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Affordable Near- and Medium-Term Solutions for Integration of Low-GWP Heat Pumps in Residential Buildings

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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 Company — 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 emission 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.

Affordable Near- and Medium-Term Solutions for Integration of Low-GWP Heat Pumps in Residential Buildings is the final report for EPC-19-016 conducted by the University of California, Davis and TRC Companies, Inc. The information from this project contributes to the Energy Research and Development Division's Electric Program Investment Charge 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

California has set ambitious climate goals to achieve carbon neutrality by 2045 including an 85 percent reduction in greenhouse gas emissions. Electrification of space heating is one of the first steps toward a broader goal of decarbonizing buildings in the United States. Heat pump technologies provide greater heating efficiency than gas furnaces while also emitting less greenhouse gas. Beginning in 2025, residential unitary heat pump systems in California must use a refrigerant with lower global warming potential than what is commonly used today. This presents new challenges for manufacturers and installers as heat pump installations scale up to meet the new state requirement.

This report describes the results from a project aimed at developing and demonstrating affordable and efficient heat pumps using refrigerants with low global warming potential (<750). The project team developed a near-term solution to address the existing need for affordable and efficient heat pump options that meet the upcoming refrigerant regulations. The medium-term solution advances air-to-water heat pump technology that offers a solution if refrigerant regulations become more stringent, requiring the use of highly flammable or toxic refrigerants.

Demonstrations of a new heat pump technology developed by Rheem Manufacturing Company were performed in 10 homes in California in climate zones 2 and 12. Results showed an increase in utility costs for many sites, though greenhouse gas emissions were reduced by 44 to 90 percent for the heat pump systems relative to the natural gas heating systems replaced in the project. These results demonstrate significant progress toward decarbonization, while also documenting user experience during the retrofit process, including the contractor and tenant experience.

Lab testing and modeling of the microchannel polymer heat exchanger, which was developed to improve the performance of air-to-water heat pumps, showed improvement over conventional coils. The research team validated heat exchanger performance in the laboratory, and modeling of injection molded versions showed 15 to 20 percent improved effectiveness compared to the commercial coil. Laboratory testing showed a 5 percent improvement in coil effectiveness resulted in 5 percent improved efficiency for the air-to-water heat pump system. The heat exchanger also showed the potential to lower the cost by 8 to 20 percent compared to similar commercial coils.

Keywords: Low Global Warming Potential Refrigerant, Heat Pump, Air-to-Water, Field Evaluation

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Executive Summary

Project Purpose and Approach

California is moving aggressively to electrify all energy sectors with a goal of being carbon-neutral by 2045. This will require switching the primary fuel for heating buildings from natural gas to electricity. To that end, California established a goal to install 6 million heat pumps for space conditioning (and water heating) by 2030. New homes are being pushed toward heat pumps with the 2025 Building Energy Code, which now uses heat pumps for both space conditioning and water heating as the baseline for single-family homes in all climate zones. By the end of this decade, all space conditioning (and water heating) systems installed in new and existing homes in California are expected to be heat pumps given the California Air Resources Board's unanimous approval of a proposed ban on the sale of new natural gas-powered water and space heaters by 2030 (CARB, 2022a). At the same time, limitations on the global warming potential of the refrigerants used in heat pumps are becoming increasingly stringent, requiring innovations in equipment design. Meeting the state's energy and carbon goals while addressing affordability will require heat pumps that can compete with furnaces on both upfront and operating costs.

This project developed and demonstrated next-generation heat pump technology that achieves high-efficiency heating and cooling, significantly reducing greenhouse gas emissions from buildings, at a lower cost relative to similar performing equipment. The project team pursued this goal amid a changing global landscape around refrigerant acceptability, meaning that, in addition to cost and efficiency, the team also needed to consider future changes in refrigerant regulations.

The team pursued two pathways, advancing both near- and medium-term solutions. Development of the near-term solution was completed by Rheem Manufacturing Company (Rheem), and performance verification was conducted by University of California Davis at the Western Cooling Efficiency Center. Following development, field testing was conducted to measure the installed performance of the new heat pump technology. For the medium-term solution, advancements in heat exchanger design were conducted through modeling and small-scale testing before testing a full-scale heat exchanger coupled to an air-to-water heat pump in the laboratory.

Key Results

Near-Term Solution

This project, in partnership with Rheem, developed and demonstrated a next-generation heat pump technology using R-454B refrigerant which has a low global warming potential of 466. The global warming potential of refrigerant is measured relative to carbon dioxide, which is defined as having a global warming potential of 1, and quantifies the impact a refrigerant has on global warming if released into the environment. It is inevitable that some refrigerants will be released through leaks in the refrigerant system, servicing, installation, and manufacturing.

The most common refrigerant used currently is R-410A which has a global warming potential of 2,088 so the alternative refrigerant used in the heat pumps for this project has a much lower environmental impact. The system was designed as a standard split-system air-source heat pump with a variable-speed blower and compressor.

Annual heating, ventilation, and air conditioning energy usage varied significantly across the 10 demonstration sites due to factors such as occupant behavior, home vintage, and existing equipment efficiencies. Upon comparing the pre- and post-retrofit periods, researchers observed electricity savings for cooling ranging from -41 percent to 68 percent across the sites. Although the heat pump systems demonstrated higher efficiency compared to the systems they replaced, the substantial variance in energy impacts among the homes can be attributed to occupant behavior and the condition of the baseline equipment. This includes changes in operating hours and thermostat adjustments, as reported in participant surveys.

Researchers evaluated the impact of this fuel switching technology on ratepayers by estimating the heating and cooling utility cost implications of the retrofit heat pump. The calculated annual costs for space heating and cooling in both the pre-retrofit gas scenario and the post-retrofit heat pump scenario were compared. The findings indicate that, based on the analyzed rate structure, operating the heat pump likely incurred higher costs compared to the baseline natural gas system. Specifically, there was an increase in energy costs ranging from 3 percent to 27 percent for five out of the eight sites for which there was valid data. The team observed 9 percent and 2 percent cost savings for Site 2 and Site 10, respectively. Site 6, which had an estimated 46 percent cost savings, switched tenants during the monitoring period, which may have impacted the result.

To put this in perspective, both gas and electricity rates and rate plans used in this analysis have experienced significant price increases. The average annual increase over a 15-year period has been 5.0 percent for gas and 7.2 percent for electricity, but in the past 5-year period the average annual increase has been much higher at 9.6 percent for gas and 13.8 percent for electricity. Based on an equivalent unit of energy delivered to a home, electricity was 7.3 times more expensive than gas with the 2024 rate, thus exceeding the cost savings potential of electric heat pumps over natural gas heaters even though they are two to five times more efficient. If electricity prices continue to increase at a higher rate than gas prices, this will significantly impact California's electrification goals and reduce the pace of market transformation to electric heat pumps, especially in existing residential homes.

Evaluating the impact of the retrofit from the perspective of decarbonization, the results paint a different picture. This project specifically evaluated the indirect emissions associated with the heat pump technology compared to the emissions of the baseline gas systems. Indirect emissions are those associated with the electricity used by the systems and depend on the electricity generation mix. This analysis did not include the emissions associated with direct release of refrigerant into the environment which would be presumed to be higher for the baseline air conditioners using R-410A. The greenhouse gas emissions reductions from the retrofit heat pump ranged from 44 percent to 98 percent across the sites, with larger savings observed for sites in climate zone 2 compared to those in climate zone 12. This discrepancy can be attributed to the fact that heating constitutes a larger portion of energy demand than cooling in climate zone 2 compared to climate zone 12.

