & Electric. The building currently has space heating but no cooling systems. The prototype heat pump would replace the heating equipment and use the existing ductwork for space conditioning. The prototype can be installed in the rooftop area and isolated away from students and staff who use the facility. Southern California Edison's Technology Test Center, a laboratory that typically evaluates emerging technologies, was identified as another suitable site. The Technology Test Center's general lab space could be used in a similar fashion to the laboratory testing in Electric Power Research Institute's facilities but would also provide California climate operational data from within California. This unique setup also provides additional opportunities to further research the system's control and operation, since the Technology Test Center staff are well suited for early-stage technology evaluation.

Key Results

The design and modeling results showed several key insights on using ammonia with CO_2. To maximize the performance of the heat pump, the temperature lift (typically up to 40°F [22°C] for space conditioning) of the heat pump should predominantly be done by the ammonia cycle (heating mode lift of 23°F [13°C] and cooling mode of 15°F [8°C]) due to its superior thermodynamic properties. CO_2 as a refrigerant is more suited for large lifts (for example, 100°F [55°C]), such as refrigeration or water heating, and may compromise the system's overall efficiency. The difficulty with this approach lies in the heating mode operation, where CO_2 will be in a supercritical state due to its low critical point at 87°F (30.6°C). Supercritical state refers to a situation in which a substance exists above its critical point and exhibits liquid and gas-like properties. The most significant difference compared to conventional refrigerant systems is that as the fluid cools in the heat exchanger, it generally does not undergo a phase change. The most common example is the use of CO_2 in transcritical operation, where heat is rejected through a gas cooler instead of a condenser. The conventional approach with CO_2 transcritical operation, where the thermodynamic cycle exceeds the critical point of the working fluid, means the use of a CO_2 compressor is necessary. However, such compressors are optimized for larger temperature lifts and using them for a relatively small lift in a cascade fashion with the ammonia cycle can significantly reduce system efficiency. One possible alternative is to investigate the potential to circulate supercritical CO_2 with a CO_2 pump instead of a compressor, since the densities of liquid and supercritical CO_2 are similar.

The existing components' availability revealed another challenge. Very few semi-hermetic compressors are available, with only two options under 50 refrigerating capacity: one eight refrigerating capacity unit that is only sold as part of a packaged chiller and one 40 refrigerating capacity unit that is a standalone product. The design ultimately used the 40 refrigerating capacity compressor for cost effectiveness and decided to only load it at 50 percent to reach the desired 20 refrigerating capacity.

Prior to the construction of the laboratory prototype, a small-scale test loop was used to evaluate the feasibility of pumping supercritical CO_2. The testing results were largely favorable, providing a first-of-its-kind CO_2 distribution system for heating purposes. While additional testing may be required to fully characterize the CO_2 pump performance, the data suggest that three pumps may be able to provide similar heating capacities in the heat pump prototype as in the CO_2 compressor, but at one-sixth the required power. Therefore, the prototype was outfitted with both compressor and pumps to further investigate this potential.

The prototype was constructed and evaluated at Electric Power Research Institute's facilities. There were significant delays with this construction due to COVID safety protocols and supply chain issues. Once assembled, the prototype was charged with refrigerant and commissioned. Several incidents with ammonia leaks further delayed this process, with one catastrophic failure occurring due to a ruptured oil return hose on the ammonia system. This hose was subsequently replaced with stainless steel tubing. The initial evaluations showed very low system capacities, possibly due to the system being undercharged and some sections of the CO_2 loop not being insulated. Despite adding insulation and refrigerant charge to both the ammonia and CO_2 sides, similar issues continued. For cooling operation, the main problem was due to the piping configuration with the CO_2 compressor. The added complexity to support four operating modes resulted in the system drawing CO_2 vapor into the indoor units instead of liquid CO_2. When the evaluation switched to using the CO_2 pump, the results showed a substantial increase in capacity. The only remaining issue was the potential for cavitation since the CO_2 receiver is in a saturated state, so pumping action may cause it to boil rapidly.

As for heating tests, the prototype saw an unusual pressure spike during start up after several tests. This pressure spike exceeded the setting on pressure relief valves, and all attempts to run further tests resulted in release of ammonia through these valves. As soon as the compressor started, the refrigerant pressure built up rapidly, likely due to a faulty expansion valve stuck in the closed position. Replacing the faulty valve should have alleviated the issue. Due to several additional concerns with host sites, schedules, and budgets, the team and the California Energy Commission decided that it was not feasible to complete the five field demonstrations, and the laboratory evaluation was suspended.

Knowledge Transfer and Next Steps

The research findings were published and presented at several industry conferences including Emerging Technology Coordinating Council summits, American Council for an Energy-Efficient Economy Summer Study, and the International Energy Agency heat pump conference. Presentations were also hosted for Electric Power Research Institute's electric utility members and the California Air Resources Board. While the prototype laboratory evaluation faced technical challenges, several key research questions were identified for future investigation. The pumped CO_2 distribution loop should be further studied and developed to realize the potential savings with CO_2, particularly as a heating distribution fluid. Additional working fluids should also be investigated, including different primary refrigerants such as propane or additional secondary fluids that may provide further energy savings.

CHAPTER 1: Introduction

Goals and Objectives

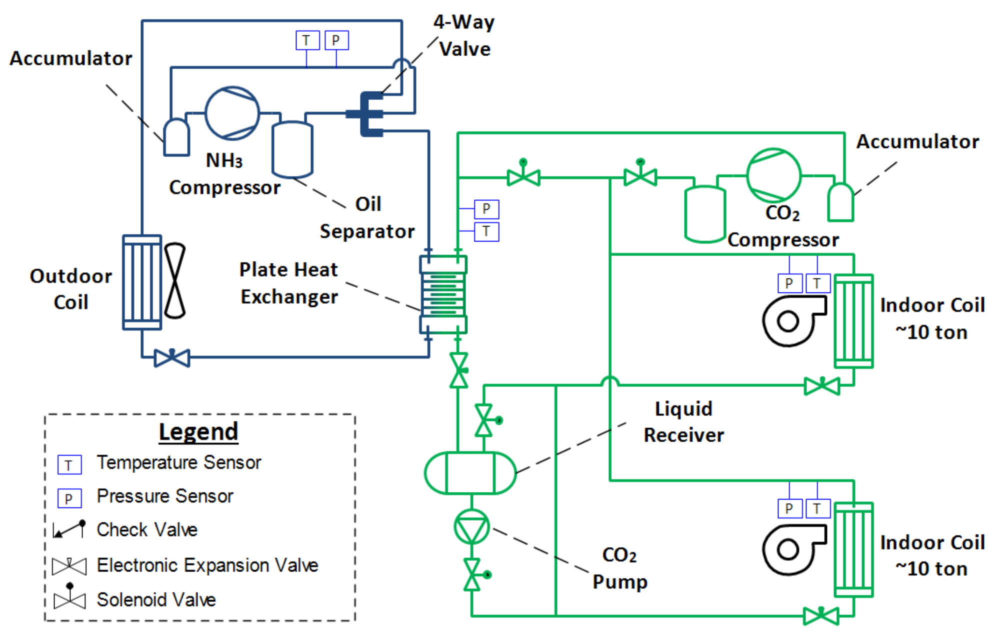
The goals of this project were to:

- Design, build, and demonstrate a prototype ammonia (NH_3) chiller with a carbon dioxide (CO_2) distribution system for cooling and heating for small commercial and multi-family applications. A simplified system diagram is shown in Figure 1.
- Demonstrate that the innovative reversible heat pump thermal cycle and distribution system is more efficient and less costly than comparable commercial heating, ventilation and air conditioning systems.
- Enable decarbonization of space heating using a reversible heat pump with near-zero global warming potential (GWP) at costs competitive with conventional refrigerants.

The objectives were to:

- Conduct laboratory optimization and evaluation of a prototype for the advanced reversible heat pump that is a cascading ammonia/CO_2 integrated refrigeration cycle and distribution system for both heating and cooling.
- Deploy five production units of the advanced reversible heat pump at three distinct California climate zones.
- Conduct measurement and verification of the field deployed units.

Figure 1: Simplified Schematic of the Ammonia Heat Pump with CO_2 Loop



Source: Electric Power Research Institute, 2021

Background and Motivation

Space conditioning accounts for a major portion of a building's energy consumption. The commercial sector accounts for up to 35 percent of total electricity consumption in the United States, and within the sector, heating, ventilation, and air-conditioning (HVAC) accounts for 26 percent of total United States electricity consumption at 185 billion kilowatt-hours (kWh) (633 trillion British thermal units) per year (EIA, 2012). In California's moderate climate, heat pumps are ideal for decarbonizing space heating. They are not reliant on fossil fuels like boilers or furnaces are, and they offer higher efficiency than electric resistance heaters. Commercial buildings are primarily cooled and conditioned by chillers and rooftop units in the United States, and these systems depend on refrigerants with high global warming potential (GWP) such as synthetic hydrofluorocarbon (HFC). GWP is a metric of the contribution to global warming resulting from the emission of one unit of mass of the refrigerant relative to one unit of mass of CO_2, which has a GWP of 1. Conventional refrigerants in space conditioning such as R-410A and R-32 are potent greenhouse gases with GWP of 2088 and 675, respectively. HFC refrigerants can also be expensive to manufacture and are subject to regulations based on their environmental impact. This presents an avenue for deploying energy efficiency technologies with low GWP refrigerants to reach emission reduction goals, as well as a potential area for decarbonization through electrification.

Historically, California has some of the most aggressive refrigerant phasedown regulations within the United States, with the California Air Resources Board standards limiting the GWP of refrigerants used in new stationary heat pumps to 750 beginning in 2025 and applying to variable refrigerant flow systems in 2026 (SNAP, 2018). In recent years, the United States 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 a phasedown of high GWP refrigerants by 60 percent starting in 2024 (EPA, 2023a). In 2023, the AIM Act Technology Transition Program final rule further restricted the use of refrigerants with GWP greater than 700 for stationary air conditioning and heat pumps (EPA, 2023b). Regulations on refrigerant GWP have 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). They require significant system redesign and face stringent limits on refrigerant charge to mitigate hazards. As a result, their use has been limited to small capacity systems (for example, refrigerators) or industrial processes with minimal risks to personnel. Currently, there are no commercially available heat pump technologies with near-zero GWP at costs competitive with HFC refrigerants.

Previous Electric Power Research Institute (EPRI) work investigated the potential of a packaged ammonia chiller for space cooling with promising results. A 10 refrigeration ton (RT) (35.2 kilowatt) ammonia chiller was coupled with a CO_2 distribution loop to the indoor units (EPRI, 2022). The heat pump system considered in this report follows a similar design and only contains about 110 pounds of ammonia charge, well below the threshold for most regulations that otherwise govern the use of ammonia.

Ammonia (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, ammonia 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 ammonia is a hazardous substance, its recognizable odor is a great safety asset. Unlike most other refrigerants that have no odor, ammonia leakages are not likely to escape detection.

However, ammonia is classified as a class B2L (higher toxicity and lower flammability) in ASHRAE standard 34, and its use in space conditioning is severely limited (AHSRAE, 2022). Ammonia is typically used in large capacity systems (greater than 50 RT), and even low-charge amounts can result in rigorous regulations. The U.S. Environmental Protection Agency (EPA) has regulations applying for site inventory thresholds of 500 pounds and 10,000 pounds and requires emergency release notification in the event of leaks exceeding 100 pounds in a 24-hour period (EPCRA, 2019). Similarly, Occupational Safety and Health Administration (OSHA) requirements apply to ammonia facilities, with additional requirements when exceeding a 10,000-pound threshold (OSHA, 2012). In California, the quantity for increased scrutiny is 500 pounds (CalARP, 2014). Inspections and reporting are required at regular intervals, and compliance audits must also be undertaken at regular intervals.

CO_2 is used as a distribution fluid in supermarket refrigeration systems, and EPRI's prior research demonstrated its use as a secondary loop for space cooling. The motivation is that CO_2 can be evaporated in the indoor air handling unit (AHU). The heat of vaporization of CO_2, at about 500 pounds force per square inch (psig) (about 40°F [4.5°C]), approximately 99.3 British thermal units per pound mass (Btu/lbm), is significantly higher than the heat capacity of water, approximately 20 Btu/lbm with a 20°F [11°C] temperature differential. CO_2 can provide up to five times the convection heat transfer per unit mass of pumped fluid, and this improvement in heat transfer can 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. This allows the system to be piped with small diameter (for example, 7/8″) high-pressure copper alloy piping, instead of welded steel or groove connected steel pipe that have higher material and associated labor costs. The savings are substantial when comparing any significant pipe length at about $19.92 per foot (ft) of installed copper pipe after markups (about $11.52/ft) to steel pipe (about $31.45/ft).

However, CO_2 requires high pressure rated components and there is an additional challenge with heating operation. The critical temperature of CO_2 (87.98°F [31.1°C]) is lower than the temperature typically delivered to heating coils. Therefore, space heating generally results in circulation of supercritical CO_2. Commonly available CO_2 compressors are typically designed for transcritical refrigeration cycles with low evaporating temperatures/pressures and a high-pressure lift. However, these parameters match poorly with the space heating operating conditions. Specifically, 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 system efficiency. Supercritical CO_2 has relatively high density, and this characteristic motivated previous EPRI work that investigated and verified the feasibility of pumping supercritical CO_2 with a high-pressure CO_2 pump (Robinson et al, 2023).

CHAPTER 2: Project Approach

The project approach was divided into three phases. First, the prototype system would go through a design and optimization process. Then, a prototype system would be constructed and evaluated at EPRI's laboratory. Finally, a total of five field systems would be tested in various California climate zones.

Project Specifics

Organization and Partners

As the prime recipient, EPRI was responsible for the overall project management as well as leading the technical approach. The grant was executed on June 15, 2020, with a reimbursable amount of $2,498,557 and $440,000 in match share from several parties, including San Diego Gas & Electric Company (SDG&E) and Southern California Edison (SCE).

The multidisciplinary project team combined strong, leading organizations with expertise in thermal system design and manufacturing. At Optimized Thermal Systems Research and Development (R&D), Paul Kalinowski and Dennis Nasuta led efforts to design the prototype heat pump system under guidance from EPRI and in collaboration with component manufacturers. Mayekawa USA, Inc. was a team partner during the proposal phase, with a scope to build and install the prototype heat pumps. However, due to internal reasons, Mayekawa USA, Inc. had to reduce its role as equipment supplier. Despite this diminished role, Troy Davis, Energy Group Manager, provided much technical guidance during the design phase.

Meetings

The project kick-off meeting took place on July 2, 2020. The meeting discussion reviewed the overall project goals and objectives, as well as the administrative logistics. The team also discussed the confirmation of project host sites and potential surveys for project benefits.

The first Critical Project Review meeting was held virtually on January 30, 2024. The EPRI team presented on the completed project tasks (prototype design and pumped supercritical CO_2 tests), current progress (laboratory prototype fabrication), and the next steps (lab testing and field evaluation).

Technical Advisory Committee

A Technical Advisory Committee (TAC) meeting webinar was held on January 29, 2021. The TAC members represented utilities and research/industrial professionals with relevant experience and guidance for the project's scope and direction.

Engineering Design and Modeling

The objective of the lab prototype design effort was to enable the implementation of a flexible laboratory device that allowed performance testing and optimization over the range of operating conditions. This prototype pushed the boundary of existing heat pump technology to overcome several technical hurdles related to low GWP heat pumps. One main technical difficulty was that CO_2 exhibits very efficient heat transfer properties with high pressures and relatively tight pressure-dependent temperature bands. In addition, while ammonia is an excellent refrigerant, it is not compatible with several engineering materials (notably, copper and its alloys due to corrosion) and care had to be taken when selecting equipment. The engineering design was accomplished with the assumptions listed in Table 1.

Table 1: Design and Model Assumptions

Air-Refrigerant heat exchanger (HX) Approach Temperature °F [°C]	27 °F [15 °C]
Plate HX Approach Temperature °F [°C]	9 °F [5 °C]
Superheat/Subcooling °F [°C]	9 °F [5 °C]
Compressor Isentropic efficiency [%]	70%
Typical Heating Mode Conditions (Indoor/Outdoor °F [°C])	70 °F [21.1 °C] / 35 °F [2 °C]
Typical Cooling Mode Conditions (Indoor/Outdoor °F [°C])	70 °F [21.1 °C] / 84 °F [29 °C]

Source: EPRI, 2021

Several iterations of performance modeling were carried out in parallel with the detailed design and selection of individual components. This modeling often consisted of multiple platforms and vendor-specific simulation tools. Detailed component data were unified into consistent models to allow for the estimation of system-level performance across a range of operating conditions.

Prototype Evaluation

The heat pump prototype was evaluated in a laboratory setting prior to field deployment. The complexity and size of the system required that it be constructed outdoors. While the ideal scenario would have tested this prototype in conventional environmental chambers for controlled conditions, the 20 RT capacity was too large for the chambers at the EPRI Knoxville laboratories. Therefore, the outdoor unit was located in the parking lot outside the building and the indoor unit within the lab space. This allowed the team to evaluate the controls and operation of the CO_2 loop, which provided critical information on the feasibility of the system. The indoor space was conditioned by the building HVAC system, which means it maintained a steady range of temperatures and humidity. This allowed the system to operate in a pseudo laboratory setting, where only the outdoor conditions were uncontrolled. This also provided an additional safety measure, as the ammonia charge could be isolated outdoors with minimal risk for releasing indoors. Figure 2 and Figure 3 show the assembled heat pump at EPRI's facilities.

Figure 2: Assembled Heat Pump Prototype at EPRI Laboratories

Source: EPRI, 2024

Figure 3: Indoor Components of the Heat Pump

Source: EPRI, 2024

Field Demonstration

The field production units had to satisfy several requirements when installed at the host sites. First, they had to be designed properly to satisfy the cooling and heating loads in the buildings. Second, the installed system had to meet safety requirements based on codes and standards. Third, the safety of the equipment operators had to be ensured.

Thermal Requirements

The primary purpose of any space conditioning HVAC equipment is to meet the thermal load. The thermal load (heating or cooling) is determined based on a number of factors, including climate zone, number of occupants, solar gain, and air infiltration. The industry rule of thumb is about 1 RT of capacity for every 400 square feet of building area, so the 20 RT system in this project should have been able to serve roughly 8,000 square feet. Additionally, this system had variable capacity compressors, pumps, and fans, allowing the total output (heating or cooling) to be modulated to precisely match the instantaneous thermal load, unless the load fell below the minimum capacity dictated by the turndown ratio of the equipment.

Safety Requirements

This novel heat pump system had to meet several safety requirements, largely due to the hazardous nature of the refrigerant. Ammonia, a B2L refrigerant, has exposure limits from regulatory bodies that are as follows (New Jersey Department of Health, 2016):

- OSHA – The legal airborne permissible exposure limit is 50 parts per million (ppm) averaged over an 8-hour work shift.
- National Institute for Occupational Safety and Health (NIOSH) – Recommended airborne exposure limit is 25 ppm averaged over 10 hours and 35 ppm not to be exceeded during any 15-minute work period.
- American Conference of Governmental Industrial Hygienists – The threshold limit value is 25 ppm averaged over an 8-hour work shift and 35 ppm as a short-term exposure limit.

This system design took advantage of the secondary fluid loop to reduce charge on the primary cycle and isolated the ammonia in the outdoor unit. Therefore, leaks were very unlikely to affect indoor occupants. The outdoor package was installed away from building openings (doors and windows) to further mitigate hazards to people in the event of a leak. Stainless steel components were used instead of copper alloys to prevent reaction between ammonia and copper components.

CO_2 is a high-pressure secondary fluid and potential leaks into the indoor environment presented health risks. The primary danger of a CO_2 leak is the displacement of oxygen in the indoor air. CO_2 exposure limits by OSHA and NIOSH are as follows:[1]

- Short Term Exposure Limit: 30,000 ppm for 15 minutes
- Long Term Exposure Limit: 5,000 ppm average over 8 hours

[1] https://www.co2meter.com/blogs/news/dangers-of-co2-what-you-need-to-know

The exposure risks can be mitigated by proper installation and selecting components with appropriate pressure ratings. Refrigerant (ammonia and CO_2) sensors were installed in the indoor space so that any refrigerant leak could be immediately detected. There were also structural and electrical requirements associated with deploying the field units. These were similar to the requirements of conventional HVAC equipment.

Operation and Maintenance

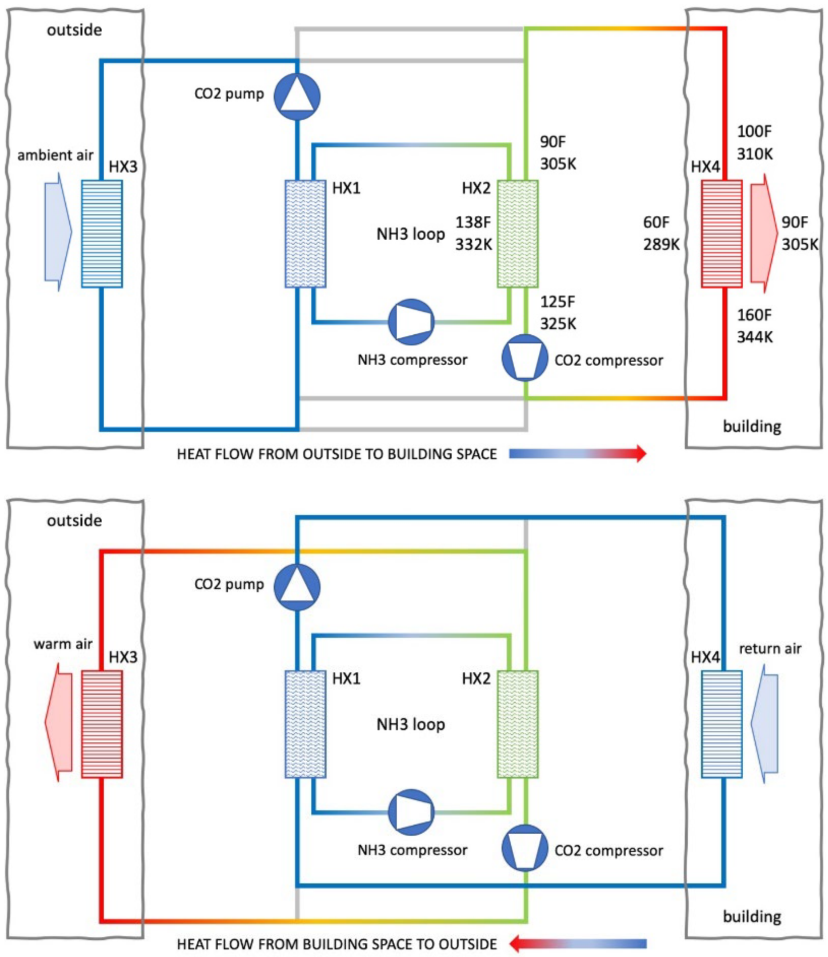
The field units were intended to be deployed with the appropriate operation manuals, and equipment users would be properly trained on the operation schedule. The installation of the units would be verified by an equipment assurance, inspection, and certification company to ensure they met UL (Underwriters Laboratories) field requirements. Maintenance of the field units would be covered by the manufacturing company that assembled the field units.

CHAPTER 3: Results

Engineering Design

The original novel heat pump design proposed by EPRI is illustrated in Figure 4. The system consisted of two CO_2 loops that transferred heat between the indoor and outdoor spaces, while a central ammonia system provided the majority of heat pumping work. In heating mode, a liquid pumped CO_2 loop extracted air from the outdoor space and transferred it to the evaporator of the ammonia cycle. Heat was rejected from the ammonia cycle condenser into evaporating subcritical CO_2, which was then compressed to a supercritical state where heat was rejected into the indoor space. In cooling mode, piping changes switched the roles of the CO_2 loops: heat was absorbed from the indoor space by a pumped liquid CO_2 loop and rejected into the ammonia cycle's evaporator. Then heat was rejected from the ammonia condenser into evaporating CO_2, which was compressed (subcritical or transcritical depending on ambient condition) to reject heat through the outdoor heat exchanger to the ambient space.

Figure 4: Prototype Schematic with Central Ammonia System and Two CO_2 Loops



Source: EPRI, 2019

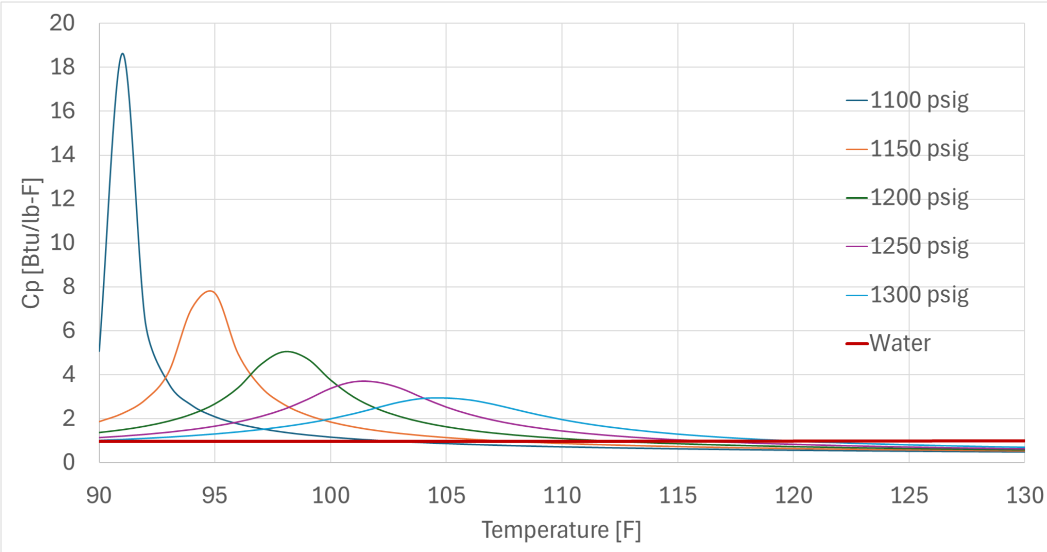
Configurations like this are used in larger capacity "ultra-low charge" systems where ammonia charge must be minimized. By operating the ammonia cycle with two fluid-to-fluid plate heat exchangers, the ammonia charge can be kept very small while using nontoxic CO_2 in the direct expansion air-to-refrigerant heat exchangers. One challenge to the operation of such a system is the management of CO_2 refrigerant quantity (charge) when operating between different modes where a relatively large quantity of cold liquid CO_2 is required in cooling mode, but that same quantity may be suboptimal for operation in transcritical heating mode. A means to control system pressures through management of refrigerant charge is necessary for successful operation.

The main disadvantage to this design is the additional heat exchange step. Each time heat is transferred across an interface, thermodynamic irreversibilities and losses occur due to the non-ideal nature of real heat exchangers. Thermodynamic irreversibility is the process of transferring heat at a temperature difference. Although the indoor CO_2 loop is required from a safety standpoint, the outdoor CO_2 loop is not strictly necessary and represented a significant inefficiency in the design.

To eliminate the additional thermodynamic losses associated with the outdoor CO_2 loop, several modifications to the system design were made. First, the ammonia cycle had to become reversible (directionally), such that its direct-expansion outdoor heat exchanger could act as evaporator absorbing heat in heating mode and a condenser rejecting it in cooling mode. This ability was accomplished simply with a reversing valve as is commonly employed in heat pumps. The indoor CO_2 cycle still required a compressor to operate transcritically in heating mode and could have continued to use a pump for cooling mode, but further simplicity was accomplished by making the CO_2 cycle a reversible vapor compression cycle, using the compressor in both heating and cooling modes. This configuration improved the heating mode performance relative to the original design because of the elimination of the outdoor CO_2 loop. In cooling mode, the use of a subcritical CO_2 vapor compression cycle was less efficient than a liquid pumped design; however, the intent was to modulate compressor speeds to minimize the CO_2 cycle pressure lift (and compressor power) by doing the majority of heat pumping work with the ammonia cycle.

In heating mode, two key variables had to be considered: the cascade heat exchange temperature and the CO_2 gas cooler pressure. The question of gas cooler pressure is linked to the thermophysical properties of supercritical CO_2, which has a unique spike in specific heat capacity at the transition temperature for a given pressure, shown in Figure 5. Operating at too low a pressure relative to the temperature at which heat must be rejected fails to leverage this heat capacity, and operating too high requires excess compressor power.

Figure 5: Variation of Supercritical CO_2 Heat Capacity

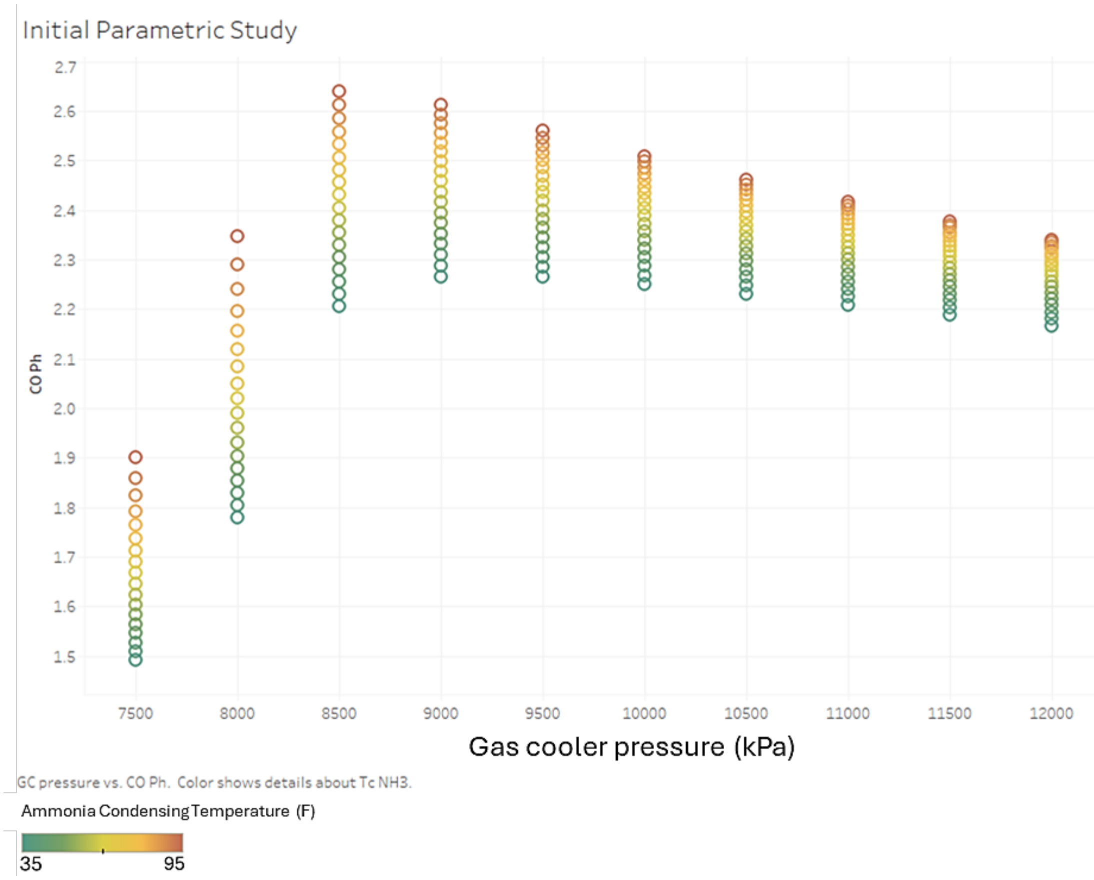


Source: EPRI, 2021

For each operating temperature, an optimum gas cooler pressure exists. In this case, when rejecting heat with a CO_2 outlet temperature of 97°F [36°C], the gas cooler pressure was found to be optimal around 8,500 kilopascals. Note: improved heat exchanger designs can operate with lower outlet temperatures and gas cooler pressures.

Figure 6 shows the result of a parametric study performed in Engineering Equation Solver; it reveals both that the optimum gas cooler pressure for this scenario was around 8,500 kilopascals and that the highest ammonia condensing temperature yielded the highest efficiency. This result is intuitive, as ammonia's thermodynamic efficiency is greater than CO_2's even given the same compressor isentropic efficiencies: the total system coefficient of performance is highest when the ammonia cycle performs the majority of heat pumping work and the CO_2 compressor is left to circulate fluid with minimal pressure lift.

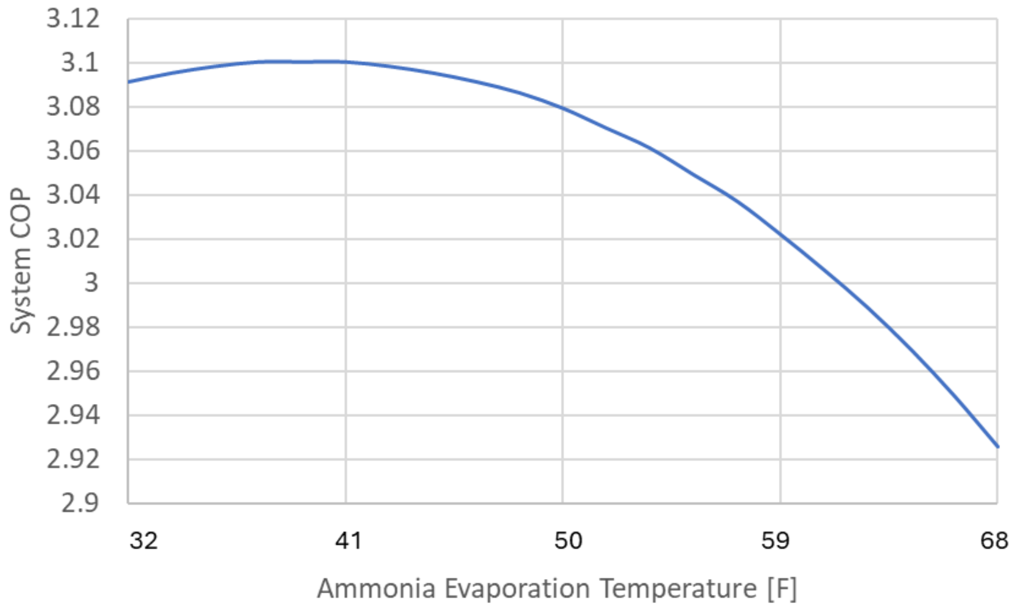
Figure 6: Heating Mode Parametric Study



Source: EPRI, 2021

In cooling mode, the result was much the same. Assuming a 70°F [21°C] indoor return air temperature and 27°F [15°C] approach temperature, CO_2 will evaporate at 43°F [6°C] to provide cooling. The CO_2 cycle must have some pressure lift (condense greater than 6°C) and the ammonia evaporator will be about 9°F [5°C] colder than the CO_2 condenser as it exchanges heat through the plate heat exchanger. Figure 7 shows the result of these Engineering Equation Solver simulations: optimum performance occurred when the ammonia evaporating temperature was around 41°F [5°C] (CO_2 condensing temperature was then 50°F [10°C], evaporating temperature was 43°F [6°C]). Again, the specifics of these points of optimality were subject to change with different heat exchanger performance and operating conditions. The trend was clear: maximizing the use of the ammonia cycle while minimizing the pressure lift/work of the CO_2 cycle is the pathway to maximum efficiency.

Figure 7: Cooling Mode Parametric Study Results



Source: EPRI, 2021

Updated System Design

Through the development of this design, the limitations of existing products clashed with the expectations from the initial theoretical modeling work in several instances. As is clear from the previous simple analyses, heat exchanger performance (approach temperatures) is critical to system efficiency and numerous iterations on heat exchanger (HX) design were carried out to push performance to its reasonable limits. However, compressor designs offer less flexibility and greater limitations. Two issues with available CO_2 compressors that limited performance below the initially considered theoretical levels included:

1. Transcritical CO_2 compressors with high suction pressures are not common or readily available
2. The efficiency of CO_2 compressors at very low pressure lift is quite poor.

In the case of the former, the best CO_2 compressor candidate cannot operate with evaporating temperatures higher than 59°F [15°C], which means the share of CO_2 work will be higher, and the total efficiency lower than the ideal case modeled. The latter finding means that efforts to limit CO_2 cycle power by minimizing pressure lift will be met by lower isentropic efficiency because the compressor is not designed for such low-pressure lift operation, further limiting system efficiency.

As weighted efficiency calculations were performed, concern arose that the system may not reach performance targets, especially in cooling mode, due to the limitations of available components. Having optimized the heat exchangers to their realistic technical potential, the remaining course of action was to return to a pumped liquid arrangement for cooling mode. This configuration should achieve higher efficiency and has been demonstrated in a prior EPRI research project that explored low GWP refrigerants for space cooling.

By constructing the laboratory prototype system with both a pump and compressor, several operating modes could be tested in the lab to determine their performance. The system could be tested using only a compressor for both modes to determine: if efficiency is adequate to justify the simplicity; higher efficiency can be tested with pumped cooling mode; and if it's possible to perform rudimentary tests with pumped supercritical CO_2, which could ultimately open opportunities for a pump-only CO_2 cycle.

Additionally, ammonia compressors in the capacity range required for this project (10-20 RT) that are semi-hermetic are relatively uncommon. A product from Mayekawa was selected, but its capacity was considerably greater than the original 10 RT requirement. Even when reducing the number of active cylinders and compressor speed, it was determined that the system would need to be redesigned for a higher capacity, nominally 20 RT. While the compressor was still oversized for the application, it was expected to operate at 20 RT with the ability to turn down its capacity by at least 50 percent in part-load conditions. To accommodate this change, the system was designed to use two indoor units, nominally 10 RT each, that could operate independently in the conditioned space(s). Figure 1 in Chapter 1 shows a schematic of the prototype system, which still consisted of a reversible ammonia outdoor cycle with direct-expansion outdoor coil but was coupled to an indoor CO_2 cycle with two parallel air handlers. The system can be operated using the CO_2 compressor in heating mode and using the liquid pump in cooling mode. This formed the basis of the design, but alternate configurations could be considered: 1) adding a bypass from discharge to the receiver tank to control pressure in heating mode, 2) reverting to a reversible CO_2 cycle using compressor in cooling mode, or 3) pumping supercritical CO_2 in heating mode rather than compression.

Performance Modeling

Performance at a single ambient operating condition and capacity was straightforward to evaluate through modeling, though it did not reveal real-world annual performance where systems operate for many hours at lower partial-load conditions. These simulations were still valuable for evaluating system performance and contextualizing it amongst competing products. Typically, Air Conditioning, Heating, and Refrigeration Institute (AHRI) standards are used to establish these rating conditions. In the case of this unit, the 340/360 standard is relevant to its capacity range and function (AHRI, 2019).

Modeling was accomplished by harmonizing vendor-supplied heat exchanger and compressor performance data into a simple model. The indoor CO_2 and outdoor ammonia tube-fin heat exchangers were modeled in CoilDesinger® 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 2 and Table 3.

Table 2: Modeled Rating Conditions at 70 kW Capacity

Condition	Temperature (outdoor/ indoor) (°F) [°C]	Ammonia Compressor Power [kW]	CO_2 Compressor /Pump Power [kW]	Fan Power [kW]	EER [Btu/hr-W] / COPh [W/W]
Cooling – *vapor compression*	95 °F [35 °C] / 80 °F [26.7 °C]	13.4	6.1	6.5	9.2
Cooling - *pumped*	95 °F [35 °C] / 80 °F [26.7 °C]	17.5	0.2	6.5	9.9
Heating – *high temperature*	47 °F [8.3 °C] / 70 °F [21.1 °C]	5.7	11.7	6.5	2.9
Heating – *low temperature*	17 °F [-8.3 °C] / 70 °F [21.1 °C]	10.6	11.7	6.5	2.4

Source: EPRI, 2021
EER = energy efficiency ratio; COPh = coefficient of performance, heating; W = watt

Table 3: Modeled Saturation Temperatures

Condition	CO_2 Evaporation Temperature (°F) [°C]	CO_2 Condensation Temperature (°F) [°C]	Ammonia Evaporation Temperature (°F) [°C]	Ammonia Condensation Temperature (°F) [°C]
Cooling – *vapor compression*	50 °F [10 °C]	68 °F [20 °C]	59 °F [15 °C]	116 °F [47 °C]
Cooling - *pumped*	50 °F [10 °C]	50 °F [10 °C]	41 °F [5 °C]	116 °F [47 °C]
Heating – *high temperature*	59 °F [15 °C]	-	32 °F [0 °C]	68 °F [20 °C]
Heating – *low temperature*	59 °F [15 °C]	-	2.1 °F [-16.6 °C]	68 °F [20 °C]

Source: EPRI, 2021

Under the 340/360 standard, four test points are defined along with a weighting formula to compute the integrated energy efficiency ratio (IEER). The efficiencies at these four temperature/capacity levels can be simulated and aggregated into a single figure. However, two limitations persist:

1. No such metric is defined for part-load seasonal heating efficiency.
2. The accuracy of this metric is often questioned since all locations and buildings will have differing climate and usage profiles resulting in varying seasonal performance.

Furthermore, controls design decisions had to be made to determine the optimal balance of fan and compressor speeds, which can be modulated to alter the heat exchanger performance (approach temperatures) and power consumption. Such controls optimization was premature at this stage of the development process and should have been carried out in parallel with laboratory prototype testing. Nevertheless, some non-optimal control decisions were evaluated in the context of the IEER rating, the results of which are shown in Table 4. The IEER made use of significant part-load fan power savings from electronically commutated motors but still remained unoptimized from a controls standpoint. Although the controls were not rigorously optimized, it was assumed here that compressors and fans could be turned down to 25 percent load ideally, and this calculation did not account for losses from variable frequency drives (VFD) or cycling. Table 5 displays the modeled saturation temperatures of the cycle in cooling mode for the four IEER conditions. It was evident that as building load decreased, the heat exchanger performance improved as seen in the reduced approach temperatures (higher evaporation temperatures, lower condenser temperatures). The cascade plate heat exchanger's performance was held at a 5K approach temperature difference due to a lack of part-load performance data; it was expected that this approach temperature would be considerably reduced in part-load conditions, which would further improve efficiency.

Table 4: Design and Model Assumptions

Condition (T_{amb} (°F) [°C] / %load)	Outdoor Fan Load [%]	Indoor Fan Load [%]	Ammonia Compressor Power [kW]	CO_2 Compressor /Pump Power [kW]	Fan Power [kW]	EER [Btu/hr-W] / COP [W/W]
A (95 °F [35 °C] / 100%)	100	100	17.5	0.2	6.5	9.9
B (81.5 °F [27.5 °C] / 75%)	70	75	9.2	0.2	3.1	14.4
C (68 °F [20 °C] / 50%)	50	50	4.1	0.2	1.7	19.8
D (65 °F [18.3 °C] / 25%)	25	25	1.4	0.2	0.9	23.7
					IEER	**16.7**

Source: EPRI, 2021
T_{amb} = ambient temperature

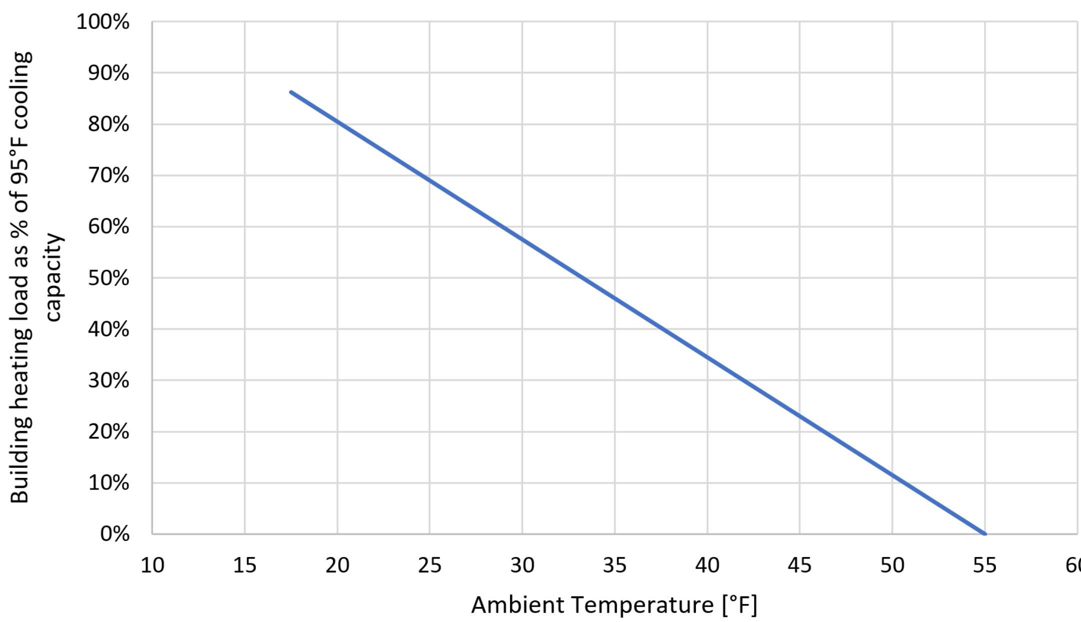
Table 5: Saturation Temperatures in IEER Cooling Conditions

Condition	CO_2 Evaporation Temperature (°F) [°C]	CO_2 Condensation Temperature (°F) [°C]	Ammonia Evaporation Temperature (°F) [°C]	Ammonia Condensation Temperature (°F) [°C]
A	50 °F [10 °C]	50 °F [10 °C]	41 °F [5 °C]	116.6 °F [47 °C]
B	53.6 °F [12 °C]	53.6 °F [12 °C]	44.6 °F [7 °C]	100.4 °F [38 °C]
C	55.4 °F [13 °C]	55.4 °F [13 °C]	46.4 °F [8 °C]	84.2 °F [29 °C]
D	60.8 °F [16 °C]	60.8 °F [16 °C]	51.8 °F [11 °C]	78.8 °F [26 °C]

Source: EPRI, 2021

To estimate more realistic part-load conditions and simulate seasonal heating performance, it was necessary to make assumptions about the system controls and the building load. First, typical meteorological year 3 (TMY3) weather data for Riverside, California was taken and binned in 5°F (2.8°C) increments. Then the heating building load calculation from the uniform test method for air conditioners and heat pumps (U.S. DOE, 2023) was used to estimate building load. This method applies to smaller consumer equipment but was used in this case because no equivalent seasonal heating performance metric exists for larger equipment under the 340/360 (Commercial and Industrial Unitary Air-conditioning and Heat Pumps) or 1230 (variable refrigerant flow equipment) standards; they require rating only at a single coefficient of performance value. The equation for this heating load line is given below in Figure 8 for climate region IV.