Medium-Term Solution

Innovative microchannel polymer heat exchangers (MPHX) were designed to improve the performance of air-to-water heat pumps. The MPHX's water-to-air coil was tested and compared to a typical commercial fin-tube coil constructed from copper and aluminum. The construction of the MPHX using stereolithography three-dimensional printing did not allow testing of the optimal geometry, and the results showed similar performance between the MPHX and the commercial coil. While the performance was similar, this is a major improvement compared to other plastic heat exchanger designs. Key findings are summarized as follows:

1. **MPHX Performance:** The MPHX was tested individually at chiller setpoints of 41.0 degrees Fahrenheit (5 degrees Celsius) and 55.4 degrees Fahrenheit (13 degrees Celsius) (one ton and two-ton). The results showed that the heat transfer rate of the MPHX ranged from 2,140 to 4,780 watts, and its effectiveness ranged from 0.63 to 0.83, with an average difference of 13.3 percent between the air-side and water-side heat transfer rates.
2. **Commercial Coil Performance:** The commercial coil underwent similar individual testing under the same conditions, achieving an effectiveness range of 0.58 to 0.85, with an average difference of 15.9 percent between air-side and water-side heat transfer rates.
3. **MPHX vs. Commercial Coil:** A direct comparison at the two-ton chiller setpoint revealed that the MPHX consistently outperformed the commercial coil in both heat transfer rate (2,150 to 3,600 watts for the MPHX, and 1,920 to 3,090 watts for the commercial coil) and effectiveness (0.65 to 0.79 for the MPHX, and 0.58 to 0.78 for the commercial coil) under equivalent testing conditions. Specifically, the MPHX demonstrated improvements of 1.3 percent and 2.4 percent in effectiveness at pressure drops of approximately 48 pascal (Pa) and 114 Pa, respectively.

   At the one-ton setpoint, the comparison showed a similar trend, with the MPHX consistently achieving higher effectiveness at higher air-side pressure drops. The commercial coil outperformed the MPHX at approximately 51 Pa, showing a 4.67 percent higher effectiveness. However, at air-side pressures of approximately 86 Pa, 121 Pa, and 129 to 132 Pa, the MPHX demonstrated improved effectiveness by 0.2 percent, 9 percent, and 10.4 percent, respectively.
4. **Hydronic Coil Coupled with Heat Pump Testing:** Integrating the commercial coil and MPHX with an air-to-water heat pump confirmed that the heat exchangers performed similarly. An improved heat exchanger was simulated in the laboratory to measure the implications of an improved design. The improved heat transfer did not result in the expected outcome due to limitation in the heat pump system condenser design. Overall, a 5 percent increase in hydronic coil capacity resulted in a similar 5 percent increase in efficiency. While performance was shown to improve, the

efficiency of the air-to-water system combined with hydronic air handler was 37 percent lower than the variable-speed air-to-air heat pump tested for this project.

5. **Future Iteration (Injection Molding):** The next iteration of the MPHX will be manufactured via injection molding, enabling the creation of geometries that are not feasible with three-dimensional printing. Preliminary projections suggest that the injection-molded version will likely surpass both the current MPHX and the commercial coil in terms of effectiveness (0.76 to 0.95) and heat transfer rate (2,580 to 4,230 watts). This translates to a 15 to 20 percent increase in effectiveness and capacity for the injection-molded MPHX.

Overall, the findings indicate that the MPHX not only competes effectively with the commercial coil but also holds significant potential for enhancements in future designs currently in progress, which could position it as a more effective heat transfer solution. Furthermore, a process-based cost analysis showed the potential to reduce the cost of the product relative to commercial coils by 8 to 20 percent.

Follow-on funding related to the development of the MPHX for different applications has been obtained by the team. In an Advanced Research Projects Agency-Energy COOLERCHIPS project, the team will be designing MPHX for higher temperature (122 to 140 degrees Fahrenheit [50 to 65 degrees Celsius]) heat rejection from datacenters to ambient air. In a United States Department of Energy Industrial Efficiency and Decarbonization Office project, the team will be developing waste heat recovery heat exchangers that operate in the 158 to 248 degrees Fahrenheit (70 to 120 degrees Celsius) range. A U.S. patent was also issued for the MPHX product (Narayanan and Rasouli, 2024).

Knowledge Transfer and Next Steps

Market Barriers

User and installer experiences in the field demonstration suggest that the use of mildly flammable refrigerants (designated by the American Society of Heating, Refrigeration, and Air-Conditioning Engineers as A2L) in heat pumps does not pose significant challenges or barriers. Neither users nor installers had concerns about the flammability of the refrigerant, and the adaptations installers made to accommodate the refrigerant were minimal. This suggests that the market for heat pumps that use A2L refrigerants will be largely driven by refrigerant policy and the market for residential heat pumps in general.

The primary barrier to adoption of heat pumps is likely to be their upfront cost. The initial cost of replacing a gas furnace with a residential heat pump is typically higher than installing a like-for-like replacement gas furnace and central air conditioning. Financial incentives are available to California homeowners through utility programs, the statewide Technology and Equipment for Clean Heating Clean California program, and most recently through the Inflation Reduction Act. Much of the state funding is earmarked for low-income customers, providing support worth up to 100 percent of project costs. Financial incentives for higher-income customers help defray the cost of heat pumps, but typically not enough to create parity with gas furnaces.

The cost of operating heat pumps in California poses another barrier to adoption. The high cost of electricity compared to gas in California's investor-owned utility territories means that many customers would face higher operating costs if they moved from furnaces to heat pumps. The possibility of higher utility bills — and the uncertainty around that outcome — dissuades market adoption.

Furthermore, general awareness of heat pumps is low among California residents. This places an additional burden on the contractors who install heat pumps to deliver basic information about the technology to prospective customers. Workforce training may be required to ensure that contractors have the knowledge and communication tools needed to effectively play this role.

Technology Transfer

Rheem is preparing to commercialize the new hybrid inverter drive heat pump with R-454B refrigerant developed for this project. However, a recent change to the certification of variable-speed heat pump equipment will require the control algorithm to be revised before commercialization. The Air-Conditioning, Heating, and Refrigeration Institute has implemented a controls verification procedure to certify variable-speed heat pumps. The controls verification procedure ensures that variable-capacity equipment modulates appropriately when installed and includes a load-based testing procedure. Rheem is working on modifying the controls sequence to the heat pump product tested in this project to ensure it meets this standard. These controls modifications are not expected to impact the performance reported in this project.

Future Research

Future research should continue to innovate around the development of efficient heat pumps at lower cost. Participants in this project were happy with the comfort provided by the heat pump but had mixed opinions about whether the operating costs were lower than their previous gas furnace. This is compounded by the fact that installation costs were higher than a gas furnace replacement due to electrical upgrades needed to support the heat pump indoor unit. All of the sites, with the exception of one that an existing heat pump as the baseline, required a new 240-volt electrical circuit to be installed to support the air handler and 5-kilowatt supplemental electric resistance heater. There should be research to validate the use of heat pumps without supplementary electric resistance heaters in California and encourage heat pump manufacturers to offer heat pump indoor units that operate on a standard 120-volt circuit for these cases. The electric resistance heaters installed for this project were seldom used, suggesting they are not necessary for maintaining comfort. Offering a 120-volt indoor unit solution would eliminate the installation barriers posed by electrical upgrades for the indoor unit, including potential panel upgrades, and reduce connected load on the grid. Manufacturers would need to continue developing strategies for dealing with defrost cycles without supplementary heating systems to reduce the potential for comfort issues related to this approach.

CHAPTER 1: Introduction

Meeting California’s ambitious energy and carbon goals will require low-cost solutions to switch the primary fuel for heating buildings from natural gas to electricity. While heat pumps have been on the market for decades, their initial and ongoing costs have thus far not been competitive with traditional natural gas furnace solutions. Though space conditioning heat pumps are a keystone technology in California’s plans to decarbonize, they are currently installed in only 54 percent of homes throughout the state (Olano, 2022)

The path to electrifying heating in California will require new advances in heat pump technology. The new technologies must achieve high efficiency to avoid increasing operating costs for ratepayers currently using furnaces with relatively low-cost natural gas. There also must be consideration about the refrigerant used in heat pump systems to ensure they meet current and expected future regulations around the global warming potential (GWP) of refrigerants.

Background

In September 2022, the California Air Resources Board (CARB) unanimously approved a proposal to ban the sale of new natural gas-powered water and space heaters by 2030 (CARB, 2022a). As a result, it is anticipated that by 2030, all new space conditioning (and water heating systems) installed in California homes will be heat pumps. CARB’s decision is part of a broader strategy to achieve federal ozone standards over the next 15 years and reduce nitrogen oxide emissions, in line with the 2022 State Strategy for the State Implementation Plan, which established zero-emission regulations for new space and water heaters (CARB, 2022b).