Figure 8: Heating Load Line for Region IV



Source: EPRI, 2021

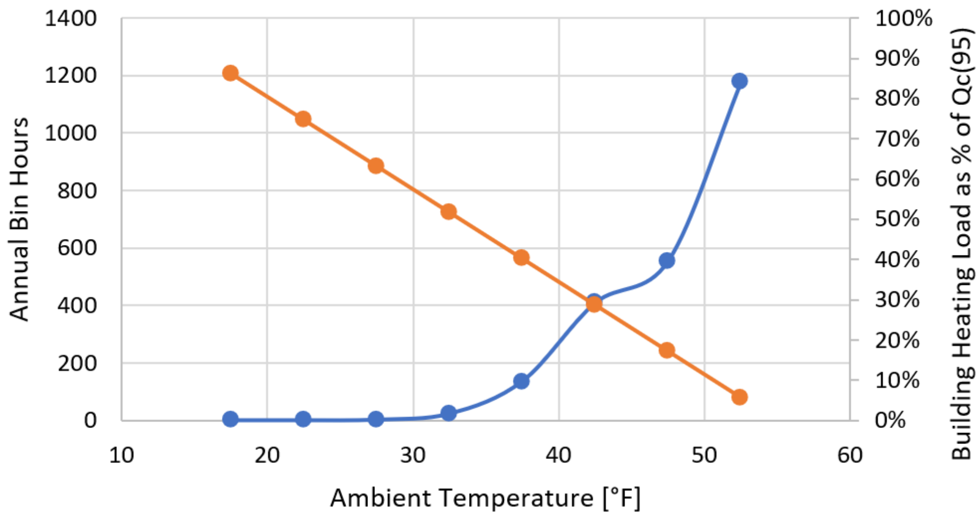
It implies that "full load" heating capacity (equivalent to 95°F [35°C] cooling capacity) is required at 11.5°F (-11.4°C). It is recognized that this approach will not be applicable for all building types, but true heating loads cannot be determined without considerable effort in auditing and modeling specific buildings, which can have extremely different loads; this approach provided a simple relation that expresses increasing loads with decreasing ambient temperatures.

$$BL\left(T_j\right) = \frac{T_{zl} - T_j}{T_{zl} - 5°F} * C * Q_c(95°F),$$

Where *BL* is Building Load, T_j is the temperature of a given bin, T_{zl} is the temperature at which there is zero heating load, *C* is a coefficient assigned regionally, and Q_c is the rated cooling capacity at 95°F (35°C).

Figure 9 shows the hours spent annually at each bin temperature in Riverside, California. The location is mild relative to much of the United States, with a large majority of hours above freezing temperatures in part-load conditions with loads less than 50 percent of the rated cooling load. This climate is highly favorable for heating performance because of the low number of hours in frost-developing conditions and the higher efficiency of part-load operation.

Figure 9: Heating Load and Hours per Temperature Bin for Riverside, California



Source: EPRI, 2021

A simplified annual performance calculation was carried out for the Riverside location by iteratively simulating heat exchanger models with compressor performance maps to reach convergence in eight bin temperatures of 5°F (2.8°C) increments. For each bin, the team identified the required saturation temperature for each heat exchanger to absorb/reject the specified amount of heat. For a simpler strategy, fan air flow rates were assumed proportional to percentage of nominal load. It was also assumed the system could turn down ideally to low part-load capacities. In actual practice, compressor limitations may limit the system's ability to reduce speed to match very low loads and instead cycle on/off at higher capacities with lower part-load efficiency. These results represent a likely upper limit of system efficiency, but it is clear that the mild climate requires lesser heating loads than cooling loads, meaning a system sized for the nominal cooling case will often be oversized (able to perform with greater efficiency) in many heating conditions. Table 6 and Table 7 contain summary results from these simulations and reveal a potential heating seasonal performance factor as high as 10.45 for this location.

Table 6: Annual Heating Calculation for Riverside, California

Bin avg temp	Frequency	Building Load	Total Power	EER
°F [°C]	Hours	kW	kW	Btu/hr-W
17.5 °F [-8.1 °C]	0	60.38	23.7	8.67
22.5 °F [-5.3 °C]	0	52.33	19.3	9.27
27.5 °F [-2.5 °C]	2	44.28	15.1	9.97

Bin avg temp	Frequency	Building Load	Total Power	EER
32.5 °F [0.3 °C]	23	36.22	11.7	10.58
37.5 °F [3.1 °C]	137	28.18	8.6	11.16
42.5 °F [5.8 °C]	410	20.13	5.9	11.57
47.5 °F [8.6 °C]	556	12.08	3.6	11.30
52.5 °F [11.4 °C]	1180	4.03	1.7	7.89
				Heating seasonal performance factor: 10.45

Source: EPRI, 2021

Table 7: Detailed Performance Results, Riverside, California

W fan indoor	W comp CO_2	W fan outdoor	W comp NH_3	Tevap NH_3
kW	kW	kW	kW	°F [°C]
1.76	9.71	3.06	9.22	4 °F [-15.56 °C]
1.55	8.27	2.10	7.33	9 °F [-12.78 °C]
1.36	6.78	1.33	5.68	14 °F [-10.00 °C]
1.18	5.47	0.74	4.30	18 °F [-7.72 °C]
1.01	4.19	0.34	3.07	22.2 °F [-5.44 °C]
0.87	2.99	0.12	1.95	27.2 °F [-2.67 °C]
0.74	1.80	0.08	1.03	32.2 °F [0.11 °C]
0.63	0.60	0.23	0.29	38.1 °F [3.39 °C]

Source: EPRI, 2021
Tevap = temperature of evaporation

Small Scale Supercritical CO_2 Test Loop

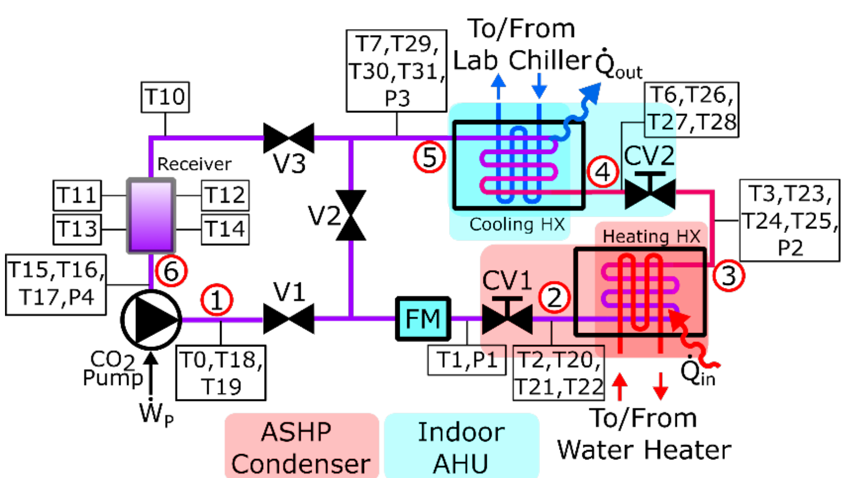
Prior to the full prototype testing, a simpler, smaller CO_2 test loop was used to investigate the CO_2 pump's ability to circulate supercritical fluid, shown in Figure 10. The design of the experimental setup was based on the operating conditions of the secondary loop, using two heat exchangers to absorb and reject heat from the CO_2 loop to the laboratory hot/chilled water loops. Stainless-steel needle valves were used to simulate a pressure drop along the heat exchanger. A total of 2.2 kilograms of CO_2 were charged into the test loop based on the design operating conditions from the modeling analysis.

Figure 10: Supercritical CO_2 Pump Test Loop as Built

Source: EPRI, 2023

Figure 11 indicates the measurements taken and their locations. Temperature was measured using T-type thermocouples and two NI USB-9213 thermocouple modules. Thirty-two temperature measurements were taken, with important measurements made with multiple thermocouples (in and out of the heat exchanger, for example). The thermocouples were taped to the pipe and insulation was added to ensure an accurate reading (reducing losses to the environment). Pressure was measured by digital and analog pressure gauges. The four digital sensors output a 0–10 volts of direct current that was read by an NI USB-6001 to convert the output signal into the ModBus protocol, which is typical building management systems use for data acquisition. The flow meter and power meter provided a Modbus interface that the data acquisition computer could query.

Figure 11: Schematic of Supercritical CO_2 Test Loop with Instrumentation Location



Source: EPRI, 2023

The first test was to verify the test loop could circulate supercritical CO_2. Initially, the CO_2 was a two-phase mixture at room temperature. The VFD was operated at a low frequency to circulate the liquid CO_2 while adding heat from the heating heat exchanger. The temperature and pressure increased while the density decreased. When enough heat was added, the temperature and pressure surpassed the respective critical points. At this point, the density stabilized and the VFD was increased. This test showed that the pump could circulate supercritical CO_2 and testing could continue, allowing the team to study how the pump performed in different conditions. A test matrix was set up as summarized in Table 8, where each variable was controlled and varied. For example, pump inlet and heating HX outlet temperature, and VFD frequency were held constant, while the control valve 1 and control valve 2 positions varied.

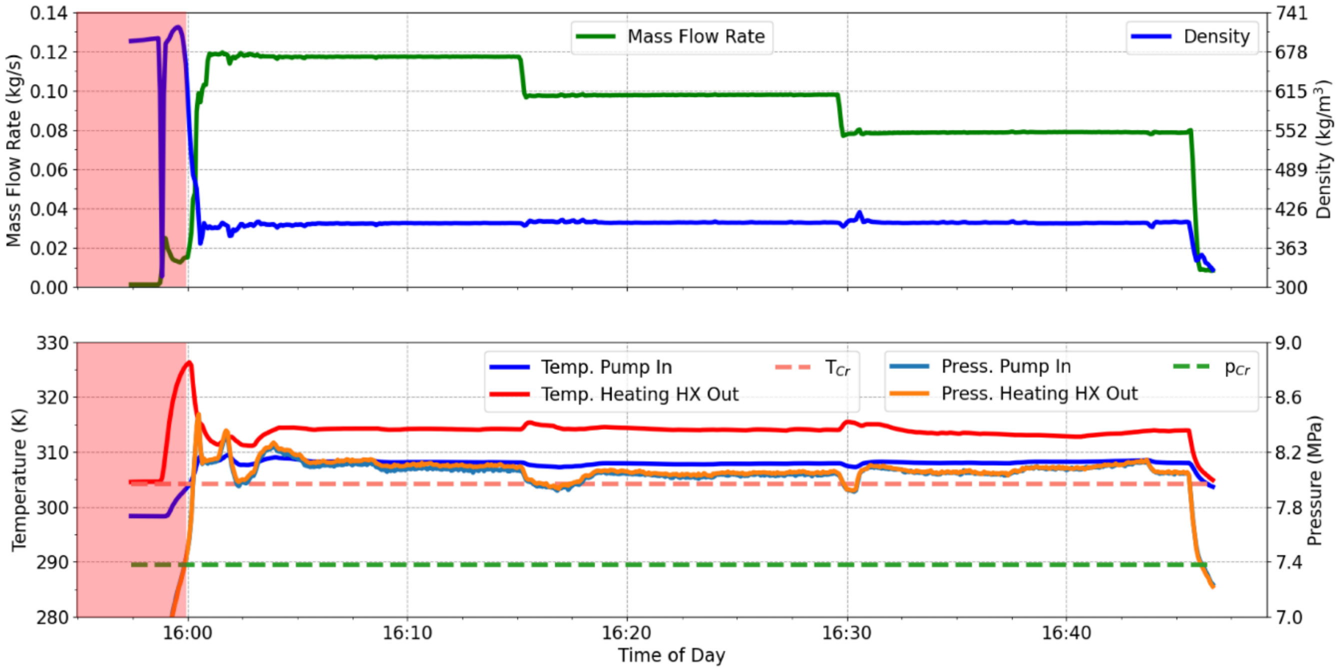
Table 8: Supercritical Loop Test Matrix

Pump Inlet Temperature (°F [°C])	95, 100, 105 [35, 37.8, 40]
Heating HX Outlet Temperature (°F [°C])	105, 110, 115 [40, 43, 46]
VFD Frequency [Hz]	40, 50, 60
Control Valve 1 and Control Valve 2 Position	100% open, 66% open, 33% open

Source: EPRI, 2023

Figure 12 is an example of a test performed with the pump inlet temperature set to 95°F (35°C) and the heating heat exchanger set to 105°F (40°C). Initially the CO_2 was subcritical as indicated by the transparent red rectangles. As heat was added through the heating heat exchanger the temperature and pressure increased until the critical point was passed. Then the cooling heat exchanger valve was fully opened to stabilize the temperature and pressure. The valves on the heating and cooling water adjusted such that the test conditions were met: temperature setpoints for pump inlet temperature and heating heat exchanger outlet. The VFD was set to 60 Hertz (Hz) for the initial test, which was performed for 15 minutes after reaching quasi-steady state conditions. The pump inlet and heating heat exchanger outlet temperatures were monitored during the test period and adjusted. The VFD was then adjusted to the next speeds (50 Hz and 40 Hz) and the valves adjusted for temperature setpoint, followed by another 15-minute period of steady state conditions. As the temperature setpoints were varied, the density changed slightly between tests, which gave an idea of how the CO_2 pump performed with a more or less dense fluid.

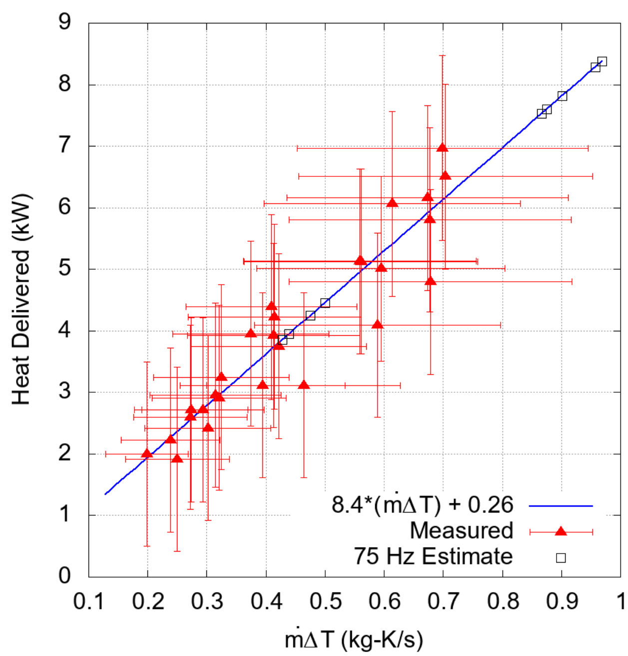
Figure 12: Sample Results of Supercritical CO_2 Pumping



Source: EPRI, 2023

Figure 13 shows the amount of heat that the pump delivered as a function of the product of the mass flow rate and temperature difference between the heating HX output and the pump inlet temperature. The rate of heat delivered behaved as expected, increasing as the mass flow rate and/or temperature difference increased.

Figure 13: The Heat Transfer Rate to the Cooling Heat Exchanger as a Function of Mass Flow and Temperature Change



Source: EPRI, 2023

The results from the test loop show that circulation of supercritical CO_2 is possible with an appropriately pressure rated/designed CO_2 pump. The mass flow rate varied between 0.07 kilograms per second at 40 Hz to about 0.120 kilograms per second at 60 Hz. The electric power provided to the pump for these flow rates was between 150 W and 200 W. On the other hand, the CO_2 compressor selected for the project was estimated to draw about 6.5 kW to operate (running at part-load). This was a significant opportunity for energy savings if the pump was capable (instead of the compressor). The test loop results indicated that multiple pumps would be needed to match the project requirements on heating capacity. The maximum heating capacity of a single pump is 7 kW (about 2 RT or 24,000 BTU/hr), so it would require at least five pumps to achieve 10–20 RT of capacity. If five pumps were being used at 200 W each, the total power draw would be about 1 kW, still significantly less power consumption than the compressor but increasing the initial cost of the system. However, from the results presented, there are opportunities for increasing the delivered heat. These include increasing the temperature difference, lowering the mass charge of the system to allow for higher temperatures at equivalent pressures, and running the VFD at frequencies greater than 60 Hz.

Bill of Materials

The main components of the heat pump were selected based on the component availability, system requirements and feedback from the Optimized Thermal Systems team and manufacturers. The full bill of materials can be found in Appendix A.

- Ammonia compressor: (Mayekawa N4KHM30) A review of ammonia compressors in the relevant capacity range turned up extremely limited choices for semi-hermetic options. Open type ammonia compressors are more commonly found in industrial applications, but the propensity for leaks was seen as a concern for this space conditioning application. The only semi-hermetic compressor close to the desired size range was offered by Mayekawa. EPRI's experience and working relationship with Mayekawa also meant that manufacturer support would be available. The compressor was selected by using the test data provided by Mayekawa for the required capacities as well as evaporating and condensing temperatures in the heating and cooling modes. The compressor required liquid cooling and Mayekawa assisted in the selection of the relevant components. Even the smallest compressor available was larger than the original target 10 RT capacity. For this application, during most hours, the compressor would be operated on only two of its four cylinders. The design of the other components of the system changed to accommodate the larger capacity of the available compressor and components were sized for nominally 20 RT.
- CO_2 compressor: (Dorin CD 2000H/OP) The compressor was selected by using the Dorin Software for the required cooling and heating capacity, evaporating temperature and gas cooling pressure. The compressor had an oil pump to ensure adequate lubrication of the moving compressor components. Evaporating temperature was a limitation for CO_2 compressors, which typically operate at lower temperatures. The selected compressor was able to operate with evaporation temperatures as high as

59°F (15°C), which was not as high as the ideal case initially modeled but was higher than other competing products.

- CO_2 pump: (Hy-Save 820-DS-050-VSD-B) The pump for the CO_2 cycle was selected based on the required refrigerant flow rate and pressure lift required in the cooling mode. Pressure drop in heat exchangers and piping was verified through vendor software and independent modeling. Laboratory evaluation also showed that the pump could circulate supercritical CO_2, providing an alternative to the CO_2 compressor for heating operation.
- Cascade/plate heat exchanger: (Alfa Laval AXP52 AN-94H) The brazed plate heat exchanger was sized by the manufacturer using the specified evaporating and condensing temperatures for cooling and heating modes. The following criteria were used for the heat exchanger selection:
    - Capacities in cooling heat modes
    - Approach temperature (9°F [5°C] or lower)
    - Corrosion resistance for ammonia (fusion-bonded, 100 percent stainless steel components)
    - High operating pressure of CO_2 (110bar/1595psi, external frames)
- Outdoor unit/ammonia coil: (Colmac Coil A+OV12I-32-66-310.0C-1-0300L-ACD-5B) The outdoor unit (fan and air coil) was selected by the manufacturer by using specified capacities, the ammonia evaporating and condensing temperatures in cooling and heating modes. The fans of the outdoor unit had variable speed drives to conserve energy by varying the fan speed. The tubes were made of stainless steel and fins were aluminum. A key concern in selecting the outdoor unit was the ability to operate in both heating and cooling modes. The selected unit was originally designed as a condensing unit with manifolds/headers for refrigerant distribution. Because the unit had to operate as an evaporator in heating mode, uniform distribution of entering two-phase refrigerant was critical; the vendor confirmed that a custom version of the coil could be manufactured with a distributor and feeder tubes for this purpose. The relatively low fin density (6.5fins per inch) would also be suitable for low temperature frosting conditions.
- Indoor unit/CO_2 coil: (MagicAire HCA40AAAAAAAGABAEBCCCABAM) The air handler was selected by using the Magic Aire selection software[2] for the specified air flow rates (400 cubic feet per min/ton). The blowers in the air handers had variable speed drives to conserve energy by varying the blower speed. The air coils were selected by using the Super Radiator Coils app[3] for the specified CO_2 evaporating or gas cooling temperatures as well as air handler dimensions. The tube was made of a copper alloy (Unilloy) for a high operating pressure (120bar/1,740psi).