CARB’s 2022 vote to ban gas equipment follows decades of California policymaking aimed at reducing carbon emissions, which arguably began with the passage of Assembly Bill 32 (Nuñez, Chapter 488, Statutes of 2006) (California State Assembly, 2006). Also known as the California Global Warming Solutions Act of 2006, Assembly Bill 32 established the long-term goal of achieving carbon neutrality by 2045. Other policies specifically aimed to increase heat pump adoption, including Senate Bill 1477, which established the Building Initiative for Low-emissions Development (BUILD) Program and the Technology and Equipment for Clean Heating (TECH) Initiatives, which promote heat pump adoption in new and existing homes, respectively (Stern, Chapter 378, Statutes of 2018).

The 2025 Energy Code, adopted by the CEC in September of 2024, strongly promotes heat pump technology in new residential buildings (CEC, 2024). It expands previous standards by using heat pumps as the baseline technology for both space conditioning and water heating for single-family homes in all climate zones. While the 2025 code does not explicitly mandate all-electric construction for new single-family homes, it creates strong market signals for developers to build all-electric homes by eliminating subsidies for gas lines to new homes and providing electrification-ready standards.

To support decarbonization of the building sector, California set a goal of installing 6 million heat pumps (for space conditioning and water heating) by 2030 (CEC, 2022a). However, the California Heat Pump Partnership (CAHPP) recently estimated the state will fall short of its target by 4 million units if the present pace of adoption continues (CAHPP, 2025). CAHPP identified high installation costs as one of the primary barriers to heat pump adoption. Estimates suggest that installing a space conditioning heat pump in California can be nearly twice as expensive as installing a furnace or air conditioner.[1]

High operating costs are also stifling the market uptake of space-conditioning heat pumps. With electricity more expensive than gas in most of California (Davis, 2023), switching to a heat pump would increase most customers' utility bills. Improving system efficiency above the federal minimum requirement, along with insulating and eliminating duct leakage, is critical to addressing affordability of heat pump operation. Advancing heat pump adoption in California through a market-based approach requires the development of technology that is cost-competitive with gas furnaces in both upfront capital and operating costs.

In parallel with efforts to promote heat pumps, increasingly stringent refrigerant policy aims minimize direct emissions from this vital decarbonization technology. To that end, CARB mandated that all new stationary air conditioning equipment must use a refrigerant with a GWP of less than 750 starting in 2025 (CARB, 2019). This requirement will be usurped by United States Environmental Protection Agency (U.S. EPA) regulation restricting space conditioning heat pumps installed in residential (and light commercial) building after January 1, 2026, to use refrigerants with a GWP no higher than 700 (U.S. EPA, 2024). Compliance with these policy changes requires modifications to equipment design which had before typically used R-410A, with a GWP of 2088.

Continuing to develop heat pump systems whose upfront and operating costs can compete with those of natural gas-fired alternatives will ease the transition to electrified buildings. Supporting California's clean energy economy by offering more affordable heating and cooling solutions to homeowners will lessen the economic burden consumers face and promote more rapid uptake. Demonstrating the technical and economic feasibility of heat pump design innovations is critical to clearing a pathway for building decarbonization.

Project Overview

This project aimed to develop and demonstrate next-generation heat pump technology that achieves high-efficiency heating and cooling, significantly reducing greenhouse gas (GHG) emissions from buildings at a lower cost relative to similarly performing equipment. The project team pursued this goal amid a changing global landscape around refrigerant

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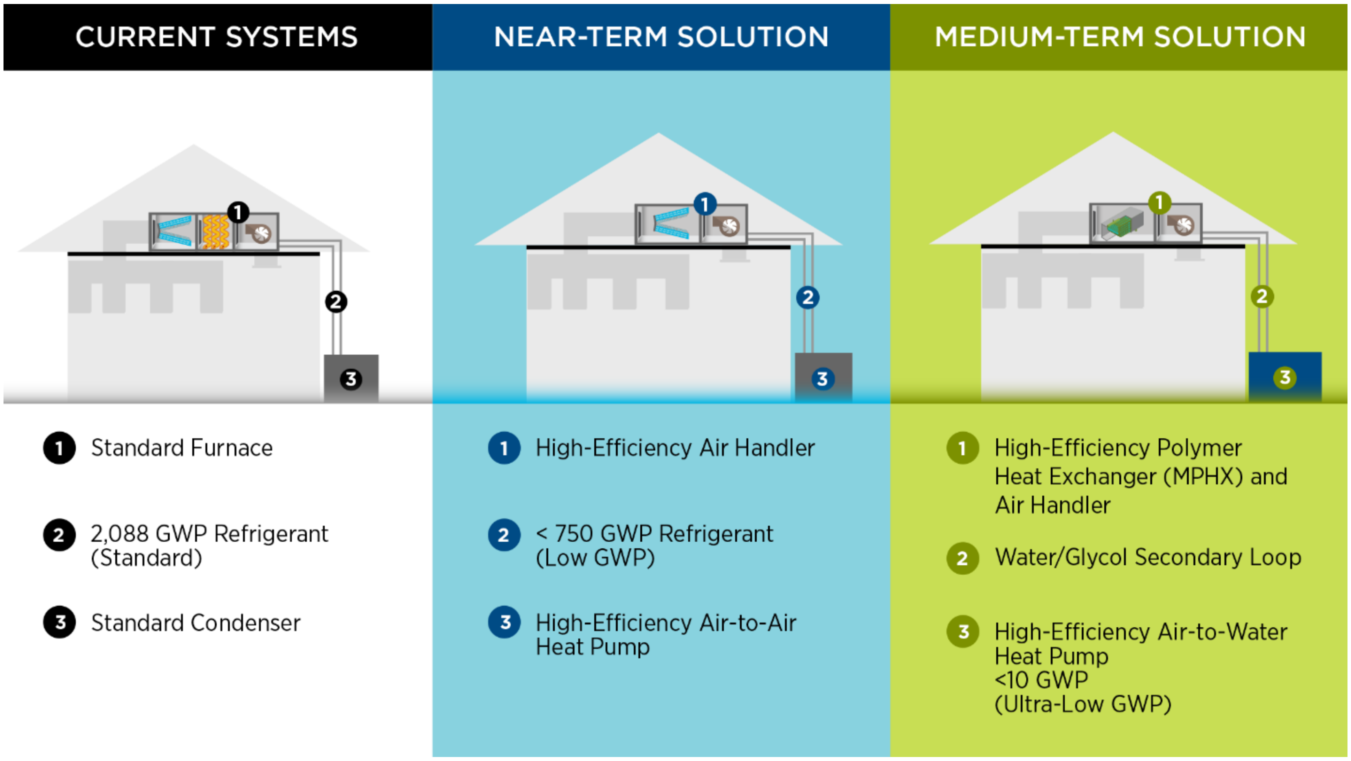
[1] The median total cost to install a natural gas furnace and 14 SEER Central AC in California is roughly $4,000 (Opinion Dynamics, 2022) and $5,000 to $6,000 (Bender, 2024). Estimates for installing a residential heat pump in California range from $10,000 (Opinion Dynamics, 2022) to $20,000 (TECH Clean California, 2024).

acceptability, meaning that, in addition to cost and efficiency, the team also needed to consider future changes in refrigerant regulations.

The team pursued two pathways, advancing both near- and medium-term solutions (Figure 1). The near-term solution focused on an air-source ducted heat pump developed by Rheem Manufacturing Company (Rheem) that can more easily integrate into existing homes due to its similarity to current air conditioning and heat pump technology. The medium-term solution advances heat exchanger technology for use with air-to-water (ATW) heat pumps that can safely integrate natural refrigerants (such as, R-290) while also providing other potential benefits such as reduced refrigerant usage and the ability to use the secondary water loop for low-cost thermal storage.