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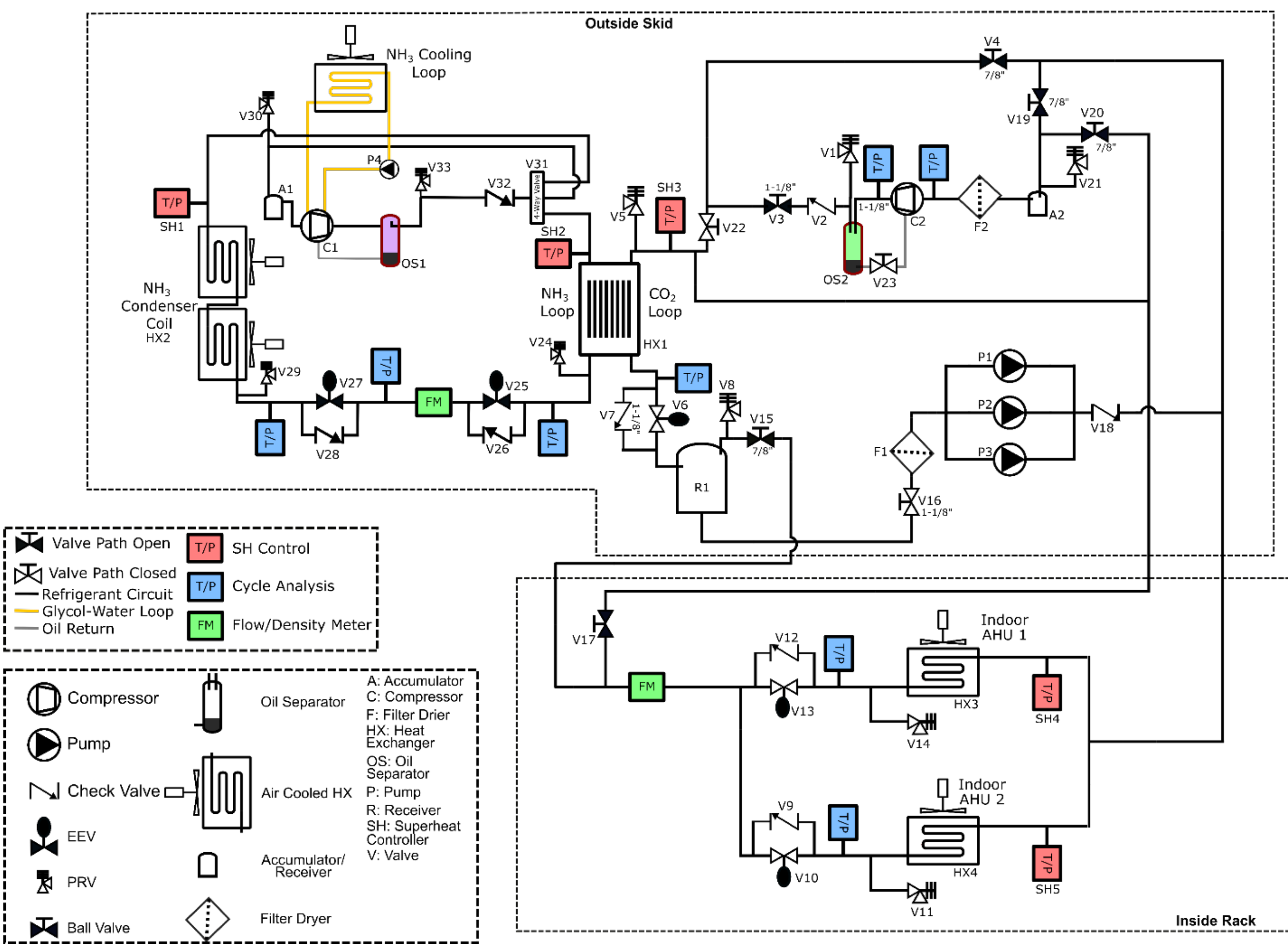
[2] http://www.magicaire.com/software/download-program/

[3] https://apps.superradiatorcoils.com/

Figure 14 shows the schematic diagram of the heat pump prototype. Each circuit had a compressor, accumulator, and oil separator. The oil from the oil separator was returned to the compressors by gravity. The speed of each compressor was controlled by frequency inverters. A four-way valve was used to direct the ammonia flow depending on the operation mode. Electronic expansion valves (EEV) were used to reduce refrigerant pressure. The EEVs were controlled by using EEV controllers as well as temperature and pressure sensors at the evaporator outlets. A single tube-fin coil with two variable speed fans was used to exchange heat with the ambient air. In the CO_2 circuit, two air handers with variable speed blowers and tube-fin coils were used to exchange heat between the refrigerant and indoor air. The air handers were located in a different section of the building for a uniform air distribution and to avoid any duct installation or reduce duct runs. The EEVs were located in the air handers. In the heating mode, the compressor was used to increase the temperature and pressure of the CO_2. In the cooling mode, the subcooled CO_2 was pumped from the liquid receiver to the indoor coils. All heat pump components except the air handlers were attached to a steel rack. The electrical drawings for the prototype heat pump are in Appendix D.

Figure 14: Schematic of the Laboratory Prototype



Source: EPRI, 2023

Prototype Construction and Commissioning

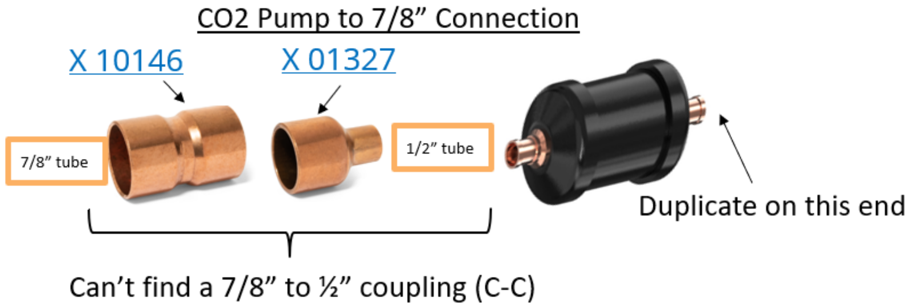
The components identified in the bill of materials were procured, either directly from the manufacturer, their preferred distributor, or special ordered through a construction equipment distributor. Due to project partner, Mayekawa, no longer participating in the demonstration, the prototype was constructed in EPRI's Knoxville laboratories. The 20 RT capacity and physical size of the prototype heat pump were too large for the psychrometric (10 RT nominal) chambers available at Knoxville, so the laboratory evaluation was modified to use the general laboratory space as an indoor chamber while the outdoor skid was installed outside. While this approach did not allow for precise control of the outdoor conditions, the indoor space was conditioned by the building's HVAC system, thus providing a relatively constant indoor air condition (70°F [21°C], 50 percent RH). This provided a suitable test bed to evaluate basic controls and monitor the capacity of both the ammonia heat pump and the CO_2 distribution loop.

The construction of the prototype was separated based on the indoor and outdoor skids. Because the components had different shipping dates, the mechanical work (welding and brazing) had to be postponed until all components arrived and were dry fitted. The indoor skid contained two AHUs, both designed to provide 10 RT of capacity. Due to constraints on available space, the two AHUs were stacked on top of one another. The indoor rack contained a control panel, which provided the controls for the VFDs for the variable speed fans of the indoor AHUs, the VFDs for the CO_2 pumps, both compressors, and the superheat controllers for the CO_2 loop's indoor EEVs. The CO_2-to-air heat exchangers were procured separately and had to be installed within the AHU.

The outdoor skid contained both ammonia and CO_2 components and used a repurposed refrigeration testing rack. The ammonia side required non-copper tubing (the project used stainless steel due to being outside) to connect components while the CO_2 side used copper-iron alloy tubing. The stainless steel tubing required welding to connect components and route the piping. The recommended oils for the ammonia compressor were either mineral oil or polyalkylene glycol. The former was selected for the prototype with a recommended amount of 2.4 gal. An oil separator was 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. Black iron tubing was added to the outlet of the pressure release valves to route exhausted ammonia from an over pressure condition to the ground and away from face level.

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 a 10 percent 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 could adapt to accommodate different diameter tubing stubs on equipment and threads, and some direct fittings were not available. For example, Figure 15 shows the adaption from the 7/8-inch system tubing toa ½-inch filter dryer. The filter dryer had *solder cups* (typically denoted with a C) that allowed for ½-inch tubing to be set in the cup and brazed. Unfortunately, a copper iron 7/8-inch cup (C) to ½-inch fitting (F) was not available. In its place, two fittings were used to make the transition: 7/8-inch coupling (C-C) and 7/8-inch to ½-inch reducer (F-F).

Figure 15: Adaption of System Components



Source: EPRI, 2024

The two loops were coupled by the plate heat exchanger made of stainless steel and rated for 110 bar (1,595 psi) since it had to be suited for both working fluids. The CO_2 distribution loop used high pressure rated components. Based on the previous test data, a total of three pumps were installed to ensure adequate heating capacity by estimating the flowrate each pump could provide. The CO_2 compressor required lubricating polyolester oil with a manufacturer recommended amount of 2.4 gallons and an oil separator. A receiver was used to manage CO_2 charge since the charges 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 1,400 psig. The outdoor skid was not covered to allow for easy access to components and piping, and a rain cover was used to protect the unit against rain or snow when necessary. Both ammonia and CO_2 compressors had accumulators to prevent liquid from travelling to the compressors.

After all components had arrived, EPRI staff and the mechanical contractor mounted all components and dry fitted the necessary piping connections. Once all components were dry fitted, field welding and brazing were performed to assemble the prototype system.

The outdoor construction resulted in several unexpected weather delays (for example, rainfall and snow). An extreme weather event in January 2024 also halted work for several weeks, when low temperatures caused EPRI to shut down the Knoxville office due to health and safety concerns.

After the brazing and welding work were completed, the system was pressure tested to ensure there were no leaks. This process included charging the system with nitrogen to a specified pressure (ammonia loop about 300 psig and dioxide CO_2 loop about 1,200 psig, when leak testing) and observing any changes in pressure over time. Numerous repairs had to be made, which was expected considering the size of the system and the number of joints. While the larger leaks were easy to identify, it became increasingly difficult to locate the smaller leaks. Both the ammonia loop and CO_2 loop were pressure tested at the expected operating range, with 350 psig for the ammonia side and 1,400 psig for the CO_2 side.

Once the team was confident that the system was holding pressure, vacuums were pulled on both sides to evacuate all gases. This step was critical to ensure no non-condensables (for example, nitrogen) or water vapor remained in the system, as their presence could degrade the thermal performance, cause issues if mixed with the lubricating oils, and in the case of water vapor, cause ice formation within the system. Once the system was evacuated, it was left under vacuum as a final check for leaks. Red stanchions were also placed about 10 feet around the outdoor unit to establish a blocked perimeter in case the pressure relief valves were to open and release while operating.

After the prototype heat pump was pressure and leak tested, the team prepared to charge the system with refrigerant. The amount of refrigerant needed for the prototype heat pump was estimated based on the overall system volumes with assumed densities. The outdoor heat exchanger coils, intermediate plate heat exchanger, accumulator, oil separator, filter dryer, and 10 feet of line set were used as the volume for ammonia. The density used for ammonia was the saturated liquid density at 90°F (32°C). The indoor CO_2 coils, intermediate plate heat exchanger, CO_2 accumulator, CO_2 oil separator, CO_2 filter dryer, and 200 feet 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 (4.4°C). The resulting refrigerant amounts are given below in Table 9. It should be noted these estimates did not consider the actual refrigerant line set volumes and assumed the full heat exchanger volume was 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 ammonia. Ammonia ordered was to a purity of 99.995 percent, and CO_2 was ordered (refrigerant grade) for the CO_2 to ensure a low moisture content in the gas.

Table 9: Estimated Refrigerant Charge Based on Component and Line Set Volumes

Refrigerant	Volume (ft3)	Density (lb/ft3)	Charge (lb)
R-717	2.93	36.94	108.2
CO_2	2.0	56.16	112.3

Source: EPRI, 2023

The refrigerant cylinders were connected to charging equipment via regulators, which served as adapters. The ammonia regulator was compatible with ammonia (no copper/copper alloys) and connected via a CGA-705 fitting and provided a 1/4-inch male national pipe thread connection for the charging equipment, and non-copper charging equipment was procured for the heat pump. The CO_2 regulator was specified for high pressure and provided a CGA-320 connection to the cylinder and a ¼-inch male national pipe thread allowed connection to the system.

EPRI reached out to several contractors to charge the ammonia system due to toxicity concerns; however, contractors either lacked the experience with this refrigerant or did not pursue this contract due to the low value from charging a small amount of ammonia. EPRI staff reviewed how to charge the system in coordination with its Environmental Health and Safety and Lab Safety groups, and it was determined that the team could do the work safely in house.

The ammonia loop was charged first. The refrigerant cylinder was weighed and then connected to the outdoor unit through an ammonia manifold gauge set to the accumulator with an ammonia rated vacuum pump connected to the available manifold gauge port to evacuate the hoses. The pump was then isolated through a valve, and the refrigerant cylinder valve was opened. The pressure reading was logged. The system valve was then opened to allow refrigerant to flow into the system. Since the system was under a vacuum, the pressure differential would pull refrigerant into the system until the system reached equilibrium. When the pressure was equalized, the compressor was turned on to pull additional refrigerant into the system. The system was configured to operate in cooling mode during the charging process. Typically, heat pump manufacturers provide guidance on the total charge amount and the optimal subcooling used to fine tune the charge amount on site. For this prototype unit, there was little information on the optimal charge, so the team decided to charge the system in a stepwise manner to identify the optimal ranges of refrigerant quantity.

During the charging process, the team experienced a minor leak at the ammonia regulator, which the team was able to fix after closing the system and cylinder valves and venting the remaining ammonia into water. The team also discovered the four-way valve and EEVs were not in the correct position. The valve positions were changed, the heating mode EEV was fully closed and cooling mode EEV was closed to 75 percent open. After these adjustments, the team resumed charging the system. The team verified that the compressor discharge was routed to the outdoor heat exchanger and noted a cooling effect on the system's low-pressure side. The high side pressure was 150 psig (corresponding to a saturated temperature of 121°F [49.4°C]) and low side pressure was 70 psig. The charging process was slow, and the team reviewed the guidance on the refrigerant cylinder tag, which indicated the cylinder should be laid on its side so liquid can be pulled instead of vapor. These adjustments worked well and the remaining charging process was completed. The refrigerant cylinder was weighed during the charging process and a total of 85 pounds of refrigerant was charged into the system. The remaining ammonia was routed via the vacuum pump exhaust and dissolved in a water bucket.

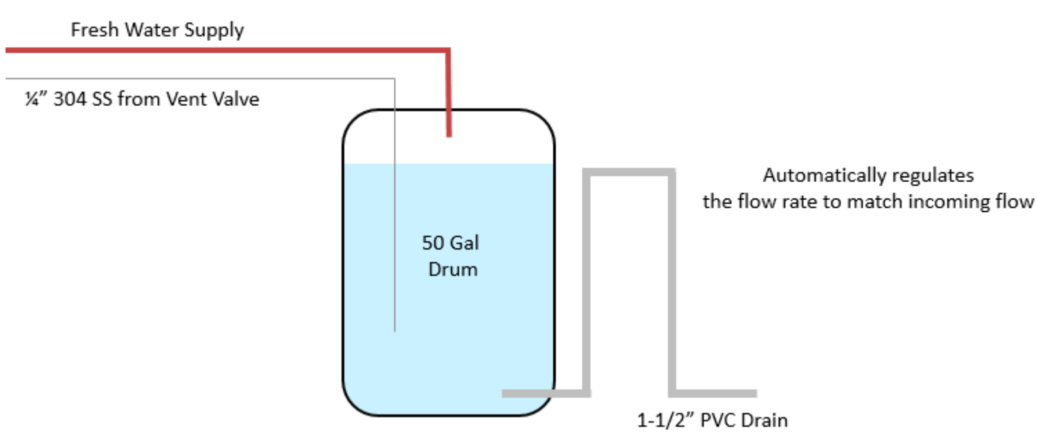
After the ammonia system was charged, the team followed a similar procedure for the CO_2 side. The CO_2 cylinder was connected to the system via a high-pressure manifold gauge set to the CO_2 compressor and a vacuum pump. Once the pressure equalized, the CO_2 compressor was turned on to pull CO_2 into the system. The ammonia system was then turned on to remove heat from the system. During the system charging process, the team heard a loud clap from the ammonia system and noticed bulging on the ammonia oil return line, as shown in Figure 16. The ammonia side of the system was shut off and the team began evacuating the refrigerant by dissolving the ammonia into water. The ammonia was diluted to less than 2 percent concentration by mass, around 0.5 pound of ammonia into 32 pounds of water. A vacuum pump was used to remove any remaining ammonia vapor. The evacuation process took a total of 10 hours and spanned two days. It should be noted that the team could smell ammonia when they discharged it into the water buckets and the ammonia detectors did not register an alert. This occurrence may be due to the leak detector being installed on the outdoor skid without an enclosure, so the concentration of the mixture was low enough to avoid detection in a large ambient area.

Figure 16: Bulging of Ammonia Oil Return Hose and Rust within Hose Jacket

Source: EPRI, 2024

The team inspected the hose and noted the outer jacket easily peeled away and there was rust on the inner tubing. While the hose should be compatible with ammonia, the vacuum test may have exceeded the hose's rating. The ammonia may have dissolved the seal on the inner tubing and began leaking into the outer jacket. A stainless-steel tubing with compression fittings was used to replace the hose. The team also implemented an ammonia evacuation protocol for future incidents where the ammonia charge may have to be evacuated. The protocol involved setting up a dilution system where ammonia is discharged into a water drum and the diluted mixture is discharged into a drain. Fresh water supply is also connected to this water drum. The flow rate is regulated to match the incoming flow so that the water column in the water drum remains constant. A schematic of this evacuation setup is shown in Figure 17. Once the ammonia oil return line was repaired and the dilution system was in place, the team resumed charging the system. The team charged 40 pounds of ammonia and 51.5 pounds of CO_2. Note the ammonia charge was lower than before due to what was available in the existing cylinder while another cylinder of ammonia was ordered.

Figure 17: Ammonia Evacuation with Water Drum



Source: EPRI, 2024

Operating Procedures

Once the prototype was charged, the team began testing the system in both cooling and heating modes. These initial tests also served to determine if the system was adequately charged.

The operation of the heat pump began with setting the valves in the appropriate positions for the desired operation: 1) Heating Mode, CO_2 Compressor; 2) Heating Mode, CO_2 Pumps; 3) Cooling Mode, CO_2 Compressor; or 4) Cooling Mode; CO_2 Pumps. Specifically, ball valves needed to be opened or closed, EEVs for the ammonia and CO_2 sides had to be set to position, and the four-way reversing valve had to be set to cooling or heating mode on the ammonia circuit.

The ball valves were set (open/closed) using the schematic shown in Figure 14. The EEVs for each circuit were then adjusted for the given operation. The EEVs for the CO_2 side were connected to the EEV superheat controller (EKE 1C), and a total of three EEVs needed to be adjusted prior to operation. For testing, the controllers were put into Service mode and adjusted via the step size, where zero was fully closed and 1,100 was fully open. The CO_2 side EEVs were typically set to 150 steps in throttling mode, then further adjusted during the tests.

The ammonia EEV (ICAD 600A) was operated through the on-board interface. This interface accepts a percentage that corresponds to the amount that the valve is opened where 0 percent is closed and 100 percent is fully open. When the valve was in throttling mode, it was typically set around 80 percent and then adjusted to achieve specific flow rates or capacities during the tests. While the test was running, temperatures, pressures, flow rates, and power consumption were being monitored in real-time. In particular, the supply temperature was watched closely to see if the unit was producing a noticeable heating or cooling effect. The EEVs would be adjusted to see if better cooling or heating could be achieved while the test was operating.

The final step prior to starting the system was to set the four-way reversing valve on the ammonia side to the appropriate mode: cooling or heating. The mode could be toggled via a switch that interfaced with the valve. When the valve was in heating mode, the indicator on the valve would be red and would display "CLOSED," while in cooling mode the background was yellow and displayed "OPEN."

The prototype heat pump start up always began with turning on the heat removal elements and verifying their operation so that any problems with these components could be addressed before energizing other components. This process began with starting the data acquisition to capture all events that occurred through the test. Next, the Colmac ammonia coil was energized via a disconnect to run the fans in the coil. The indoor control panel was then powered through a main switch, which also served as an emergency stop. This panel was connected to the compressors, pumps, and indoor AHUs. The compressors and pumps speed were controlled using VFDs. The drives' power was controlled by rotary switches, and potentiometers were used to adjust the output frequency of the VFDs. The ammonia compressor was started first by applying power to its VFD and then increasing the speed of the VFD. The associated pressures and temperatures were monitored as well as visual inspection performed of the compressor because it was such a large machine. After the ammonia compressor was seen to be operating as expected, the CO_2 side was powered. Similar to the ammonia side, the indoor AHU fans were energized first. These fans had integrated speed control and did not require any external VFDs. The CO_2 pumps or compressor was then started in a manner similar to the ammonia compressor. If operating the pumps, there was the additional step of deciding how many pumps to operate. This required closing additional breakers for the pump(s) and opening respective valves.

The following is an example of the operation process:

1. The ammonia compressor VFD was switched to the "On" position.
2. The VFD was increased to about 15 Hz.
3. Visual inspection of the ammonia compressor while starting up was done to monitor vibrations and abnormal behaviors.
4. The refrigerant flow rate, pressure, and temperatures were monitored as the ammonia compressor started.
    a. If pressures were not stable, the ammonia compressor was shut off and valve orientation was re-checked.
    b. If excessive vibration was witnessed, the VFD was adjusted.
    c. If pressures were stable and as expected, the CO_2 loop could be started.
5. The two AHUs were energized via rotary switches and controlled by potentiometers (0–10 VDC) signal.
6. Air flow rate at the AHUs was verified.
7. The CO_2 compressor/pumps were energized through rotary switches and the VFD's speed adjusted via potentiometer.

As the system operated, temperatures, pressures, and flow rates were monitored through the data acquisition program in addition to watching and feeling air flows. EEVs and VFDs were adjusted to verify the system's response and change in performance (for example, supply air temperature and relative humidity).

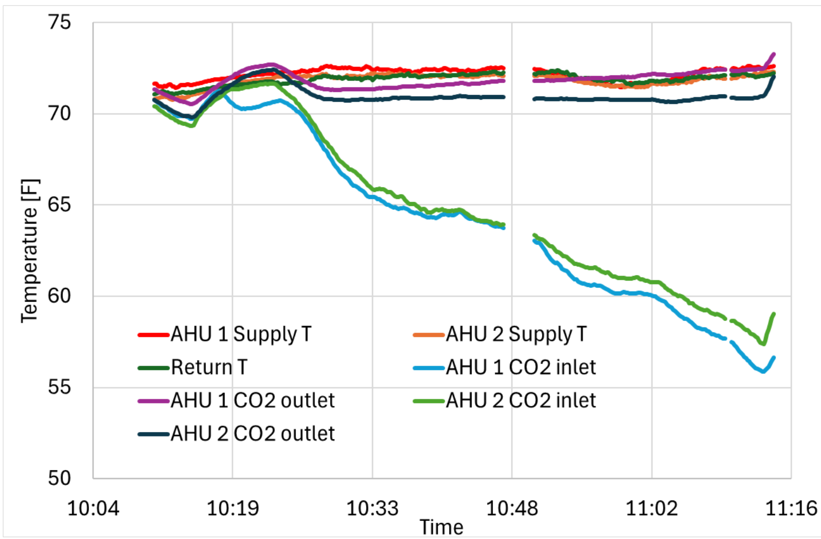
To shut the system down, the ammonia and CO_2 compressors were de-energized. The indoor AHUs and outside condenser were operated until pressures reached saturation pressure at the given ambient temperature, typically within 10 minutes. The indoor AHUs and condenser were then turned off. Finally, the data acquisition program was terminated and power to EEVs was removed.

Laboratory Evaluation

The evaluation of the prototype heat pump began with cooling and heating tests while operating the CO_2 compressor. One of the goals of these initial evaluations was to determine if the system was appropriately charged. In these evaluations, the prototype heat pump did not exhibit steady state behavior in heating or cooling mode.