Figure 1: Overview of Near- and Medium-Term Solutions



Source: UC Davis

Demonstrations for the project focused on the near-term solution that featured an innovative, cost-effective compressor drive that offers the advantages of a variable capacity system using lower-cost components, thereby reducing installation costs relative to competing technologies. Additionally, upcoming CARB regulations will mandate that refrigerants used in all new stationary residential air conditioning systems have a 100-year GWP value of 750 or less starting in 2025. This system was designed to accommodate a low-GWP refrigerant (R-454B) to comply with the new regulations.

The heat pump was installed in 10 homes in California to evaluate its performance. Surveys of participants and contractors were conducted to identify market barriers to the adoption of the heat pump technology, including considerations related to the mildly flammable refrigerant.

Development of advanced water-to-air heat exchangers for improving the performance of ATW heat pumps was the focus of the medium-term approach. Innovative microchannel polymer heat exchangers (MPHX) were designed and tested at the University of California, Davis (UC Davis) to reduce the penalty introduced by a secondary fluid loop for thermal distribution. Laboratory testing was performed to measure the efficiency improvement of the ATW heat pump when coupled with the MPHX compared to a commercial coil. In addition, different MPHX manufacturing methods were explored to determine the most cost-effective approach for commercialization.

CHAPTER 2:
Project Approach

The primary goal of this project was to develop and demonstrate heat pump technologies that achieve higher efficiency at a lower cost relative to other high-efficiency systems, while also considering future changes in refrigerant regulations. Different approaches were pursued for the advancement of the near- and medium-term solutions for this project. Development of the near-term solution was completed by Rheem, and performance verification was conducted by UC Davis. After the development, field testing was conducted to measure the installed performance of the new heat pump technology. For the medium-term solution, advancements in heat exchanger design were conducted through modeling and small-scale testing before testing a full-scale heat exchanger coupled to an ATW heat pump in the laboratory. This section describes the project approach for both the near- and medium-term solutions.

Near-Term Solution

The development phase of the near-term solution involved rigorous laboratory testing of the equipment to verify the heating and cooling capacities and power consumption under various operating conditions. Following the laboratory tests, 10 field sites in California were selected for demonstrating the real-world performance of the new heat pump technology. This thorough approach ensured that the new heat pump technology was tested not only in controlled laboratory conditions but also in diverse real-world scenarios, providing a holistic understanding of its performance, user satisfaction, and potential market barriers related to adoption.

Heat Pump Development

The next-generation heat pump developed for this project was designed as a standard split-system air-source heat pump. The indoor unit consisted of a variable-speed blower paired with a heat pump refrigerant coil. Depending on the application, the indoor unit would also contain an electric resistance backup heater for maintaining comfort during defrost and providing additional capacity at low ambient temperature conditions. The outdoor unit consisted of the variable-speed compressor, refrigerant coil, and fan.

Heat pumps resemble standard air conditioning units but have a refrigerant reversing valve that allows the refrigerant to flow in either direction. This effectively allows the evaporator and condenser to switch roles depending on whether the building needs heating or cooling. Space heating requirements are therefore satisfied using electricity without the need for fossil fuels. A major advantage of providing heat using a refrigerant cycle is that the coefficient of performance (COP) can be greater than 1, meaning more heat is produced per unit of energy consumed by the system. This allows heat pumps to operate much more efficiently than gas furnaces or electric resistance heaters, as those systems have a maximum possible COP of 1.

Standard-efficiency heat pumps on the market today use fixed-speed compressors designed to operate on single-phase power, typically driven by permanent split capacitor (PSC) motors

(Goetzler et al., 2013). The novel compressor drive used in the heat pump developed for this project was a low-power inverter that operated the PSC motor at reduced speeds, similar to a variable-speed compressor driven by a brushless permanent magnet motor. Reduction in energy usage was primarily expected to result from the variable speed operation due to the ability of the system to modulate capacity to match the cooling/heating demand without the need for cycling. Reducing the speed of the compressor and associated refrigerant flow also allowed the heat exchanger coils to become more effective, leading to higher efficiency.

The compressor drive is disabled during full-load operation; specifically, the compressor motor runs on supply power without intervention from the inverter. Thus, the full-load efficiency of the system remains unchanged. At lower operating speeds, the efficiency of the combination of the novel compressor drive, PSC motor, and compressor is lower than that of the combination of an electronic inverter drive, brushless permanent-magnet motor, and compressor. This results in the proposed technology having a lower efficiency than a conventional variable-speed system, but higher efficiency than a fixed-speed system. The cost of the PSC motor is less than that of a brushless permanent magnet motor. Because the novel compressor drive was only engaged during part-load operation, the components in the inverter drive were sized only for low-power operation. This strategy helped to keep the cost of the proposed technology lower than that with a conventional variable-speed heat pump system.

Heat Pump Laboratory Testing

Laboratory testing was performed to evaluate the performance of the near-term heat pump. This data was used to develop performance maps of the system for simulation modeling of performance in different climate zones. The performance of the novel heat pump was tested by Rheem in its facility across a variety of cooling and heating conditions to encompass the full range of performance. Identical test points were taken for cooling operation at the UC Davis environmental control chamber, which was used to validate data that was collected by Rheem. Performance mapping of the unit was guided by a process outlined by the Florida Solar Energy Center (Raustad, 2012). The unit was tested at full speed at three indoor wet-bulb temperatures and six outdoor dry-bulb temperatures, assessing the performance at each combination of these conditions while keeping the indoor dry bulb temperature at a constant 80 degrees Fahrenheit (°F) (27 degrees Celsius [°C]). The full speed cooling test matrix is provided in Appendix A. Additional part-load test points were taken with the compressor running at reduced speeds. Due to the design of the variable-speed drive, part-load conditions above about 70 percent compressor speed were not possible.

A wide range of heating conditions were also taken by Rheem at its facility. Six different full speed outdoor dry-bulb, wet-bulb pairs were taken across three different indoor dry-bub temperatures. Only two of these points were matched at the Western Cooling Efficiency Center (WCEC) chamber, due to the limits of the chamber’s capabilities. Part-load tests were also taken by Rheem at various compressor speeds that were not repeated at WCEC. The part-load test conditions can be found in Table A-4 and Table A-7 in Appendix A.

Heat Pump Field Testing

This section outlines the approach used for the assessment of the near-term heat pump retrofit implementations for the selected residential properties. The assessment included impacts on heating, ventilation, and air conditioning (HVAC) energy use, utility cost, and GHG emission. The field demonstrations aimed to provide a comprehensive technical evaluation of retrofitting existing HVAC systems with the next-generation heat pump in real-world scenarios. Ten detached single-family homes, constructed between 1976 and 2006, were selected for this study. The conditioned areas of these homes ranged from just more than 1,000 to 2,600 square feet. These homes were situated in two California climate zones (CZs), specifically CZ2 (Santa Rosa) and CZ12 (Sacramento).[2]

Among the selected homes, eight used residential split air conditioning (AC) units for cooling, one home in CZ2 lacked cooling, and one in CZ12 already employed a heat pump. All but one of these homes used a natural gas furnace for heating. The cooling capacities of the existing AC units varied from 2 to 5 cooling tons (24 to 60 thousand British thermal units per year [kBtu/yr]), and the outputs of the gas furnaces ranged from 36 to 96 kBtu/hr. The retrofit heat pumps selected for this study had cooling capacities similar to the existing AC units. The electric panel capacity at the sites ranged from 100 to 200 amperes (amps) and all supported the additional 240-volt (V) circuit to power the air handler and 5-kilowatt (kW) supplemental heater. However, one site required a subpanel to be installed to allow room for the new circuit which added to the installation cost, and another site with a 100-amp panel has limited remaining capacity to support further electrification, such as a heat pump water heater, or electric stove.

To determine the appropriate capacities for the retrofit heat pumps, for four of the homes the research team conducted American National Standards Institute/Air Conditioning Contractors of America Manual J load calculations which is the standard recognized in the U.S. for equipment sizing for many residential building types. All heat pumps also included a 5-kW supplemental heater kit. Table 1 provides a summary of the test site information and retrofit systems.

Table 1: Summary of Demonstration Sites and Retrofit Heat Pumps

Site	Climate Zone	Year Built	Conditioned area (ft^2)	Existing System	Retrofit Heat Pump
1	2	1993	1,100	Single zone split AC w/natural gas furnace AC: 2-ton, 10 SEER Furnace: output 58,000 Btu/hr, AFUE 80%	3-ton with 5 kW supplementary heater kit (16 SEER2, 8.1 HSPF2)

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[2] The city selected in CZ2 has a 0.5 percent dry bulb cooling design temperature at 96°F (35.6°C), and cooling degree days of 456, the Winter Median of Extreme at 24°F (-4.4°C) and heating degree days of 2,980. The city selected in CZ12 has a 0.5 percent dry bulb cooling design temperature at 100°F (37.8°C), and cooling degree days of 1,470, the Winter Median of Extreme at 21°F (-6.1°C) and a heating degree days of 2,653.

Site	Climate Zone	Year Built	Conditioned area (ft^2)	Existing System	Retrofit Heat Pump
2	2	1976	2,007	Single zone two-stage natural gas, no cooling Furnace: output 97,000 Btu/hr, AFUE 90%	2-ton with 5 kW supplementary heater kit (16 SEER2, 8.1 HSPF2)
3	02	1996	1,400	Single zone split AC w/natural gas furnace AC: 3-ton, 10 SEER Furnace: output 56,000 Btu/hr, AFUE 80%	3-ton with 5 kW supplementary heater kit (16 SEER2, 8.1 HSPF2)
4	12	2005	2,652	Two-zone system AC w/natural gas furnace AC: 4-ton, 10 SEER Furnace: output 68,000 Btu/hr, AFUE 80%	4-ton with 5 kW supplementary heater kit (16 SEER2, 8.1 HSPF2)
5	12	1985	1,036	Single zone split 2-ton heat pump with 5 kW supplementary heater kit, efficiency unknown	2-ton with 5 kW supplementary heater kit (16 SEER2, 8.1 HSPF2)
6	12	2004	1,588	Single zone split AC w/natural gas furnace AC: 3-ton, efficiency unknown Furnace: output 40,000 Btu/hr, AFUE 80%	3-ton with 5 kW supplementary heater kit (16 SEER2, 8.1 HSPF2)
7	12	2003	1,801	Single zone split AC w/natural gas furnace AC: 4-ton SEER 13 Furnace: output 71,000 Btu/hr	4-ton with 5 kW supplementary heater kit (16 SEER2, 8.1 HSPF2)
8	12	2006	2,121	Two zone split AC w/natural gas furnace AC 3-ton, SEER 13 Furnace: output 68,000 Btu/hr, AFUE 80%	3-ton with 5 kW supplementary heater kit (16 SEER2, 8.1 HSPF2)
9	12	2003	1,857	Single zone split AC w/natural gas furnace AC: 3.5-ton, SEER 13 Furnace: output 71,000 Btu/hr, AFUE 80%	4-ton with 5 kW supplementary heater kit (16 SEER2, 8.1 HSPF2)
10	12	1990	2,336	Single zone split AC w/natural gas furnace AC: 5-ton, 10 SEER Furnace: output 80,000 Btu/hr	5-ton with 5 kW supplementary heater kit (16 SEER2, 8.1 HSPF2)

AFUE=Annual Fuel Utilization Efficiency; Btu/hr=British thermal units per hour; ft^2=square feet; HSPF2=Heating Seasonal Performance Factor 2; SEER2=Seasonal Energy Efficiency Ratio 2; w/=with
Source: UC Davis

Field Data Collection

The research team implemented a comprehensive monitoring system to gather data continuously over a period exceeding two years, from August 2021 to May 2024. One year into the data collection process, the installation of the next-generation heat pump marked the division of data into pre- and post-retrofit monitoring periods. Throughout the entire metering period, the following real-time data were collected at one-minute intervals, meticulously stored, verified, and analyzed at each of the 10 sites. The data collected included:

- HVAC natural gas consumption (limited to the baseline period)
- HVAC electricity usage (fan, compressor, and auxiliary use)
- HVAC air flow differential pressure
- Zone temperature and relative humidity
- HVAC supply and return temperature and relative humidity
- Outdoor air temperature and relative humidity

The research team also documented the HVAC air flow rate as a function of power and differential pressure, which were used to determine cooling and heating capacities. Subsequently, the measured data were used to calculate annual energy consumption and to characterize both equipment efficiency and thermal comfort conditions.

HVAC Energy Use

The research team used International Performance Measurement and Verification Protocol Option B (retrofit isolation with all parameter measurement) to quantify normalized annual HVAC energy use for the baseline and retrofit systems. Annual energy use was calculated using daily energy consumption data before and after the retrofit. The hourly energy use data were then input into a change-point regression model using the time-of-week-and-temperature regression method for each site. This model was selected to account for variations in energy use as heating and cooling systems were engaged. The developed models were subsequently applied to typical meteorological year weather files, normalizing the measured energy use to consistent weather conditions. The normalized energy use was then computed for the entire year, separately for the heating and cooling operation periods.

The comparative analysis also included an evaluation of HVAC equipment efficiency. For both the existing AC units and the next-generation heat pumps, the COP was characterized using measured cooling load, heating load, and electrical input data. The intent of the analysis was to validate the field performance of the next-generation heat pumps, acknowledging that field installation conditions differ significantly from laboratory conditions.

Utility Cost

The utility cost analysis sought to quantify the financial implications of the retrofit across the residential properties. Following the guidelines outlined in California Public Utilities Commission (CPUC) policy (CPUC D.15-07-001), the three major investor-owned utilities in California transitioned their customers to a default time-of-use (TOU) schedule in 2019. For this study, the research team scrutinized the energy costs associated with the post-retrofit system, as well as the calculated baseline for all homes, using the Pacific Gas and Electric Company (PG&E) TOU-C rate (PG&E, 2024a), as outlined in Table 2. Note that a baseline credit of $0.1073 was applied to baseline use.

Under the TOU rate plans, the utility imposes a fixed price per kilowatt-hour (kWh) based on both the time of day and the time of year. Notably, late afternoon and evening periods are subjected to higher rates compared to other times of the day, with summer season rates surpassing those of the winter season. To compute the baseline gas energy cost, the baseline allocation for each home was determined and the Tier 1 rate ($2.15 per therm) for usage was applied.

Subsequently, the research team applied the applicable rate to the normalized energy consumption for both pre- and post-retrofit systems to calculate costs, which were then summarized annually, including a breakdown for heating and cooling expenses.

Table 2: Electricity Rate for PG&E TOU-C

Electricity ($/kWh)	Peak (4:00 p.m. to 9:00 p.m.)	Off Peak
Summer (June – Sept.)	0.63	0.54
Winter (Oct. – May)	0.52	0.49

$/kwh=dollars per kilowatt-hour
Note: Baseline credit of $0.1073 was applied.
Source: UC Davis

Environmental Impact

The environmental impact assessment focused on quantifying the reduction in GHG emissions by leveraging the hourly GHG emission factors published by California Energy Commission (CEC) (CEC, 2022b). These factors estimated the environmental benefits of transitioning to electric heat pump systems for residential space heating by converting predicted site energy use to long-run marginal GHG emissions. The hourly factors varied by location, time of day, and season. The multipliers were applied to the hourly gas and electricity use in both the pre-retrofit and post-retrofit scenarios to determine the differences in GHG emissions. For this analysis, the average grid GHG electricity emission factor was 0.1988 pounds (lb) carbon dioxide equivalent per kilowatt-hour (CO_2e/kWh), and the average natural gas GHG emission factor was 13.29 lb CO_2e/therms.

Occupant Survey

Field testing included both baseline monitoring of the existing equipment and post-retrofit monitoring after the installation of the new heat pumps. To gain comprehensive insights, participant surveys were conducted to gather feedback on participants' experience with the installation and operation of the new equipment. Additionally, installation contractors were interviewed to understand any changes in installation procedures, particularly regarding the handling of the new refrigerant, R-454B, which, like many low-GWP refrigerants, is classified as an A2L (mildly flammable) fluid.