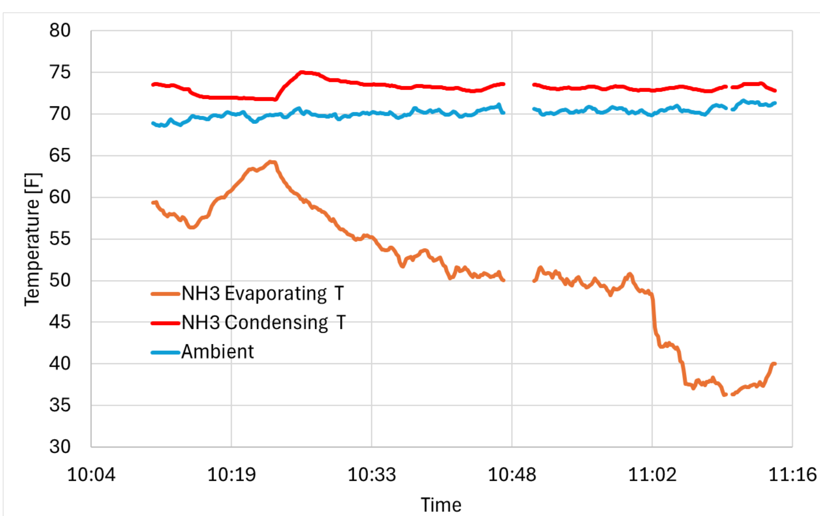
Figure 18 shows a cooling mode test, where the CO_2 loop temperature and pressure continued to decrease. The supply air (72°F [22.2°C]) was within 1°F (0.5°C) of the return air (72.5°F [22.5°C]), but the measured temperature of the CO_2 entering the AHU reached well below 60°F (15.5°C). A larger temperature difference was expected based on the measured values. EPRI's hypothesis is that CO_2 vapor instead of CO_2 liquid was entering the AHU, leading to a lower heat transfer rate to the air (lack of mass to carry the heat). This may have been caused by the CO_2 piping around the receiver, where several adjustments had to be made to accommodate all four operating modes and the available space, resulting in the system drawing CO_2 vapor from the top of the receiver toward the indoor AHU instead of the bottom where the liquid CO_2 would be collected. This issue with CO_2 vapor may be resolved by using the pump instead of compressor to circulate CO_2. The lack of capacity on the CO_2 loop was also reflected on the ammonia side behavior (Figure 19), where the condensing temperature began to decrease.

Figure 18: CO_2 Temperatures During Cooling Mode Test with CO_2 Compressor



Source: EPRI, 2024

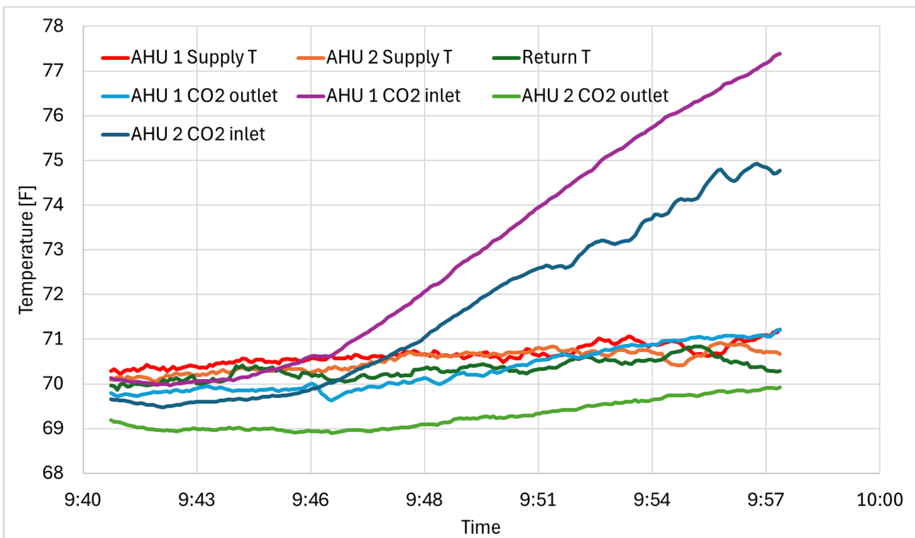
Figure 19: Ammonia Temperatures During Cooling Mode Test with CO_2 Compressor



Source: EPRI, 2024

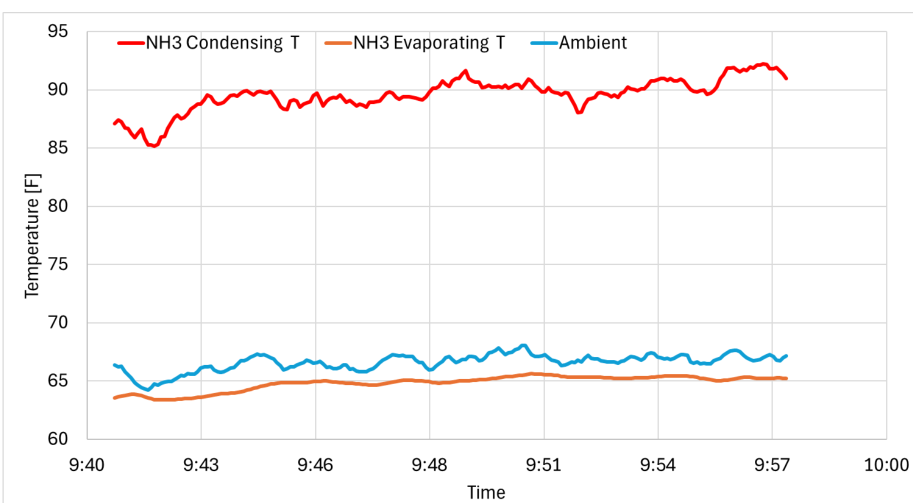
Figure 20 shows a heating mode test where the CO_2 side did not stabilize. The CO_2 pressure and temperature at the ammonia-to-CO_2 heat exchanger outlet increased, reaching up to 130°F (54.4°C), but the CO_2 temperature entering the indoor AHU did not exceed 80°F (26.7°C). Due to this low temperature, the air side measurements showed no significant changes to the air temperature existing in the AHU. Figure 21 shows the ammonia side temperatures. The ammonia condensing temperature was slightly increasing over the course of the test; the change was relatively low and may be influenced by the outdoor temperature. The lack of capacity may suggest that both ammonia side and CO_2 side were undercharged, and the CO_2 temperature changes may be caused by heat loss at some uninsulated pipe sections. These sections were left uninsulated to accelerate the system commissioning and testing.

Figure 20: CO_2 Temperatures During Heating Mode Test with CO_2 Compressor



Source: EPRI, 2024

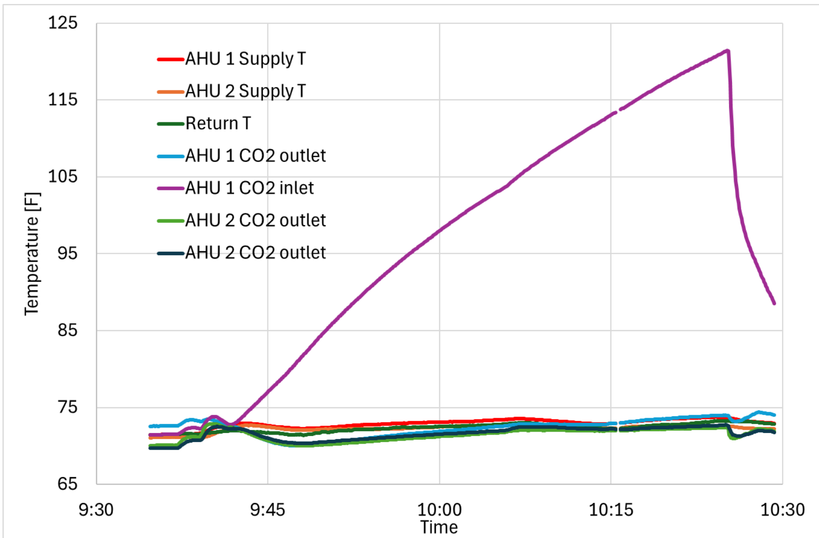
Figure 21: Ammonia Temperatures During Heating Mode Test with CO_2 Compressor



Source: EPRI, 2024

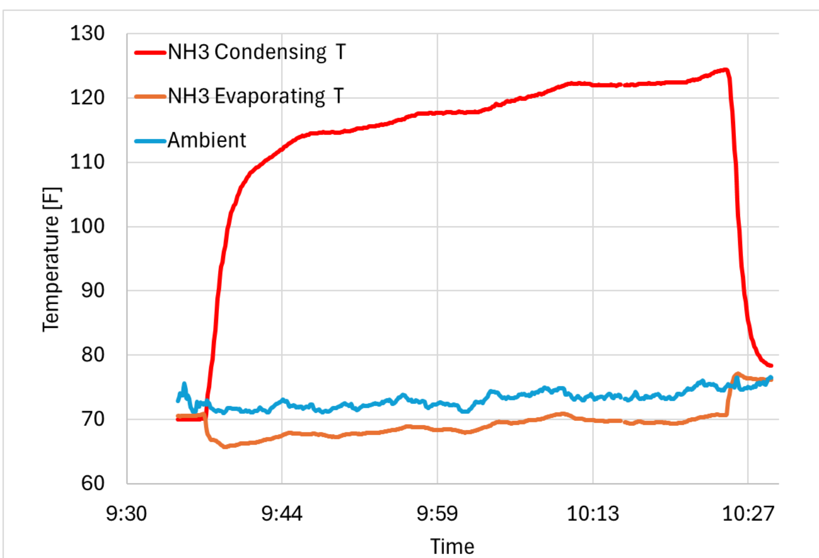
After the initial tests, EPRI added insulation to all CO_2 and ammonia piping and increased the system charge to 72.6 pounds of ammonia and 62.6 pounds of CO_2. Subsequent heating mode test data showed behavior similar to the previous tests. The CO_2 entering the AHU reached up to 120°F (48.9°C) with the additional insulation as shown in Figure 22. The temperature and pressure continued to rise on the ammonia side (Figure 23), which suggests the CO_2 side may still have been undercharged and could not adequately remove the heat. It may have been caused by having an oversized ammonia compressor (40 RT). EPRI hypothesizes that reducing compressor speed on the ammonia side while increasing compressor/pump speed on the CO_2 side may solve this issue.

Figure 22: CO_2 Temperatures During Heating Mode Test with CO_2 Compressor, with additional insulation



Source: EPRI, 2024

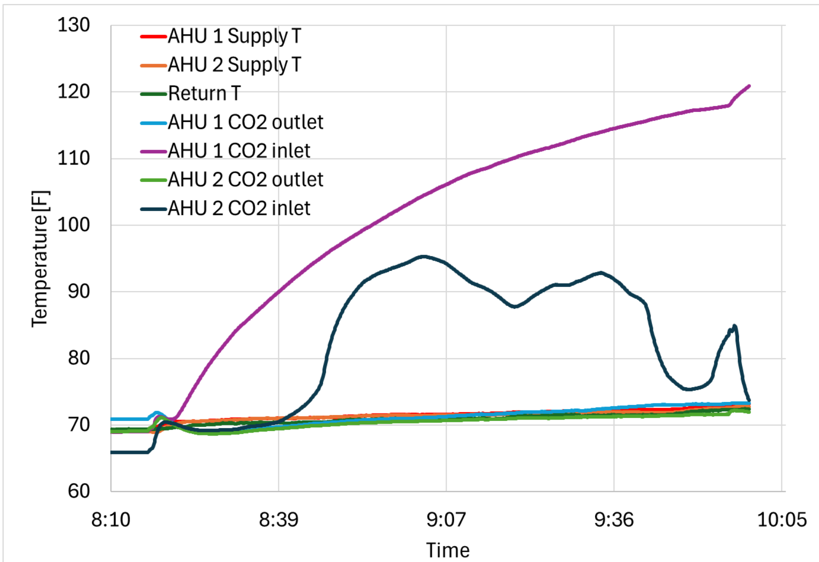
Figure 23: Ammonia Temperatures During Heating Mode Test with CO_2 Compressor, with additional insulation



Source: EPRI, 2024

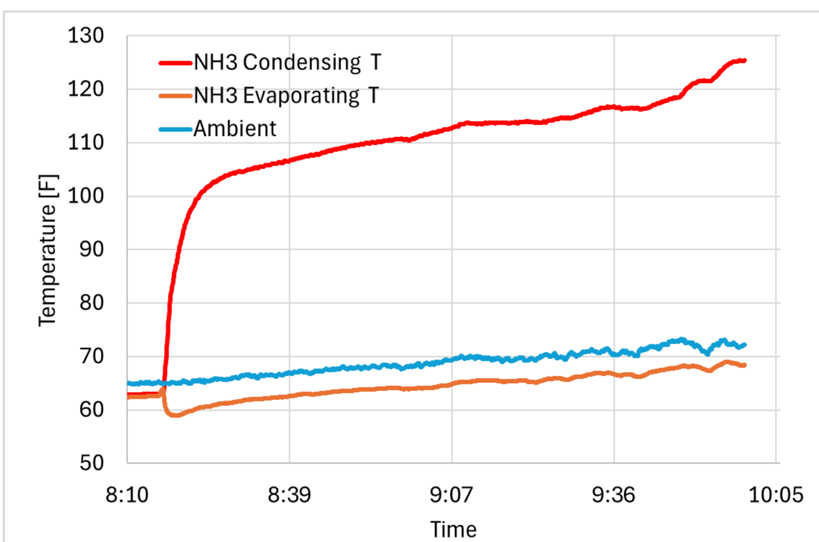
The next test was performed with the EEV in a 64 percent closed position on the ammonia side; however, similar issues of a potentially undercharged system were observed (Figure 24 and Figure 25). The CO_2 pressure readings during these heating tests were below 900 psig as shown in Figure 26, but the design model suggested the pressure should have been around 1,200 psig. The low pressure can significantly affect the supercritical CO_2 heat transfer coefficient, which may explain the lack of heating capacity. The low heat capacity coming from the ammonia system was likely causing this low pressure. The team decided to increase the charge of both sides to reevaluate the system's heating performance.

Figure 24: CO_2 Temperatures During Heating Mode Test with CO_2 Compressor, with Additional Insulation and EEV Adjustment



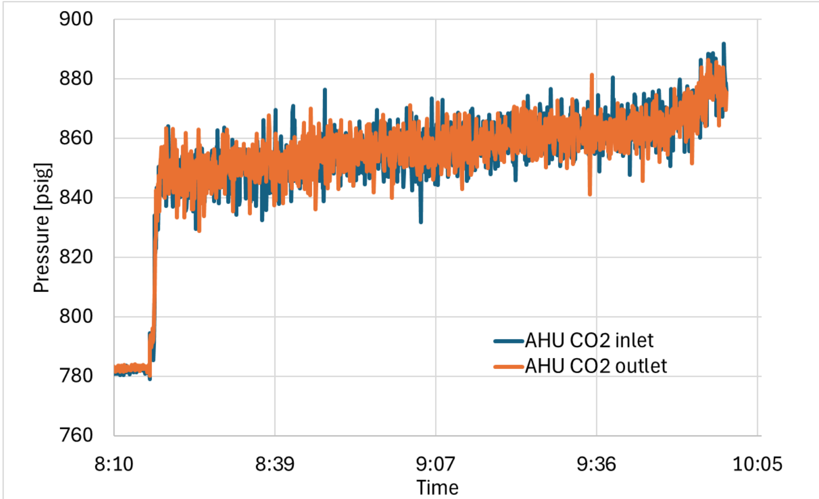
Source: EPRI, 2024

Figure 25: Ammonia Temperatures During Heating Mode Test with CO_2 Compressor with Additional Insulation and EEV Adjustment



Source: EPRI, 2024

Figure 26: CO_2 Pressures During Heating Mode Test with CO_2 Compressor, with Additional Insulation and EEV Adjustment

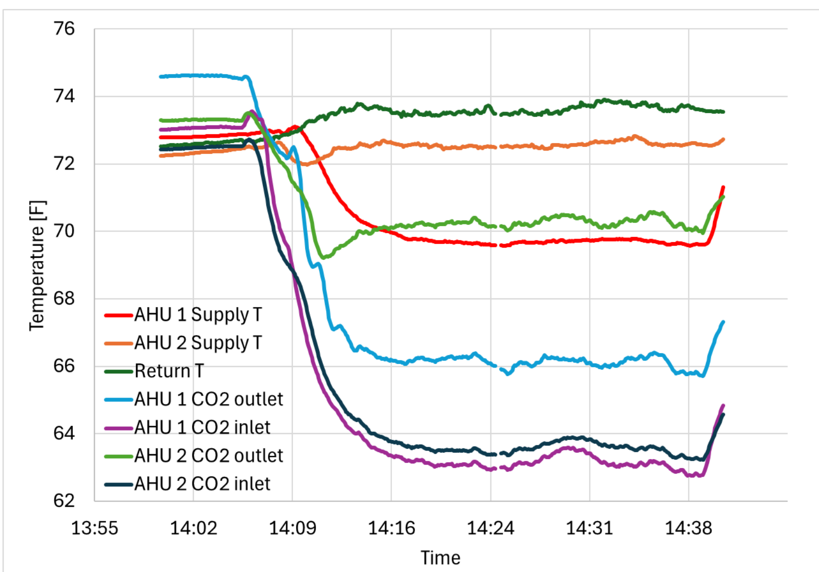


Source: EPRI, 2024

The system charge was increased to 99.7 pounds of ammonia and 105.4 pounds of CO_2. EPRI further evaluated the cooling mode performance with one CO_2 pump, which showed marked increase in the demonstrated capacity (greater than 100 percent) compared to previous tests with the compressor. Figure 27 shows the CO_2 temperature data during this test where AHU supply temperatures decreased in a detectable manner unlike previous tests with the CO_2 compressor. Figure 28 shows the ammonia operation with quasi-steady behavior.

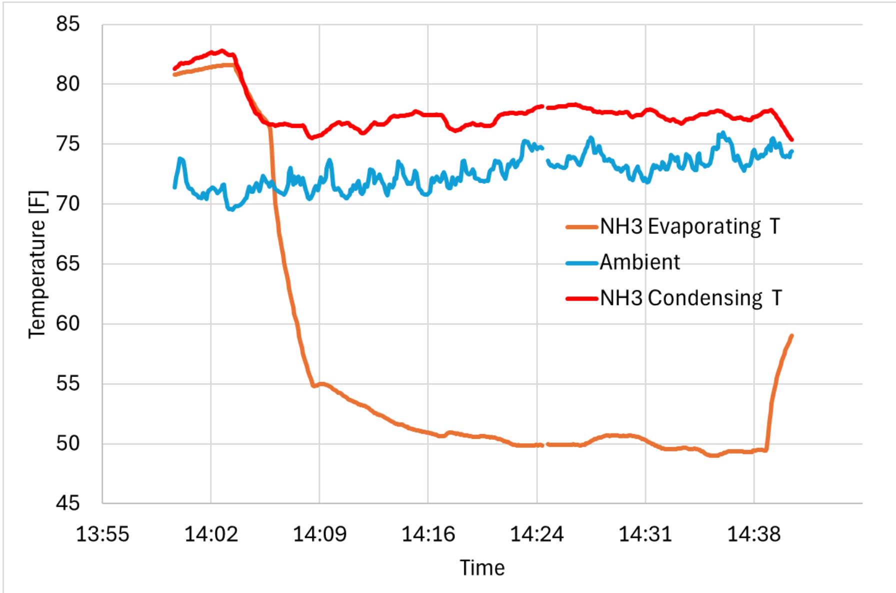
Additional testing with multiple pumps resulted in screeching noises, likely caused by cavitation. Liquid CO_2 pumping can lead to cavitation if insufficient pressure head is present at the pump inlet. CO_2 in the receiver tank was in a saturated state so pumping action tended to cause rapid boiling and cavitation.

Figure 27: CO_2 Temperatures During Cooling Mode Test with CO_2 Pump



Source: EPRI, 2024

Figure 28: Ammonia Temperatures During Cooling Mode Test with CO_2 Pump

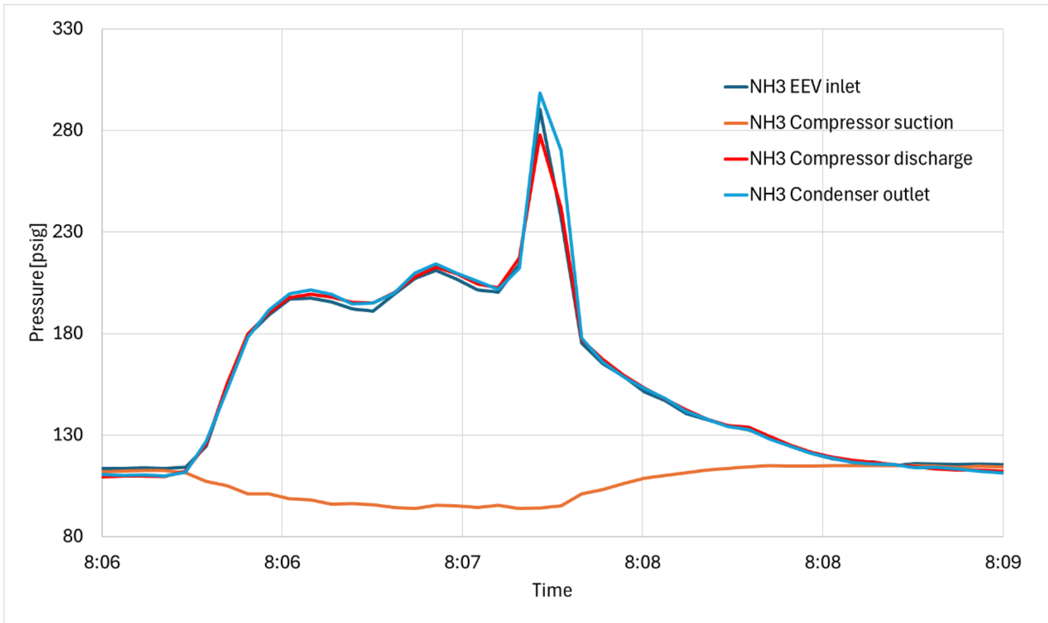


Source: EPRI, 2024

As the testing continued with heating mode evaluation, the ammonia side experienced significant increase in refrigerant pressure after system start up (about 5 minutes), exceeding the limits on the pressure release valves and causing ammonia to leak out of the system. Figure 29 shows the pressure readings during the compressor start up, where the pressure at the compressor discharge, condenser (intermediate heat exchanger), and EEV increased significantly. Figure 30 shows the temperature readings on the ammonia side during this incident.