As part of the field testing, participants were surveyed to gather data on their experience with the installation and operation of the next generation heat pumps. Surveys of the participants were conducted in the winter and summer during both baseline and retrofit periods. Quantitative and qualitative analysis of the survey responses were conducted to identify general trends and common themes to characterize user experience.

Heat Pump Performance Modeling

Performance modeling of the near-term heat pump was conducted to further quantify the ratepayer benefits associated with the low-GWP heat pump developed. The performance maps developed through lab testing were incorporated into the building energy modeling software. The primary focus was the annual impact of the new heat pump on ratepayer cost and GHG emissions.

Model Summary

Performance modeling of the near-term heat pump solution was conducted in EnergyPlus. A United States Department of Energy (U.S. DOE) prototype single-family residential home meeting the 2006 International Energy Conservation Code was used for the modeling. The prototype model was two-stories with approximately 2,400 square feet of conditioned floor area. Table 3 provides a summary of some of the key performance characteristics of the home.

Table 3: Key Performance Characteristics for Prototype Single-Family Home Model

Fenestration U-Factor	Glazed Fenestration SHGC	Ceiling R-value	Wall R-value
0.65	0.40	30	13

Fenestration U-factor=a measure of how efficiently a window or door assembly (including the glass, frame, and spacers) transfers heat. A lower U-factor indicates better energy efficiency, meaning the window or door is better at preventing heat loss; SHGC=Solar Heat Gain Coefficient
Source: UC Davis

The prototype model was simulated in each of the 16 California climate zones (CZ). Three heating and cooling systems were considered: 1) Baseline heat pump meeting minimum U.S. DOE efficiency standards, 2) Baseline air conditioner and natural gas furnace, and 3) low-GWP heat pump system. Table 4 shows the heating and cooling equipment performance specifications for each system modeled. The EnergyPlus autosize feature was used to size the system for each simulation.

Table 4: Heating and Cooling Equipment Specifications for Systems Modeled

	SEER2	HSPF2	AFUE
Baseline 1: Heat Pump	13.3	7.0	N/A
Baseline 2: AC with Furnace	13.3	N/A	0.80
Low-GWP Heat Pump	16.0	8.1	N/A

Utility costs were estimated based on the PG&E E-1 residential electricity rate and residential gas rate in effect on January 1, 2024. The utility costs used in this analysis are shown in Table 5.

Table 5: Utility Cost Data Used for the Analysis

Electricity Rate	Gas Rate
\$0.42	\$2.44
\$/kWh	\$/therm

\$/therm=dollars per therm
Source: PG&E, 2024a; 2024b

The GHG emissions data used in this analysis came from the CARB Emissions Factor Database and are summarized in Table 6. CARB collects emissions data from the various sources of electricity generation in California and provides a weighted average of the emissions related to on-site electricity and natural gas use. Emissions related to electricity use vary throughout the day and seasonally, but the savings calculated for some of the systems evaluated in this project did not have the resolution necessary to account for hourly emissions estimates.

Table 6: Emissions Factors Used in the Analysis of Greenhouse Gas Emissions

Emissions Source	Factor	Units
California Average Grid GHG Electricity Emissions Factors (2020)	0.4644	lb CO_2e/kWh
Natural Gas GHG Emission Factor	11.70	lb CO_2e/therm

Source: CARB, 2024

Heat Pump Market Analysis

A market outlook for the next-generation heat pump technology in California's single-family home retrofit market was conducted. Implications of the transition from high-GWP hydrofluorocarbon refrigerants such as R-410A to lower-GWP refrigerants like R-454B and R-32 were discussed. The underlying market conditions for residential heat pumps in California were also described, including upfront and operating costs, incentive programs, and workforce implications. The research drew upon data from relevant market studies and government sources and findings from the field study. Interviews conducted with eight stakeholders — including two HVAC market experts, a state HVAC subsidy program administrator, two residential HVAC contractors, and residential HVAC program staff members from three California utilities — also informed the market outlook.

Medium-Term Solution

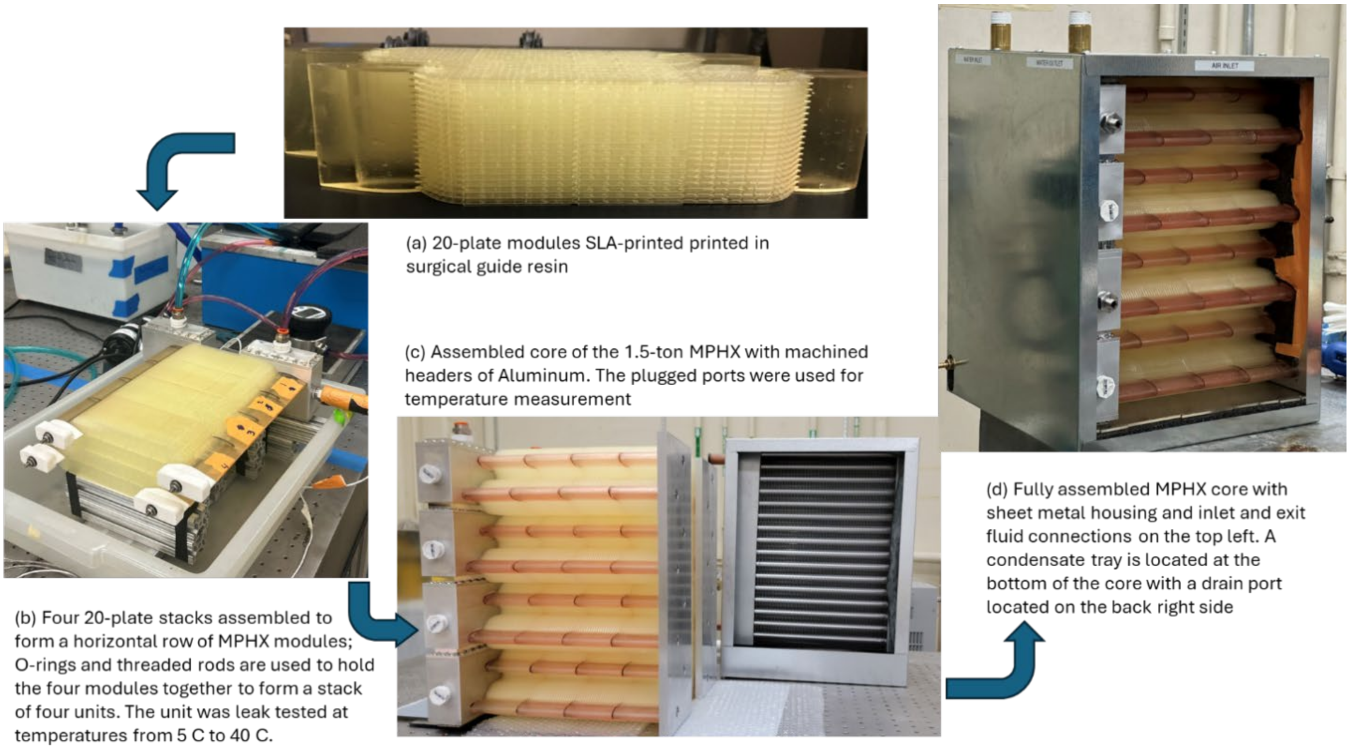
Improving the performance of ATW heat pumps would be an important step if future regulations require ultra-low-GWP refrigerants of 10 or less. Practical ultra-low-GWP refrigerants include hydrocarbons, which are highly flammable, and therefore cannot be safely used in many direct-expansion heat pumps where refrigerants enter the building. To avoid this risk, one solution would be to isolate the refrigerant in a heat pump outside of the home and use a secondary water loop to distribute the heating and cooling in the building. The secondary loop introduces significant efficiency penalties that must be overcome to avoid sacrificing performance to facilitate the use of ultra-low-GWP refrigerants. This project developed the novel MPHX to improve the performance of the secondary loop and reduce the efficiency penalties of ATW heat pumps. The development of the MPHX started with small-scale prototypes that were tested for thermal effectiveness and mechanical strength before developing a full-scale MPHX that was tested in combination with an ATW heat pump. Two different manufacturing techniques were explored for building the MPHX and a process-based cost model was used to evaluate the lowest cost path for commercialization.

Heat Exchanger Development

The MPHX prototypes were tested for thermal performance and mechanical strength. Prototypes were fabricated using three-dimensional (3D) printing and injection molding approaches, though multiple challenges with the injection molding technique resulted in only the 3D printed versions being tested. New injection molding strategies were developed, however, creating a feasible pathway for future production of the MPHX. Process-based cost models were developed to evaluate the two fabrication methods. Process-based cost models break down contributing costs into various categories (for example, labor, equipment, material, and so forth); process steps can then be broken down into associated costs in these categories, with these costs being tied to process and design parameters such as cycle time, part size, and material (Kirchain and Field, 2000).

Due to the challenges with the injection-molded (IM) prototypes, the full-scale heat exchanger was fabricated using stereolithography (SLA) 3D printing. The 3D printing process had other practical limitations that required the full-scale MPHX tested for this project to have sub-optimal channel heights. The MPHX was designed to be assembled using individual water plates for better control of channel heights, providing superior performance. The process of combining individual plates lent itself to fabrication using injection molding in future production. A 1.5-ton unit sized equivalent to a commercial water coil was assembled consisting of several 20-plate core modules for a total of 320 water plates (Figure 2). Figure 3 shows the manufacturing process for the MPHX including material and energy inputs used for the process-based cost model.

Figure 2: Assembly of the Prototype 1.5-ton MPHX



(a) 20-plate modules SLA-printed printed in surgical guide resin

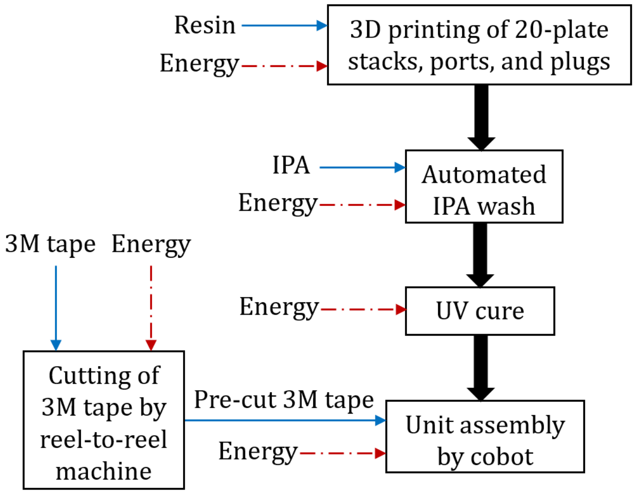
(b) Four 20-plate stacks assembled to form a horizontal row of MPHX modules; O-rings and threaded rods are used to hold the four modules together to form a stack of four units. The unit was leak tested at temperatures from 5 C to 40 C.

(c) Assembled core of the 1.5-ton MPHX with machined headers of Aluminum. The plugged ports were used for temperature measurement

(d) Fully assembled MPHX core with sheet metal housing and inlet and exit fluid connections on the top left. A condensate tray is located at the bottom of the core with a drain port located on the back right side

Source: UC Davis

Figure 3: Manufacturing Steps for the SLA Printed MPHX Unit



Source: UC Davis

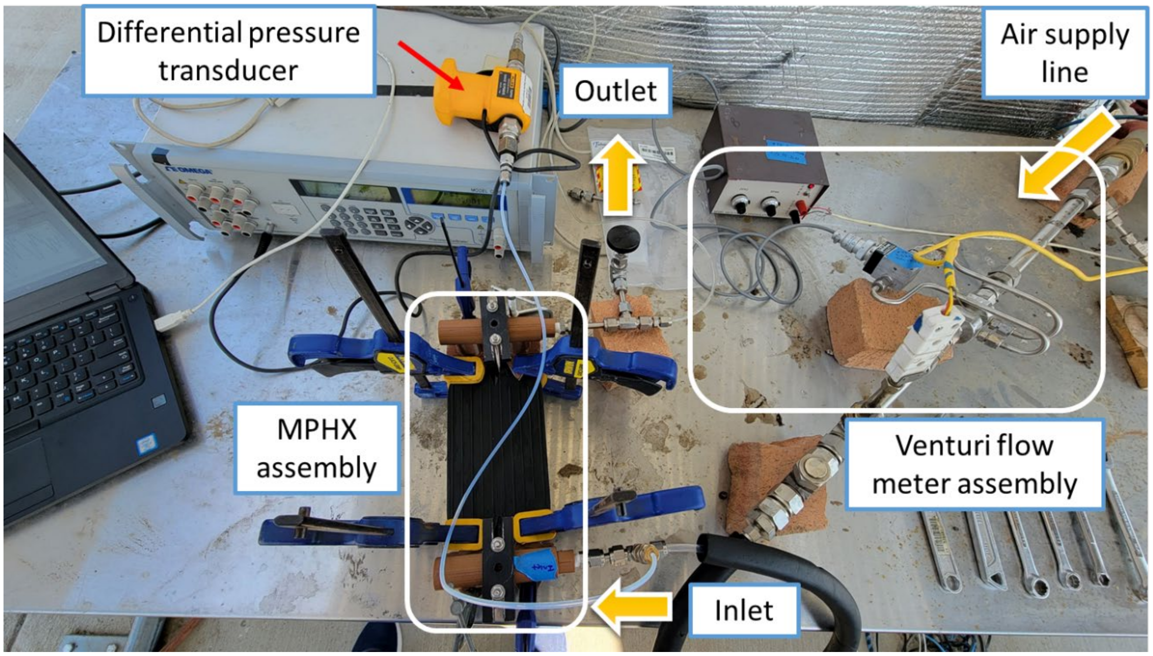
Heat Exchanger Testing

The MPHX was evaluated in several ways including small-scale and full-scale testing. Small-scale tests were conducted including mechanical testing for durability and thermal testing for heat transfer performance were conducted at UC Davis laboratories. Full-scale testing included isolated testing of the MPHX, and testing combined with an air-to-water heat pump. The following sections describe the testing process used to evaluate the MPHX.

MPHX Mechanical Testing Approach

Assessment of structural integrity was performed by pressurizing the MPHX in static and cyclic tests. Additional pressure drop tests were performed on the water side of the MPHX. The MPHX assembly was connected to a pressure drop testing loop shown in Figure 4. The test rig could supply a known mass flow rate of compressed air to the MPHX assembly with fine control of flow rate. A precise differential pressure transducer shown in Figure 4 was used to measure pressure drop of flow across the MPHX for different mass flow rates.