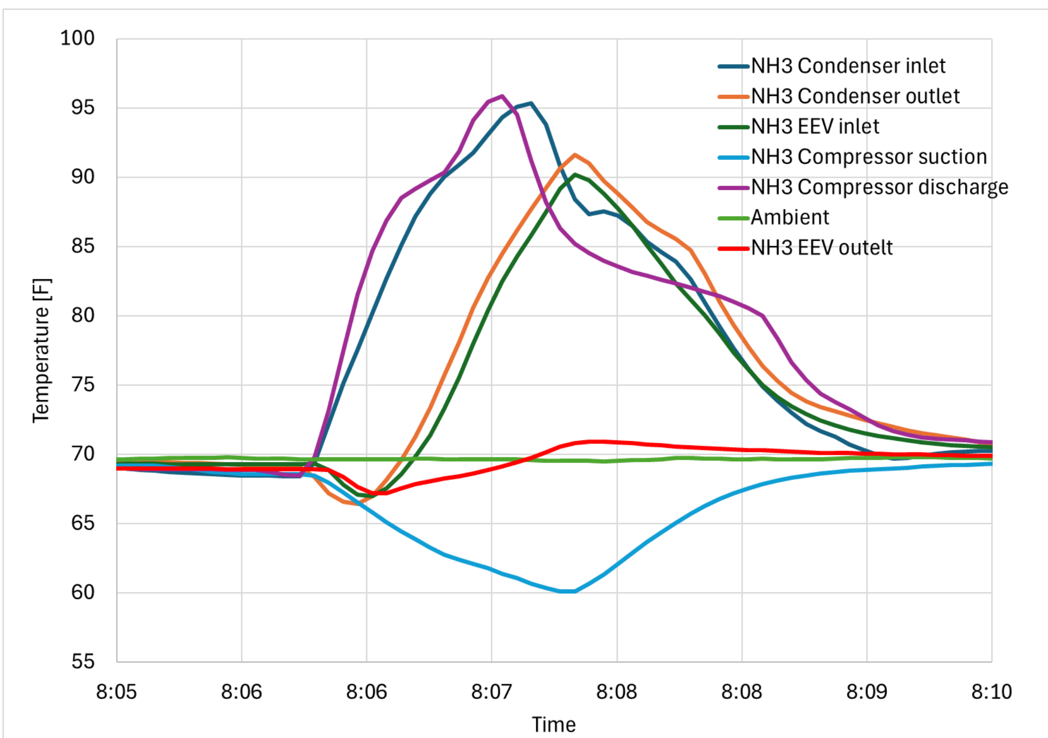
Fortunately, the system was shut off quickly and the system did not lose a significant amount of ammonia, estimated to be about 1 pound per pressure release valve leak incident. The issue with pressure spikes continued after several adjustments with the EEV. There may have been some blockages before the EEV was used for heating mode, since cooling tests did not experience the same issue. Another possibility is the EEV was stuck in the wrong position and blocking the path for the ammonia refrigerant. This possible blockage may be resolved by replacing the EEV, which could have been damaged or had a one-off defect. At this stage of the project, several factors resulted in an early conclusion of the evaluation. See Chapter 5 for details.

Figure 29: Ammonia Pressures During Heating Mode Startup



Source: EPRI, 2024

Figure 30: Ammonia Temperatures During Heating Mode Startup



Source: EPRI, 2024

Decommissioning

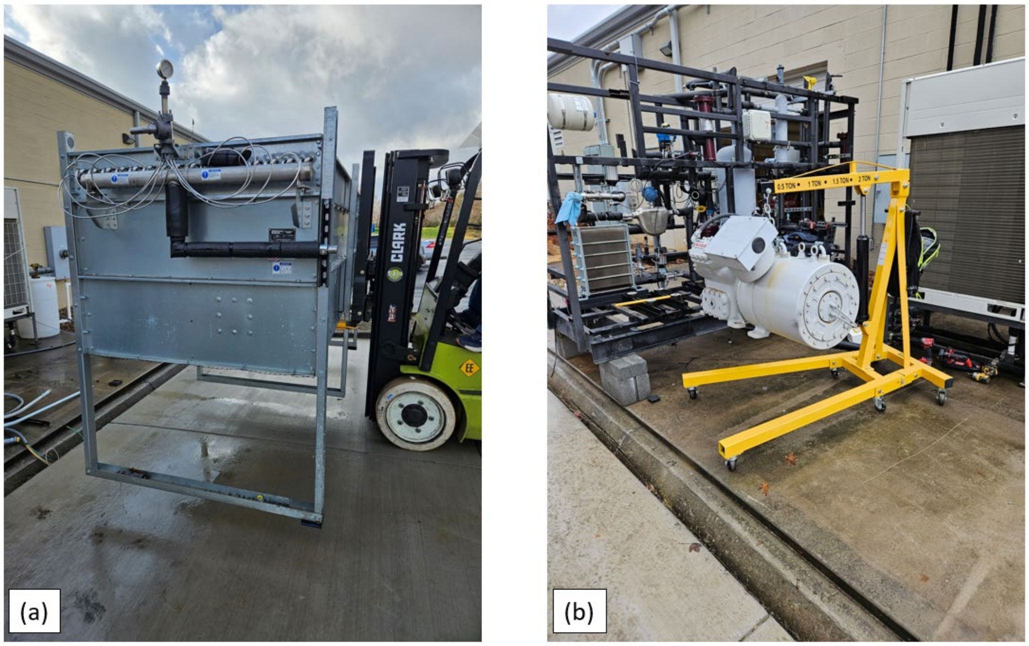
After the laboratory evaluation of the prototype was concluded, the prototype system was decommissioned. The major tasks included disposal of refrigerant and lubricating oils and partial disassembly of the ammonia system. Not all components were removed from the skid so that future work may use this framework and continue refining this system. Venting of the ammonia was done through the dilution system along with fresh water running into the dilution system. The venting process only occurred if staff were present for safety considerations. Venting began on December 9, 2024, and lasted for three hours. The following morning (December 10), venting was attempted but the vent valve was stuck. The hypothesis is that water from the dilution tank moved from the tank up the stainless steel tube and subsequently froze since the outdoor temperatures were below freezing. This may have been due to a pressure drop from the venting process, which reduced the temperature of the stainless steel tube and created low pressure suction to draw the water up to the valve.

When the valve was reopened, the team realized that the sealing mechanism within the valve was damaged, since ammonia did not stop flowing even when the valve was closed. In future scenarios, it is recommended to wait for longer durations so that any frost build up in the valve's interior can be defrosted to avoid damaging the valve. This may be done through sun exposure (if temperatures and weather allow) or with heat blankets. These setbacks caused the evacuation process to run into the night and concluded around 8:30 p.m. The ammonia system was further evacuated with a vacuum pump to remove residual ammonia in the oil. Overall, the venting process worked aside from the valve failure.

The CO_2 side of the system was vented parallel to the ammonia side. The CO_2 was slowly vented from the outdoor unit. For future systems, it is recommended that a filter is in place on the vent such that the lubricating oil is captured and not released into the surrounding area or onto nearby automobiles and buildings. All valves on the CO_2 loop should be opened such that sections of the system do not trap CO_2.

The electrical connections were then removed. The goal was to keep the system as complete as possible for potential reuse in future work. First, the ammonia condenser was moved. The pipes running to the condenser were cut and a forklift was used to move the condenser (Figure 31a). The ammonia compressor was then removed from the outdoor skid. The suction and discharge connections were cut using a portable band saw and the larger stainless steel tubing was cut using a Sawzall. The flanges were then removed, and the compressor was unbolted from the skid. Because of the location of the skid and compressor, an engine hoist was used to lift compressor and set it on the forks of a forklift (Figure 31b). A wooden frame/pallet was fabricated to make moving the compressor by forklift or pallet jack easier.

Figure 31: Disassembled Prototype



Source: EPRI, 2024

Next, the CO_2 refrigerant lines entering/exiting the building were cut and the outdoor skid was moved. The skid, without the compressor, was picked up with the forklift and moved to the opposite side of the parking lot (Figure 32). Finally, the indoor section of the prototype was deconstructed. While this was standard equipment removal, it should be noted that oil can migrate to many parts of the system. It is recommended to cap the end of the pipes after cutting to avoid any oil spills, particularly on the indoor air handling units.

Figure 32: Prototype Skid Removed with Forklift

Source: EPRI, 2024

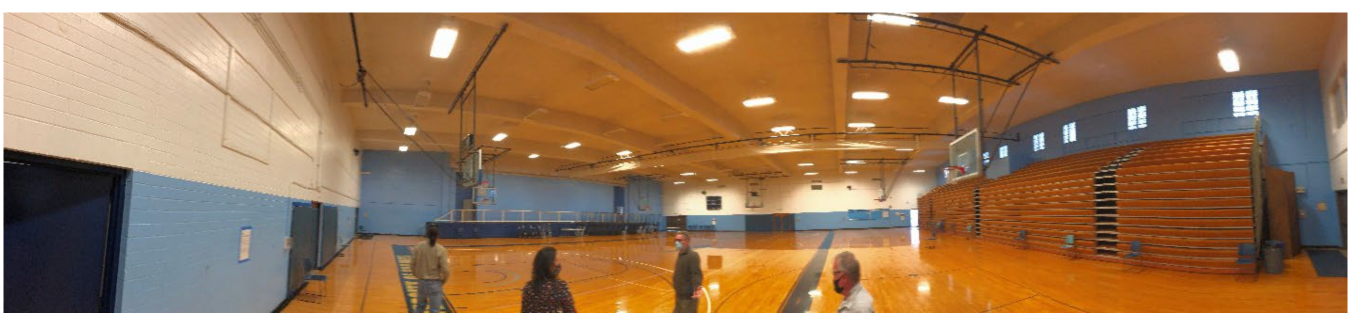
CHAPTER 4: Field Site Preparations

Concurrent with laboratory evaluation of the prototype heat pump, EPRI prepared for the field demonstration in collaboration with the demonstration host sites. These meetings with the host site lead (for example, facility/building/project managers) included presentations of the heat pump schematics, discussion of the laboratory evaluation results, site visits to inspect the buildings and existing HVAC systems, reviews of site safety requirements, and formulation of site agreements. In addition, EPRI coordinated with contractors who would be responsible for the transportation, fabrication, design, installation, and permitting of the field units so that the field demonstration would progress smoothly. EPRI worked with preferred contractors from the host sites where possible, as these contractors had familiarity with the site requirements and layouts. Although California State University Maritime Academy (CSUM) and City of La Mesa had to drop out of the project, the preparation work for those sites is detailed in Appendix B for reference.

University of San Diego

EPRI and University of San Diego (USD) project managers conducted several meetings to review the needs of the demonstration effort, including the USD staff's responsibilities, schematics of the heat pump, power requirements, physical dimensions of the assembled system, transportation from EPRI's contractor facilities, installation, commissioning procedures, operation and maintenance, and decommissioning process. The USD team provided structural and mechanical drawings of the Sports Center for EPRI to review. The building had existing space heating through a forced air system, and it was planned for the heat pump to leverage existing ducting in the building to deliver space conditioning. EPRI visited the site on June 11, 2021, to install HOBO sensors to establish baseline data. The indoor conditioned space is shown in Figure 33.

Figure 33: USD Site Indoor Conditioned Area



Source: EPRI, 2021

In 2023, USD identified its Sports Center as a potential recipient of a large solar array for a large solar installation across campus. The USD project managers had concerns about the heat pump installation as it may have interfered with the solar installation, as shown in Figure 34.

EPRI hosted several meetings with USD to review the heat pump schematics and it was determined that the heat pump and solar installation could happen concurrently.

Figure 34: USD Site Rooftop Area for Field Installation

Source: EPRI, 2021

Since students, staff, and faculty visit the Sports Center, releasing any refrigerant charge could cause significant health hazards. EPRI presented the ammonia dilution system installed at the Knoxville laboratory with the USD project managers to solicit their feedback on a similar system for USD. USD had concerns about discharging the diluted ammonia mixture directly into storm drains, which may cause damage. For example, in the ammonia evacuation at EPRI, the dilution at 1.4 percent concentration by mass caused discoloration on the storm drain grates. USD proposed an alternate approach with large water containers that could dilute the ammonia to safe levels without discharging, and the container could be replaced after potential leaks.

Another concern from USD was with the orientation of pressure relief valves on the ammonia system. USD expressed its preference for the pressure release valves to face downward and be connected to vessels that contained the ammonia charge; however, manufacturer instructions and industry practices both indicated that pressure release valves should be faced upward for safe operation. The ambient air can dilute the leaked ammonia to safe levels and ammonia is less dense than air so the leaked ammonia would not descend to occupants. Also, leakages would be reported using leakage detectors.

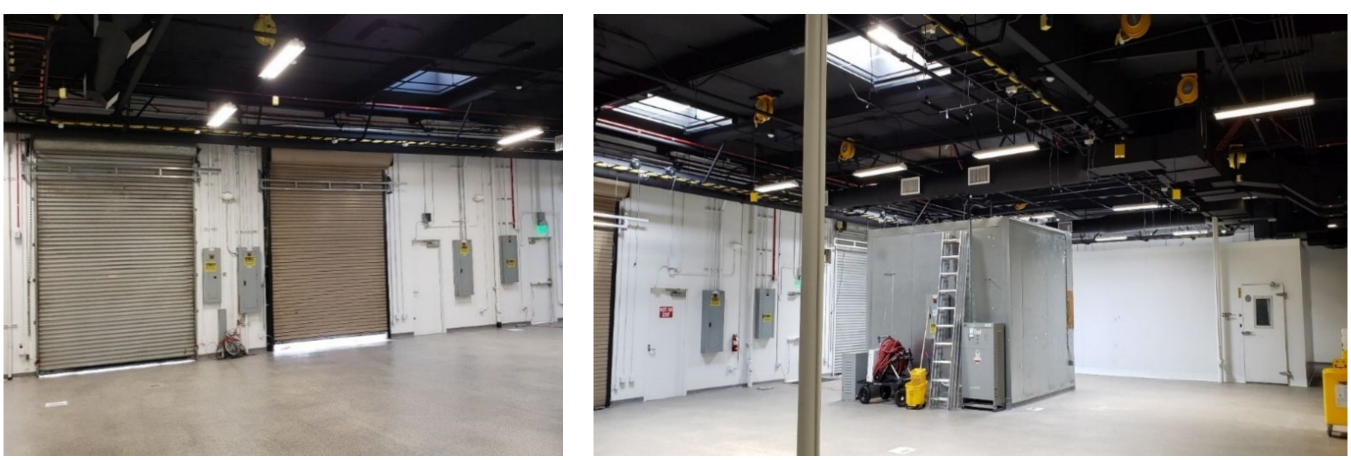
USD expressed concern about students, staff, and contractors encountering the prototype unit on the rooftop. While the rooftop should not be accessible to students, the USD project managers noted students had entered the rooftop in the past and may cause damage to the prototype heat pump or be exposed to ammonia. A similar issue may be faced by contractors and staff who maintain the newly installed solar array and other equipment on the rooftop. EPRI proposed to install fencing, safety warning signs, and OSHA signs for hazardous materials around the heat pump to deter people from interacting with the equipment, and that was satisfactory for USD. USD also recommended that the weatherization cover be designed for the coastal climate's high humidity and salinity.

USD also had concerns about the long-term maintenance of the equipment since this is a novel technology that was designed by EPRI and not a typical HVAC product by a manufacturer. USD project managers would prefer to keep the equipment if the performance is satisfactory, and USD wanted EPRI to be responsible for maintenance needs for the life of the system. This was challenging for EPRI as the equipment life may extend well beyond the project's period of performance, and the custom components may be costly to repair. EPRI decided to install a replacement commercially available system (to be owned by USD) at the end of the project that provides comparable capacities to the building as the prototype heat pump. These details were then included into a site agreement for review and signature by the vice president at USD on October 15, 2024.

Southern California Edison

Southern California Edison's (SCE's) Technology Test Center (TTC) was identified as a demonstration site for this project after several other sites had to drop out of the project (see Chapter 5 for details). This decision came after an exhaustive search with both SCE and SDG&E project managers. While the TTC's main purpose is to perform laboratory evaluation of technologies, EPRI proposed using the general laboratory space to evaluate the heat pump prototype (Figure 35).

Figure 35: SCE Site Indoor Lab Space with Garage Doors



Source: EPRI, 2024

EPRI and SCE project managers conducted several meetings to review the needs of the demonstration effort, including the SCE staff's responsibilities, heat pump requirements, transportation from EPRI's contractor facilities, commissioning procedures, operation and maintenance, and decommissioning process. These details were included in a site agreement with SCE, which was executed on August 7, 2024.

The outdoor system was going to be installed in the parking lot area, where it could be fenced off from lab personnel/staff. The refrigerant piping and electrical wiring for the outdoor unit was going to be trenched, so it would be safe for pedestrians and traffic in the parking lot.

Equipment Transportation and Installation

A key activity for the preparation of field installation at the TTC was the transportation of the prototype. The intent was to repurpose the prototype that EPRI evaluated at Knoxville for field installation at the TTC with modifications completed by a custom chiller manufacturer.

There were two options to ship the prototype to the manufacturer: 1) ship by component and 2) ship by skid.

Shipping by component presented an opportunity to optimize the piping of the outdoor unit onto a new skid, allowing the team to optimize the form factor for smoother installation at the TTC due to the strict limits on allocated space. On the other hand, this approach presented a significant time and labor commitment, including disassembling at EPRI and reassembling at the chiller manufacturer.

Shipping by skid was a simpler approach. The prototype could be shipped on three skids. The indoor unit would be its own skid, and the outdoor unit would be separated into two shipments: the outdoor heat exchanger coils and the refrigeration rack with components mounted. After several discussions between EPRI and the manufacturer, shipping the equipment by the three skids was the preferred cost-effective option.

The chiller manufacturer would be responsible for reassembling the unit, making adjustments on piping, and constructing necessary covers for weatherization. SCE also indicated the cover should include proper warning signs for the hazardous refrigerant and description of the project.

Shipping the field unit presented an additional challenge. The chiller manufacturer proposed to ship the outdoor unit on a long skid, where the outdoor skid and heat exchanger are connected in series. But the outdoor unit must fit within the allocated parking spots, meaning the outdoor skid and outdoor heat exchanger need to be parallel to each other. However, this configuration was too wide for standard trailers for shipping. Extra wide trailers may have been available but were costly. Another approach was to ship the two outdoor components separately, then field weld the connections (similar to the Knoxville build process). This also presented some drawbacks, since field welding may not have been in the typical scope for refrigeration contractors. There may also have been additional restrictions on site from SCE's real estate management, since the unit was going to be installed in the parking lot. The final decision on shipping the heat pump to the TTC depended on the contractor quotes on extra wide trailers, field welding, and SCE's preference.

The mechanical contractor completing the installation was going to be responsible for dry fitting and brazing the CO_2 piping, dry fitting and field welding the ammonia piping if required, performing leakage tests, and charging the system with refrigerant. It may have been desirable for the system to be pre-charged prior to shipping if no mechanical work needed to be completed. This would have simplified the installation but also increased risk due to restrictions on ammonia.

Safety Plan

The outdoor unit was outfitted with ammonia detectors. If the ammonia leak detector registered a leak, then the heat pump must be shut off. The weatherization cover should be opened to allow fresh air to dilute the mixture, and for staff to locate the source of the leak. If the leak originated from a pressure relief valve, then simply shutting off the system would allow the pressure to drop to equilibrium at the outdoor conditions. This type of leak would not result in leaking significant amounts of ammonia. On the other hand, if the leak originated from catastrophic component or piping failures, then the refrigerant charge should be evacuated per the ammonia evacuation protocol. It may be possible to isolate the ammonia charge in the outdoor heat exchanger or accumulator without fully evacuating the system, depending on the location of the leak. Once all ammonia is pumped out of the system, the failed component can be replaced. The system should then be pressure tested for additional leaks before refrigerant can be recharged. Any personnel working on the equipment must use personal protective equipment.

A similar plan was put in place for the indoor units. While CO_2 does not have the same hazardous properties as ammonia, a sufficiently large leak is still dangerous for the indoor occupants. The CO_2 is also at a much higher pressure than the ammonia loop, so there can be rapid leakages. Indoor CO_2 detectors should be placed in proximity of the indoor AHU. If a leak is detected, the system should be shut off. Like the ammonia plan, the type of leak would dictate the necessary remedial actions. Pressure relief valve leaks should typically be resolved without further interventions. Small leaks through fittings and valves may be addressed by simply tightening these connections. Due to the small diameter piping, care must be taken to not damage the piping when tightening these components. If this does not resolve the leak, then the system would have to be pumped down to fully address the source of leakage. It may be possible to isolate the CO_2 charge in the CO_2 receiver without fully evacuating the system.

CHAPTER 5: Project Challenges

This chapter documents the significant challenges and hurdles experienced by this project. At the conclusion of the laboratory evaluation, EPRI presented alternate options for field evaluation. Ultimately, EPRI and CEC reached an agreement to cancel the remaining tasks, document findings, and lessons learned and conclude the project.

Project Partners

At the proposal phase, Mayekawa's role was to build and assemble the prototype heat pump and provide the field units. However, EPRI was notified by Mayekawa in March 2021 that it was no longer able to complete its scope at the Torrance, California facility. The reason was that Mayekawa was consolidating its manufacturing production in Tennessee and likely selling its Torrance facility. At that time, most of the staff remaining in California were focused on oil and gas projects and the production team was not suited to do custom manufacturing projects provided by external design. This situation resulted in EPRI procuring components and assembling the system. This system, due to its size, had to be assembled in-place and this led to weather delays. This process may have been simpler at dedicated fabrication facilities.

Component Delays

Supply chain issues significantly delayed the construction of the lab prototype since component lead times increased dramatically (in some cases from less than 20 weeks to about 40 weeks). Several components arrived much later than the lead time provided by the distributor at the time of quote, leading to delays and difficulties in coordinating time of laboratory personnel. The procurement process was challenging, especially when manufacturers did not have a California distributor. This led to delays with quotes, submittals, and purchase orders. EPRI dedicated considerable effort to finding alternates for these components and vendors; however, most custom components were only sold by limited manufacturers.

Contractor Delays

The complexity of this early technology readiness level technology presented challenges in construction, such as limited availability of qualified technicians and configuring CO_2 piping to accommodate four distinct operating modes. Interstate Mechanical, EPRI's contractor for laboratory work in Knoxville, faced severe staff shortages during the period of performance. This staff shortage was compounded by several weather events in Tennessee that caused a large number of service calls from their customers. EPRI proactively reached out to other mechanical contractors, but experienced similar issues.

EPRI transferred the contract to Interstate Mechanical's special projects team where staff shortages were less severe. The special projects team did not assist with dry fitting the components prior to brazing (a task previously done by Interstate with EPRI staff), which led to miscommunication with the contactors. For example, after multiple walk-downs, the contractor sent for CO_2 pipe brazing did not understand how to braze a steel-to-copper-iron joint. The contractor was a plumber and not an HVAC service technician. The latter, if qualified, would have the knowledge to perform the necessary brazes.

Host Site Issues

This project faced several challenges associated with the demonstration host sites. At the time of proposal, the following five sites were identified for field demonstration:

- SDG&E Energy Innovation Center
- Southland Envelope Production Plant (two units)
- CSUM – Machine Shop Building
- California State University Maritime Academy – Boat House

In July 2020, the facility manager at Southland Envelope left the company and Southland indicated they no longer wanted to be part of this project. In October 2020, SDG&E notified EPRI that the Energy Innovation Center would no longer be able to host this project as the facility was going to be sold. EPRI worked closely with SDG&E to identify any potential alternative buildings that could serve as replacements. In March 2021, the city of La Mesa public works department building was identified and the city provided a letter of intent. The outreach with SDG&E also identified the Sports Center Building on the University of San Diego campus. This list shows the host sites at the time of June 2021:

- CSUM – Machine Shop Building
- City of La Mesa – Public Works Department Building (two units)
- University of San Diego – Sports Center Building (two units)

EPRI worked on finalizing agreements and scheduling system installation with the sites and gathering baseline data where applicable.

In March 2023, the La Mesa notified EPRI that they would be undergoing energy service company energy efficiency upgrades that included the public works department building; therefore, they would not be able to participate in the study. La Mesa forwarded the project information to other city departments to scout replacements but received no responses.

EPRI met with Google buildings manager, Phillip Williams, in June 2023 and briefed him on the project. He was open to a demonstration at a Google building. EPRI then hosted a follow-up meeting in July 2023 with one of their facilities managers to further pursue the demonstration opportunity. Unfortunately, EPRI received no contact from Google following the meeting.

CSUM was one of the original host sites, but several staff changes at CSUM led to significant delays and ultimately CSUM decided not to participate in the demonstration. The list below outlines the changes at the site:

- September 2022, Professor Sheikh Nayeem left his position. He provided comments to the site agreement and a draft scope of work for the CSUM faculty and students. The new contact at CSUM was Mark Goodrich.

- February 2023, EPRI reached out to Mark Goodrich over the course of several months but never received a response. The Commission Agreement Manager sent an additional note to solicit CSUM's response.
- March 2023, Andrew Balmat, a new grant manager at CSUM, reached out to EPRI and indicated he was reviewing the EPRI documents.
- June 2023, Rizal Aliga, coordinator for sustainability and energy, took over as the new manager, and indicated to EPRI he was reviewing the latest site agreement.
- September 2023, Rizal Aliga notified EPRI that his position at CSUM was being dissolved and Scott Kern, the director of operations, would be taking over the project.
- October 2023, Scott Kern notified EPRI that Craig Dawson, a new sustainability coordinator, would be leading the project for CSUM. However, no contact information for Craig Dawson was provided. EPRI reached out for this information but received no response.
- November 2023, grant manager Andrew Balmat notified EPRI that Scott Kern had left his position at CSUM and provided the contract information for Craig Dawson. EPRI then sent the site agreement for review.
- January 2024, Craig Dawson sent a note to EPRI explaining that CSUM could no longer participate in this project. The specific development was the reduction in personnel, which reduced the bandwidth for accommodating such an effort. This reduction in personnel was coupled with severe campus-wide budget reductions and the executive management wished to only focus on the most critical tasks and avoid the optics of explaining a no-cost demonstration.

Following the loss of La Mesa and CSUM, EPRI worked with both SDG&E and SCE on outreach efforts to identify three new sites. Apart from the TTC, no additional buildings were identified. EPRI made design changes such that each prototype would provide less capacity to allow the USD Sports Center to host a total of four prototypes to satisfy the five required sites.

Prototype Technical Challenges

EPRI evaluated the prototype heat pump in different operating modes but faced challenges with achieving steady state behavior with a number of testing conditions. Troubleshooting issues with prototype operation is typical for early technology readiness level technologies; however, the complexity of this prototype made the process much more difficult. Several factors could be responsible for transience and underperformance, including ammonia compressor size, ammonia compressor speed, ammonia side EEV position, CO_2 compressor or pump speed, CO_2 side EEV position, indoor AHU fan speed, and refrigerant charge of the system. The interinfluence of these factors made it difficult to ascertain which factor was the primary cause of the system issues.

1. The oversized ammonia compressor may have been providing too much capacity during system start up. The high amount of refrigerant flow may have overwhelmed the EEV, and it could have caused the pressure spikes mentioned above.

2. The precise refrigerant charge for both ammonia and CO_2 needed for this equipment was unknown. This was partly due to the number of operating modes (cooling/heating with CO_2 compressor, cooling/heating with CO_2 pumps) the system had to accommodate, leading to complex piping configurations. EPRI and the design subcontractor estimated the refrigerant charge with modeling efforts and component specifications, but the unknown length of the CO_2 loop at the time of design resulted in much uncertainty on these estimates.
3. Troubleshooting during tests proved difficult, as several components had to be manually started or adjusted. It was unsafe for staff to be in close proximity to the system during operation because of the ammonia leakage incidents. This was largely due to the custom-made and experimental nature of the system, and field-ready designs with proper controls could accommodate automatic or remote adjustments.
4. Additionally, there were little literature and references available for this prototype due to the lack of commercial or market-ready ammonia heat pumps at this capacity range (11.5 RT–20 RT).
5. The laboratory evaluation provided much needed information on the operation of this prototype, including refrigerant leakage management, system start-up procedures, settings for compressor speed and EEVs, and refrigerant charge management. However, the difficulty in reaching steady state system behavior led EPRI to conclude that this technology is not mature enough for a safe and successful field demonstration at the USD site.

Significant Increase in Equipment Costs

EPRI's proposal allocated a total of $1,193,800 toward the field demonstration. However, the project faced several significant challenges with prototype construction. EPRI had to work with a construction equipment distributor on special orders when manufacturers did not have a California distributor, with over 30 percent markup in some cases. Supply chain issues and inflation resulted in further price increases. EPRI dedicated considerable effort in finding alternates for these components, but these compounding factors resulted in a much more expensive prototype than expected during the proposal phase. The expense of components and materials on the Knoxville prototype was almost $470,000.

EPRI estimated the cost of the field units using the latest information, including the prototype construction costs and contractors' quotes. The field units would be constructed by a custom HVAC contractor, and their markup would be 30 percent in addition to their labor costs. EPRI also received quotes from contractors on field design, with a total of $52,983 for a single field unit. It should be noted that the prototype system leveraged some existing EPRI equipment in the laboratory, such as a refrigeration skid, which would have to be procured for field units. Additional materials for weatherization not included in the lab prototype also had to be procured.

EPRI made design choices to minimize the field unit costs. Avoiding the use of oversized components can reduce cost and improve efficiency; however, component selection for the ammonia system was very limited for the 11.5 RT to 20 RT capacity range. This was particularly difficult when selecting a semi-hermetic compressor, one of the most expensive components, where the smallest capacity available was 40 RT. Even with reusing the Knoxville prototype for the SCE site, the estimated total to install all five field units was more than $2 million, far exceeding the original estimates.

Schedule

The CEC granted two, one-year, no-cost time extensions to this project to account for COVID delays, but the project was still behind schedule and would not be completed before the March 2026 liquidation date. Most of the delays stemmed from supply chain issues with the construction of the lab prototype since component lead times increased dramatically. EPRI had to procure most components with a California-based construction equipment distributor instead of directly procuring from manufacturers (or their preferred distributor), which led to delays with quotes, submittals, and purchase orders.

The complexity of this early technology readiness level technology also presented challenges in construction, such as limited availability of qualified technicians and configuring CO_2 piping to accommodate four distinct operating modes. The delays with the laboratory prototype directly delayed the field demonstration, since EPRI could not execute the contracts without at least verifying the performance of this technology. This was important to meet the solicitation requirements on capacity and efficiency, plus the prototype would be the only equipment providing space conditioning for the USD site.

EPRI Proposals on Alternate Paths

EPRI committed nearly the full cost-share amount ($433,400 out of $440,000) to resolve the technical challenges. This resulted in major research findings with CO_2 distribution loops that EPRI presented over several technical papers and conferences. However, based on the findings in laboratory evaluation and remaining budget and time, EPRI concluded that it was unsafe and unfeasible to finish the field demonstration with five units. EPRI proposed the following paths to address the issues:

1. Continue with a Field Demonstration at SCE Only

The deployment at USD according to project plan presented significant risks, including the leakage of ammonia near staff/students and insufficient cooling and heating for the facility. These risks could be minimized at the SCE TTC site. The TTC's engineers and technical specialists lead innovative technology evaluations to support SCE's Clean Power and Electrification Pathway. The laboratory setting would be less risky for unexpected issues with the heat pump prototype, which may include ammonia or CO_2 leakages. For this early technology readiness level prototype, many adjustments may also be required in the field, and the TTC's expertise with emerging technologies would be invaluable for such tasks. EPRI outlined some improvements to the Knoxville prototype, such as configuring the CO_2 piping for improved circulation of liquid CO_2.

EPRI met with SCE several times to discuss details of the field demonstration, and SCE signed a site demonstration agreement. The proposed configuration was to test the heat pump prototype using its general lab space (similar to the setup in EPRI's Knoxville laboratory), which already has space conditioning. This would allow the indoor air to act as a heat reservoir to test the prototype, and the site would still have space conditioning if the prototype did not provide adequate capacity.

CEC decided not to proceed with this approach because the solicitation required a minimum of five demonstration sites, and the CEC did not see the same value with a demonstration of only one site.

2. Demonstrate an Alternate but Related Technology at Five Sites

EPRI identified an emerging high efficiency rooftop unit from Daikin, which uses R-32 refrigerant. R-32 has a GWP of 675 and meets the solicitation requirement of GWP less than 750. This equipment may also include several advancements developed in preparation for the U.S. DOE's Commercial Building Heat Pump Technology Challenge. EPRI proposed to demonstrate these advanced reversible heat pumps with R-32 refrigerant in place of the ammonia heat pumps at five sites and two California climate zones according to the requirements of the solicitation. R-32 based rooftop units are considered emerging technologies and are just being introduced into the market. More than 70 percent of all commercial building floor space in California is conditioned with rooftop units, with an average equipment age of more than 10 years. This HVAC fleet will be replaced with low GWP refrigerants starting in January 1, 2025, according to state (GWP less than 750) and federal (GWP less than 700) regulations.

This alternate approach, proposed by EPRI, has two main advantages: 1) the system is a packaged solution provided by a manufacturer, so the equipment would be under warranty and there would be no risk to the demonstration sites; 2) high efficiency R-32 equipment has seen limited market penetration in the state, and this project would provide much needed field data for utilities to consider for including such technologies in their customer incentive programs. While many R-32 products are advertised on manufacturers' websites, most manufacturers have yet to offer and sell these systems to distributors/consumers. R-32 has emerged as one of the most widely used low GWP refrigerants around the world, with significant uptake in Europe and Asia. As California and the United States phase down the use of high GWP gases, performance data of installed equipment in the field is necessary to understand the impact of wider adoption.

The CEC decided not to proceed with this approach, because the project was funded for the ammonia heat pump and this alternate technology may not have scored high enough on the original solicitation to be awarded.

CHAPTER 6: Technology Transfer

This chapter describes the technology transfer effort, lists information on products developed, and provides information on venues for disseminating the products. This is a crucial step in introducing the ammonia heat pump with CO_2 distribution loop to a wider audience, since there are no such commercially available products in the world and technology transfer is an important objective of all Electric Program Investment Charge (EPIC) projects.

California-Focused Events

As this project was funded by the EPIC program, the research results were shared at the 2023 EPIC Symposium and several Energy Transition Coordinating Council (ETCC) Emerging Technologies summits.

The EPIC Symposium is hosted by the CEC and features the latest developments and innovations in projects funded through the EPIC program. Symposium attendees include various clean energy professionals, researchers, grantees, policy makers, industry leaders, technology developers, and entrepreneurs.

The ETCC is a collaboration between CEC, Pacific Gas and Electric Company, SCE, Southern California Gas Company, SDG&E, and Los Angeles Department of Water and Power. The focus of the ET Summit is to accelerate the market adoption of emerging technologies that are underutilized across different end use sectors.

As part of the technology transfer activities, the project findings were presented at these events:

- Emerging Technologies Summit 2022
    - Session: Low-GWP Refrigerants Projects
- Emerging Technologies Summit 2023
    - Session: Low-GWP Refrigerants: The 9-Million Metric Ton CO_2E Gorilla in the Room
- EPIC Symposium 2023
    - Session: Building Energy Efficiency and Improving Affordability
- Emerging Technologies Summit 2024
    - Session: Low-GWP Refrigerants: New Equipment vs. Retrofits

Technical Conferences

The research findings were presented at HVAC and energy efficiency industry conferences to audiences consisting of key market participants, including educators, researchers, energy efficiency advocates and state and federal energy agencies. As part of the technology transfer activities, conference papers and presentations were submitted to several organizations:

- 14th International Energy Agency Heat Pump Conference, hosted by the Technology Collaboration Programme on Heat Pumping Technologies by the International Energy Agency (Robinson et al, 2023)
    - Topic Area: Innovation and R&D
    - Paper Title: On the Use of CO_2 as a Heat Distribution Fluid for Sustainable Ammonia Heat Pump Solutions. https://heatpumpingtechnologies.org /publications/presentation-no-1067-on-the-use-of-co2-as-a-heat-distribution-fluid-for-sustainable-ammonia-heat-pump-solutions-14th-iea-heat-pump-conference-chicago-usa/
    - 2024 Summer Study on Energy Efficiency in Buildings, hosted by American Council for an Energy-Efficient Economy (Tam et al, 2024)
    - Panel 3: Commercial Buildings: Technologies, Design, Operation, and Industry Trends.
    - Paper Title: A Near-Zero-GWP Heat Pump System for All-Electric Heating & Cooling in California. https://www.aceee.org/sites/default/files/proceedings /ssb24/pdfs/A%20Near-Zero-GWP%20Heat%20Pump%20System%20for%20All-Electric%20Heating%20&%20Cooling%20in%20California.pdf

Additional EPRI Activities

EPRI led separate recurring meetings with utilities that provided cost share for this research effort, primarily SCE and SDG&E. Meetings with the SCE advisors typically occurred once a quarter and were accompanied by a quarterly progress report. Meetings with SDG&E typically occurred every three to four weeks, and updates were typically provided verbally during the meeting.

Additional meetings included EPRI advisory meetings and webcasts such as the Heat Pump Working Council. This project was highlighted in a 2024 Heat Pump Working Council meeting, which focused on the refrigerant regulatory landscape in the United States. The meeting highlighted the need for advanced technologies that employ low GWP natural refrigerants, and this project showcased the potential of ammonia as a refrigerant for space conditioning.

Furthermore, EPRI held a meeting with the California Air Resources Board to discuss the prospects of natural refrigerants for the State of California. The preliminary results from this project were shared, including the use of ammonia for space conditioning and the use of CO_2 as a space conditioning distribution fluid.

CHAPTER 7: Conclusion

This project evaluated a first-of-its-kind natural refrigerant based heat pump. The design and model efforts with commercially available products showed promising results. However, the laboratory evaluation of the prototype revealed several technical challenges with this early-stage technology that still need to be overcome for successful demonstrations and eventual commercialization. The project planned to demonstrate the technology with five field units, one at SCE's TTC, and four at USD's Sports Center. Based on the laboratory results, the team recommended against the deployment of four prototype heat pumps at the USD site due to risk of refrigerant exposure and insufficient heating and cooling capacities. The CEC ultimately opted to cancel the field demonstration part of this project due to additional budget and schedule concerns.

While the field demonstration cancelations are a disappointing result, this project was an unequivocal success story for advancing the technology and executing critical development work to demonstrate the feasibility and technical challenges with a novel heat pump system. Technical challenges are typical for research projects, particularly when involving early readiness levels of the technologies. Even though the project did not perform field demonstrations, the laboratory evaluation accomplished a significant milestone for the HVAC industry, which is demonstrating the circulation of supercritical CO_2 with a high-pressure CO_2 pump for space heating. This accomplishment is a substantial development for electrification and decarbonization efforts. The results also amassed interest from U.S. DOE and the HVAC industry as hazardous refrigerants (for example, ammonia and propane) are emerging as long-term sustainable low-GWP solutions, and the CO_2 distribution technology can be an excellent fit for heat pump application.

To further develop this technology, and to better understand the extent to which ammonia and CO_2 may be applied effectively in HVAC systems, several additional efforts should be considered:

- The pumped CO_2 distribution loop should be further researched to optimize its operation. From a thermodynamic standpoint, this is an inherently more effective heat transfer process than conventional chilled or hot water loops. If mature, this technology can be broadly applied to HVAC systems to help realize significant energy and cost savings for the end users.
- Another recommendation stemming from this research effort is the need for a larger variety of offerings from manufacturers. The only two commercially available semi-hermitic ammonia compressors were the selected 40 RT product and an 8 RT compressor that was only available as part of a packaged chiller system. As a result, the 40 RT compressor was loaded to 50 percent to match the needs of the prototype, and its controls may have caused some of the technical challenges. A bigger selection will

give end users access to products that more closely match their cooling requirements and facilitate the transition away from HFC-based technologies.

- The concept investigated in this project should be extended to additional working fluids. For primary refrigerants, propane (R-290) is beginning to establish itself as the next generation refrigerant of choice in Europe in light of strict regulations around refrigerant GWP and PFAS (per- and polyfluoroalkyl substance) concerns. A similar approach with propane and CO_2 can be an efficient alternative to conventional HVAC systems in the United States. Moreover, other secondary fluids should be considered. CO_2 has several benefits over water loops, but its high-pressure requirement can be challenging for installers and contractors. There is an increasing number of new fluids being developed by manufacturers in the market, and while there are some environmental concerns over PFAS issues, their impact on energy efficiency needs to be well understood.
- As the industry transitions toward hazardous refrigerants, the secondary loop configuration will be deployed in greater numbers. This creates an opportunity to integrate additional components such as thermal energy storage. While there may be some penalties on energy efficiency due to the harsher operating conditions during the charge cycle, the grid benefits of flexibility may be substantial.

GLOSSARY AND LIST OF ACRONYMS

Term	Definition
ACGIH	American Conference of Governmental Industrial Hygienists
AHU	air handling unit
AIM Act	American Innovation and Manufacturing Act of 2020
ASHRAE	American Society of Heating, Refrigerating and Air-Conditioning Engineers
Btu/lbm	British thermal units per pound mass
CEC	California Energy Commission
COPh	coefficient of performance, heating
CO_2	carbon dioxide
CSUM	California State University Maritime Academy
EER	energy efficiency ratio
EEV	electronic expansion valves
EIA	Energy Information Administration
EPCRA	Emergency Planning and Community Right-To-Know Act
EPRI	Electric Power Research Institute
ETCC	Energy Transition Coordinating Council
GWP	global warming potential
HFC	Hydrofluorocarbon
HVAC	heating ventilation and air conditioning
HX	heat exchanger
Hz	Hertz
IEER	integrated energy efficiency ratio
kW	kilowatt
kWh	kilowatt-hour
NH_3	ammonia
NIOSH	National Institute for Occupational Safety and Health
OSHA	Occupational Safety and Health Administration
PFAS	Per- and Polyfluoroalkyl Substances
ppm	parts per million
psig	pounds force per square inch gauge
R&D	research and development

Term	Definition
RT	refrigeration ton
RTU	rooftop unit
SCE	Southern California Edison
SDG&E	San Diego Gas and Electric
SEER	seasonal energy efficiency ratio
TMY3	typical meteorological year 3
TTC	Technology Test Center
UL	Underwriters Laboratories
U.S. DOE	U.S. Department of Energy
U.S. EPA	U.S. Environmental Protection Agency
USD	University of San Deigo
VDC	volts of direct current
VFD	variable frequency drive
W	watt

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Project Deliverables

- CPR Meeting #1 Report
- TAC Meeting #1 Report
- Report on Engineering Design of Optimized Laboratory Prototype
- Updated System Design for Field Deployment Report
- Kick-off Meeting Benefits Questionnaire
- Mid-term Benefits Questionnaire
- Technology/Knowledge Transfer Plan

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APPENDIX A: Technical Advisory Committee

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APPENDIX A: Technical Advisory Committee

The TAC members who agreed to support the project in an advisory role are:

- Anachal Kohli – California Air Resources Board
- Birk Jones – Sandia National Laboratories
- Cara Martin – Optimized Thermal Systems R&D
- Dominique Michaud – San Diego Gas & Electric
- George Gurlaskie – Duke Energy
- Jerine Ahmed – Southern California Edison
- Keshmira Engineer – Bonneville Power Administration
- Mark Alatorre – Pacific Gas and Electric Company
- Patrick Phelan – Arizona State University
- Reinhard Radermacher – University of Maryland
- Robert. Weber – Bonneville Power Administration
- Tyler Sybert – San Diego Gas & Electric

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APPENDIX B: Prototype Heat Pump Bill of Materials

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APPENDIX B: Prototype Heat Pump Bill of Materials

Table B-1: Valve List

System	Type	Part Number	Manufacturer	Schematic Tag
CO_2	Pressure Relief	0264BD02-KG - D23	Kunkle	V1
	Check Valve	CVN1118K0006	Refrigera	V2
	Ball Valve	A 17865XHP	Mueller Streamline	V3
	Ball Valve	A 17864XHP	Mueller Streamline	V4
	Pressure Relief	0264BD02-KG - D23	Kunkle	V5
	EEV	027H7202 - CCMT 8	Danfoss	V6
	Check Valve	CVN1118K0006	Refrigera	V7
	Pressure Relief	0264BD02-KG	Kunkle	V8
	Check Valve	CVN1022K0000	Refrigera	V9
	EEV	027H7201 - CCMT 4	Danfoss	V10
	Pressure Relief	0264BD02-KG	Kunkle	V11
	Check Valve	CVN1022K0000	Refrigera	V12
	EEV	027H7201 - CCMT 4	Danfoss	V13
	Pressure Relief	0264BD02-KG - D23	Kunkle	V14
	Ball Valve	A 17864XHP	Mueller Streamline	V15
	Ball Valve	A 17865XHP	Mueller Streamline	V16
	Ball Valve	A 17864XHP	Mueller Streamline	V17
	Check Valve	TBD	TBD	V18
	Ball Valve	A 17864XHP	Mueller Streamline	V19
	Ball Valve	A 17864XHP	Mueller Streamline	V20
	Pressure Relief	0264BD02-KG - D23	Kunkle	V21
	Ball Valve	A 17865XHP	Mueller Streamline	V22
	Ball/Solenoid	TBD	TBD	V23
NH_3	Pressure Relief	0264BD02-KG - K22	Kunkle	V24
	EEV	Actual EEV - 027H1186	Danfoss	V25
		EEV Body - 027H1166	Danfoss	
		EEV Actuator - 027H9120	Danfoss	
	Check Valve	027X0184	Danfoss	V26
	EEV	Actual EEV - 027H1186	Danfoss	V27
		EEV Body - 027H1166	Danfoss	
		EEV Actuator - 027H9120	Danfoss	
	Check Valve	027X0184	Danfoss	V28
	Pressure Relief	0264BD02-KG - K22	Kunkle	V29

System	Type	Part Number	Manufacturer	Schematic Tag
	Pressure Relief	0264BD02-KG - K22	Kunkle	V30
	4-Way Valve	BV44X028X0001	Refrigera	V31
	Check Valve	027X0184	Danfoss	V32
	Pressure Relief	0264BD02-KG - K22	Kunkle	V33

Source: EPRI, 2023

Table B-2: Compressor List

System	Type	Part Number	Manufacturer	Schematic Tag
NH_3	Ammonia Compressor	NHM30	Mayekawa	C1
CO_2	CO_2 Compressor	CD2000H/OP	Dorin	C2

Source: EPRI, 2023

Table B-3: Filter List

System	Type	Part Number	Manufacturer	Schematic Tag
CO_2	Filter Dryer	023Z8412	Danfoss	F1
	Filter Dryer	023Z8412	Danfoss	F2

Source: EPRI, 2023

Table B-4: Oil Separator List

System	Part Number	Manufacturer	Schematic Tag
NH_3	92400717	Temprite	OS1
CO_2	13504352	Temprite	OS2

Source: EPRI, 2023

Table B-5: Accumulator List

System	Part Number	Manufacturer	Schematic Tag
NH_3	96108200 (SA8-20)	Temprite	A1
CO_2	AV-402	Iso-Therm	A2

Source: EPRI, 2023

Table B-6: Receiver List

System	Part Number	Manufacturer	Schematic Tag
CO_2	RV-1403	Temprite	R1

Source: EPRI, 2023

Table B-7: Pump List

System	Part Number	Manufacturer	Schematic Tag
CO_2	820-DS-050-VSD-B	Hy-Save	P1
	820-DS-050-VSD-B	Hy-Save	P2
	820-DS-050-VSD-B	Hy-Save	P3
Glycol	2400-45SY	Taco	P4

Source: EPRI, 2023

Table B-8: Heat Exchanger List

System	Type	Part Number	Manufacturer	Schematic Tag
NH_3/ CO_2	Plate HX	AXP52 AN-150H	Alfa Laval	HX1
NH_3	Air-to-Refrigerant	A+OV23I-32-132-36.5C-1-0200L-ACD-SB	Colmac Coil	HX2
CO_2	Air Handler	HCA40AAAAAAA-GAB-AEB-CC-ABAM	Magic Aire	HX3
	High Pressure Air-to-Refrigerant	32x48-4R-0.375/144	Super Radiator Coil	
	Air Handler	HCA40AAAAAAA-GAB-AEB-CC-ABAM	Magic Aire	HX4
	High Pressure Air-to-Refrigerant	32x48-4R-0.375/144	Super Radiator Coil	

Source: EPRI, 2023

Table B-9: Superheat controller list

System	Part Number	Manufacturer	Schematic Tag
NH_3	EKC315A/084B7086 - Controller	Danfoss	SH1
	060G6323 - Pressure Transmitter		
	084N0038 - Temperature Sensor		
	EKC315A/084B7086 - Controller	Danfoss	SH2
	060G6323 - Pressure Transmitter		
	084N0038 - Temperature Sensor		
CO_2	EKE1C/080G5400 - Controller	Danfoss	SH3
	060G6343 - Pressure Transmitter		
	084N0038 - Temperature Sensor		
	EKE1C/080G5400 - Controller	Danfoss	SH4

System	Part Number	Manufacturer	Schematic Tag
	060G6343 - Pressure Transmitter		
	084N0038 - Temperature Sensor		
	EKE1C/080G5400 - Controller	Danfoss	SH5
	060G6343 - Pressure Transmitter		
	084N0038 - Temperature Sensor		

Source: EPRI, 2023

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APPENDIX C: Additional Host Site Preparation Activities

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APPENDIX C: Additional Host Site Preparation Activities

California State University Maritime Academy (CSUM)

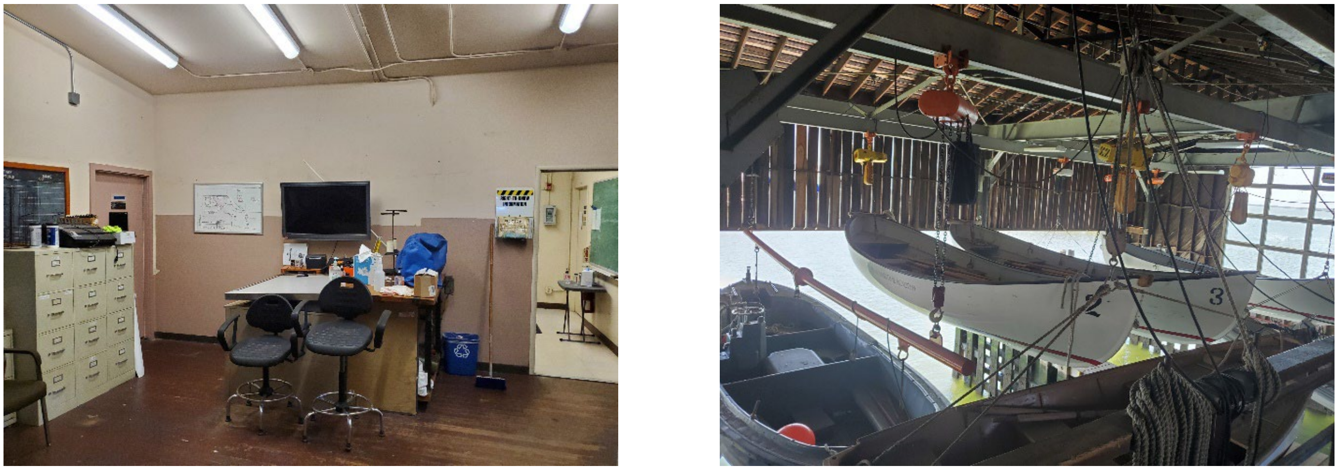
EPRI visited the site on June 18th, 2021, with Interface Engineering and toured the buildings with the CSUM project managers. Interface Engineering is an engineering design firm that was recommended by CSUM to be a contractor. Interface Engineering's role was to design the field unit in collaboration with EPRI, identify specific configurations needed for the host buildings, coordinate structural calculations, review any safety requirements, and obtain necessary permits. Two different buildings were designated for the field demonstration: the machine shop and boat house. The team reviewed the existing HVAC system at the machine shop and concluded the building was a good fit for demonstration the prototype heat pump. The team inspected possible locations for the installation, including the rooftop area and open spaces close to the building (Figure C-1). One key consideration was the heat pump must have clearance away from any open doors or windows due to the risk of ammonia leakages.

Figure C-1: CSUM Site Machine Shop Building Interior and Proposed Location for Installation

Source: EPRI, 2021

On the other hand, the tour of the boat house revealed the building was split into two sections, a small and conditioned office area, and a large unconditioned dock area that is exposed to the ambient conditions (Figure C-2). It was concluded that the office area was too small, and the dock area would not be a good fit for a demonstration since it is exposed to ambient conditions and difficult to obtain steady state data.

Figure C-2: CSUM Site Boat House Interior Office and Exposed Dock Area



Source: EPRI, 2021

The team toured an additional CSUM building that was available to host the prototype heat pump, which was a prefabricated building used as a temporary office (Figure C-3). However, the team concluded that this office was too small to host the heat pump after touring the building.

Figure C-3: CSUM Site Alternate Office Building

Source: EPRI, 2021

CSUM project managers requested a contract for CSUM faculty and students to participate in the evaluation of the prototype heat pump. This project presented a unique opportunity for their engineering students to study a prototype system in person. CSUM and EPRI also discussed the potential for a dashboard that allows faculty, researchers, and students to view the system’s performance in real time. EPRI was able to accommodate the request and included this provision in the draft site agreement between the two parties.

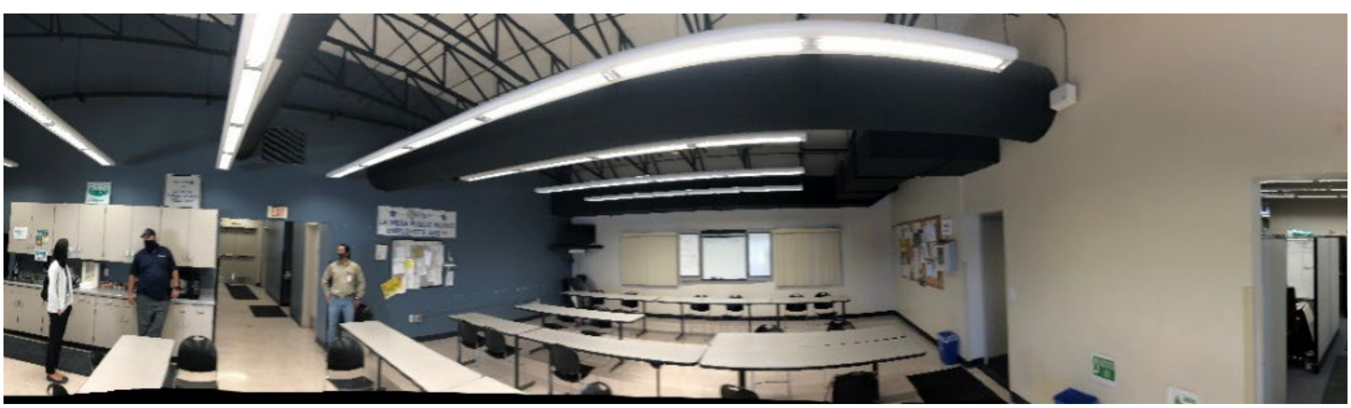
City of La Mesa (CLM)

EPRI visited the site on June 14th, 2021, to inspect the building and existing equipment. It was concluded that the office building is a good fit for the capacity of two prototype heat pumps. The building had two existing heat pumps at 20 RT and 12 RT, both of which contribute to space conditioning of the office space (Figure C-4 and Figure C-5). EPRI installed HOBO sensors on June 10th, 2021, to establish baseline data, which informed typical occupancy and behavior patterns (e.g. thermostat setpoint).

Figure C-4: CLM Site Installed HVAC Unit

Source: EPRI, 2021

Figure C-5: CLM Site Interior Office Space



Source: EPRI, 2021

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APPENDIX D: Prototype Heat Pump Electrical Connections

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APPENDIX D:
Prototype Heat Pump Electrical Connections

Figure D-1 through Figure D-4 show the electrical drawings for the prototype heat pump. This information was conveyed to the host sites to ensure adequate electrical capacity on site.

Figure D-1: Power Distribution Schematic for the Prototype Heat Pump (part 1)

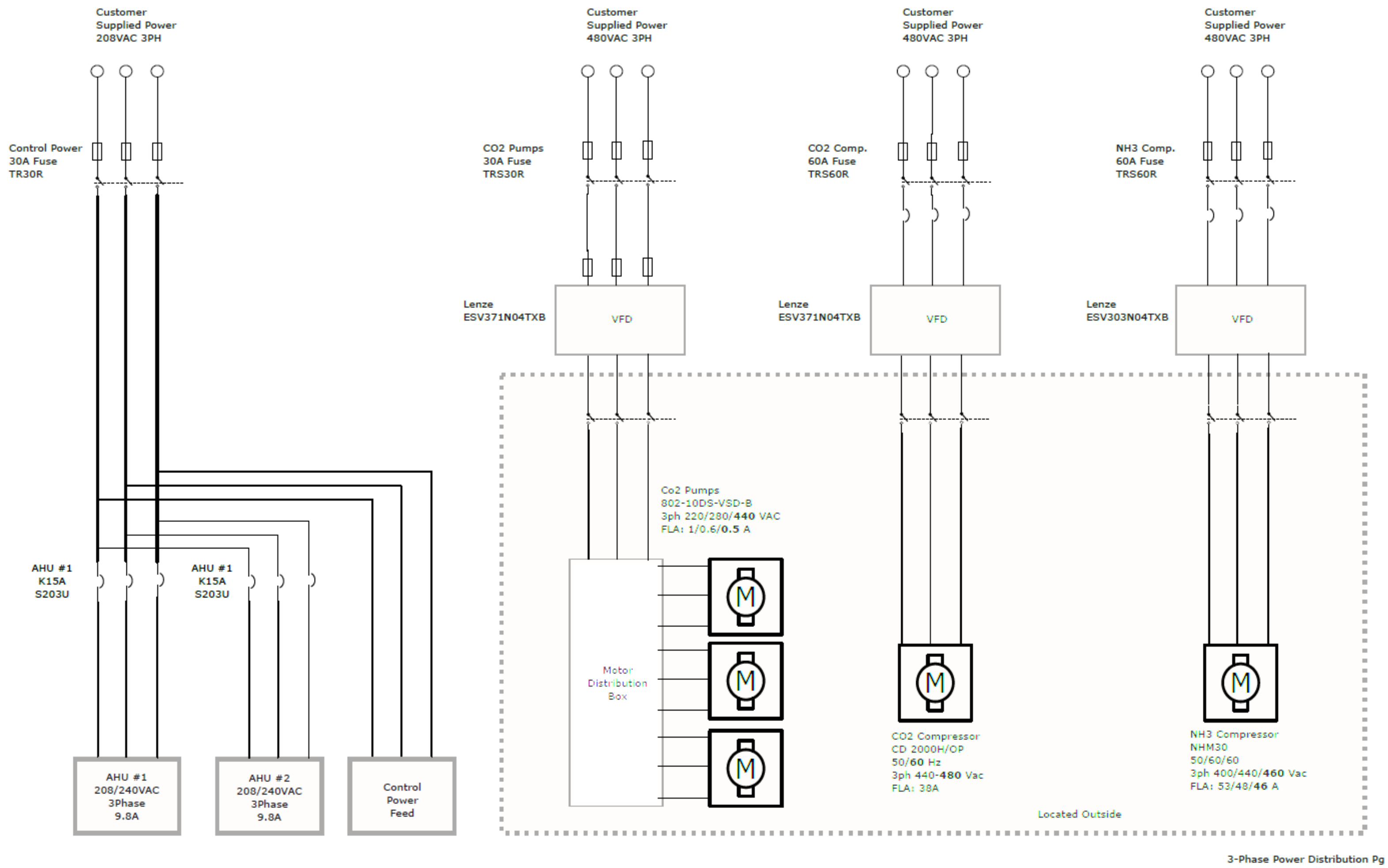


Figure D-2: Power Distribution Schematic for the Prototype Heat Pump (part 2)

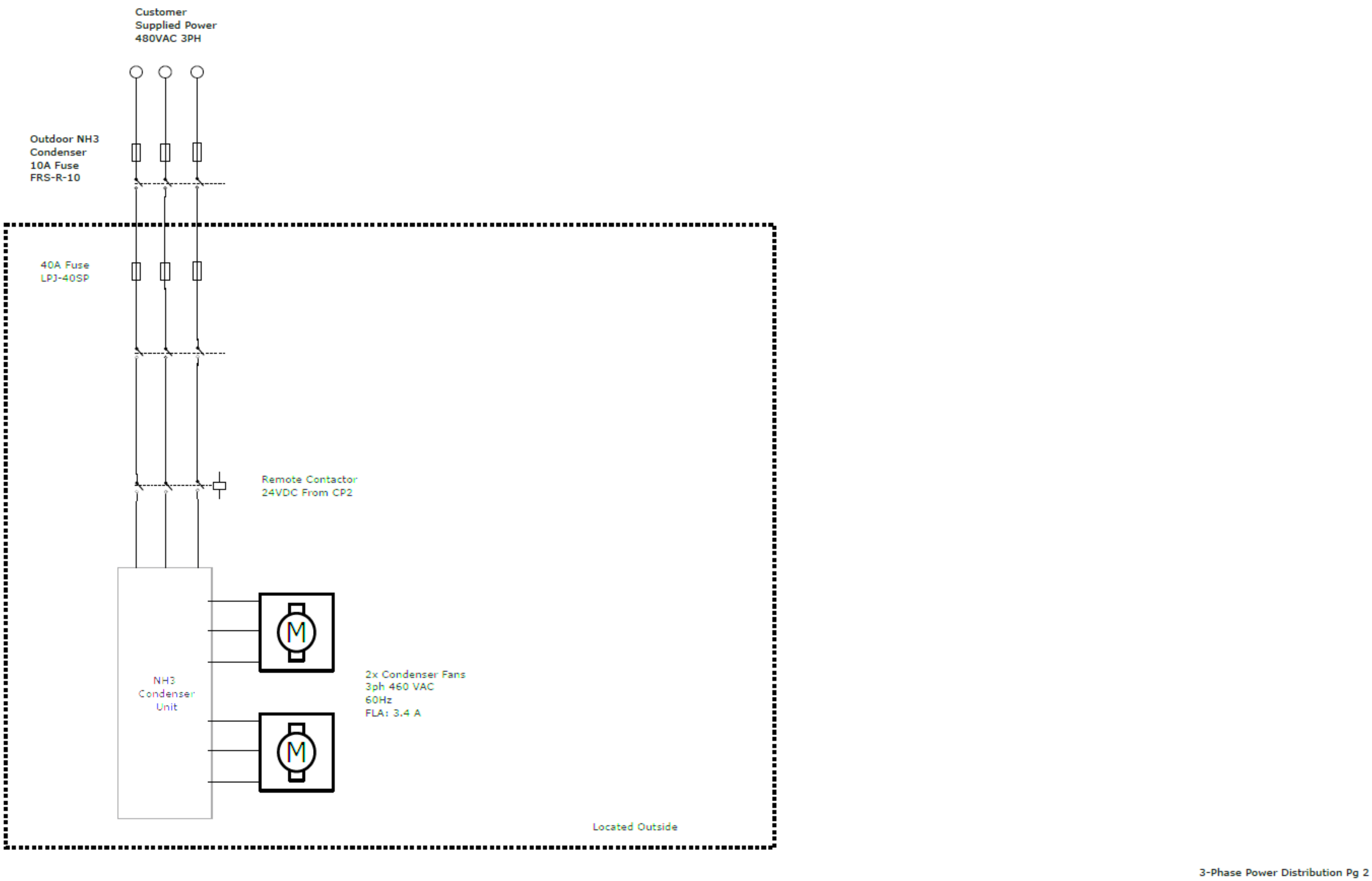
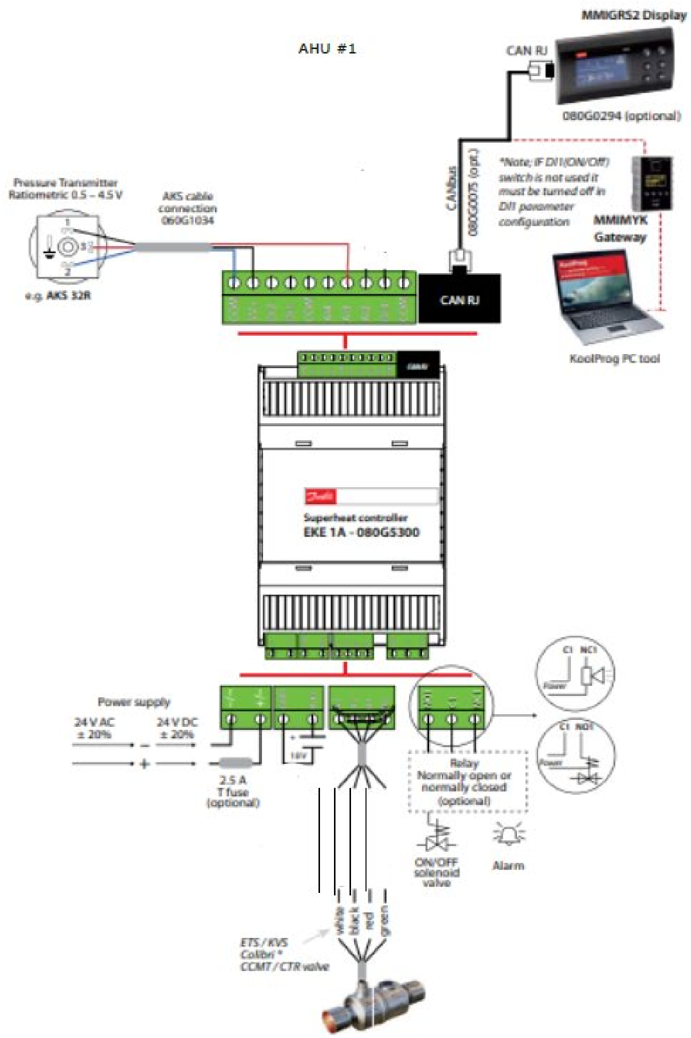
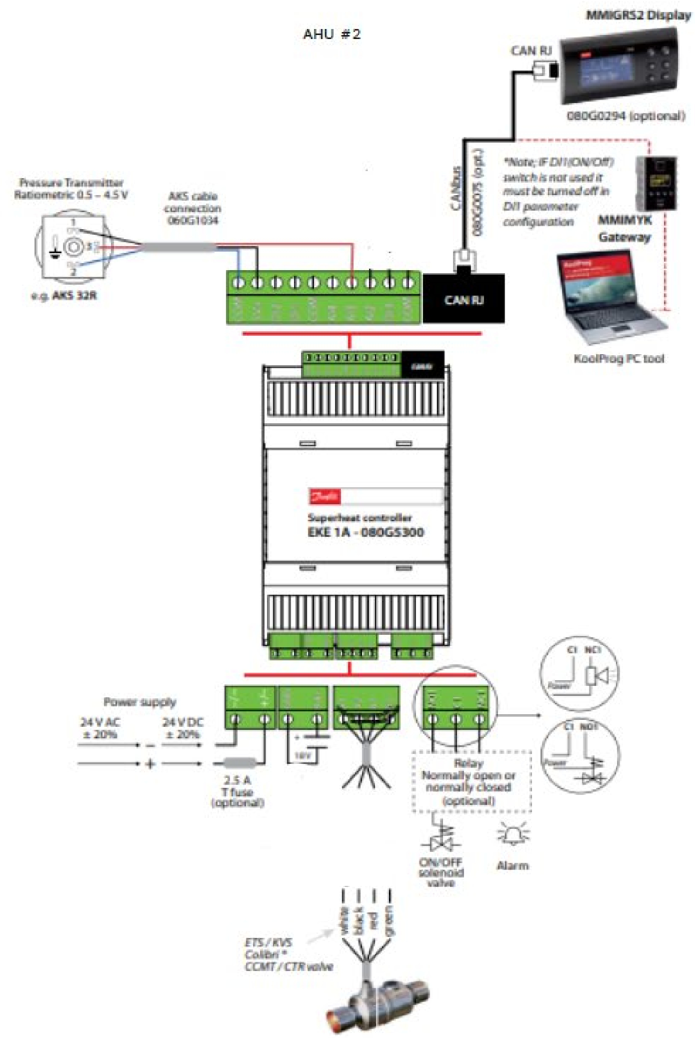


Figure D-3: CO2 Superheat Controller Schematic





Co2NH3 Electrcal As Built- Control Devices CP1_4

Figure D-4: Heat Pump Controls Cabinet Schematic