Figure 4: MPHX Pressure Drop and Static/Cyclic Pressure Testing Setup



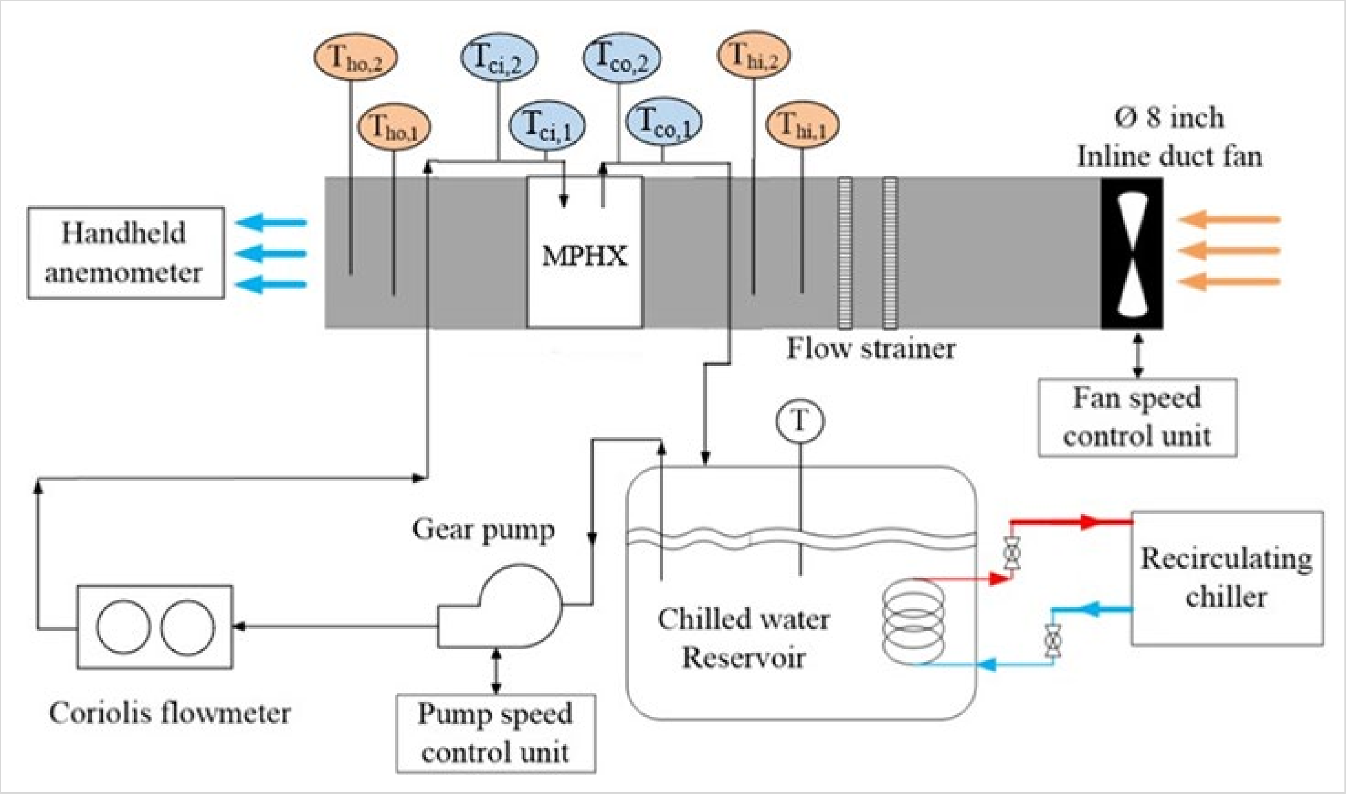
Source: UC Davis

MPHX Thermal Testing Approach

To evaluate the MPHX thermal performance, room-temperature air was cooled by the MPHX under a range of conditions. Figure 5 shows the schematic of the test loop. Chilled water was pumped through a Coriolis flowmeter to measure mass flow rate before entering the four inlet tubes for the MPHX. On the air side, a blower fan forced air through a duct followed by a flow straightener before reaching the MPHX. After going through the MPHX, the air was funneled through a smaller section of duct where a handheld anemometer measured the air flow rate. Two resistance temperature detectors measured temperatures for each stream at the inlet and outlet. Additionally, two sensors measured temperature and relative humidity of the ambient air and exhaust air, respectively.

Air flow rate varied from about 55 to 165 cubic feet per minute (CFM), and water flow rate from about 1.1 to 5.3 pounds mass per minute (lbm/min). The ratio of heat capacity rates, *Cr,* of approximately 0.25, 0.5, 0.75, and 1 were the test points. The procedure for setting the flowrates for each test condition was to first fix the air flowrate (by choosing the fan setting, which ranged from 1 to 10; actual settings used ranged from 3 to 8). Then, the water flowrate was adjusted to approximately achieve the desired *Cr* value, with the minimum heat capacity rate being on the air side and the maximum heat capacity rate on the water side. Two chiller setpoints were used to cool the water reservoir: 41°F (5°C) and 50°F (10°C). At each test condition considered, one minute of steady state data was collected to obtain an average.

Figure 5: Schematic of the Test Loop Used for Pilot-Scale MPHX Testing



Source: UC Davis

Notes:

T_{hi} – Temperature of air entering MPHX (subscripts 1 and 2 refer to two independent measurements of the fluid)
T_{ho} – Temperature of air exiting MPHX (subscripts 1 and 2 refer to two independent measurements of the fluid)
T_{ci} – Temperature of water entering MPHX (subscripts 1 and 2 refer to two independent measurements of the fluid)
T_{co} – Temperature of water exiting MPHX (subscripts 1 and 2 refer to two independent measurements of the fluid)

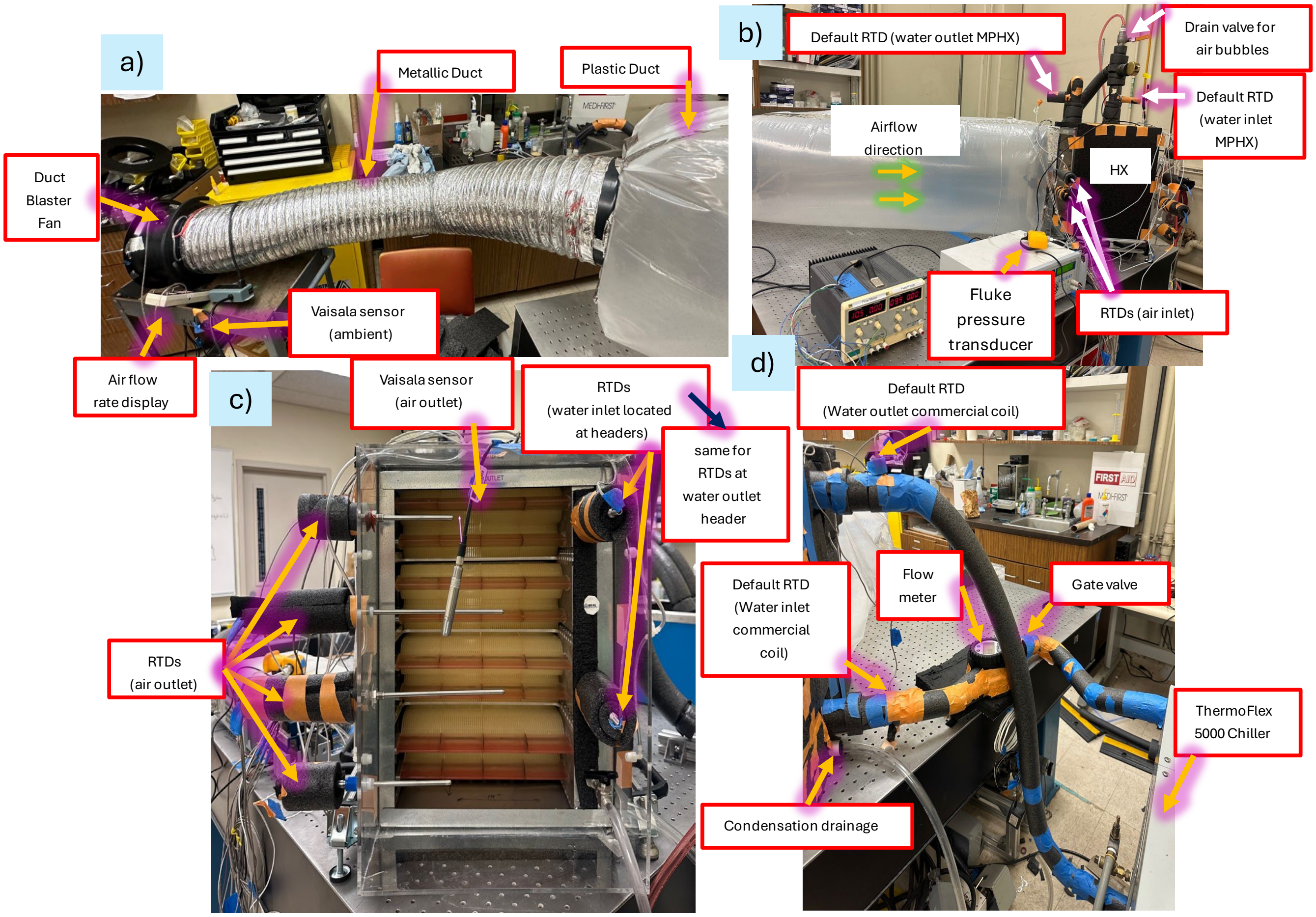
Full-Scale System Testing

The goal of this task was to set up and evaluate the performance of the MPHX combined with an ATW heat pump in the laboratory to evaluate the new coil design. The performance of the MPHX was compared to the performance of a commercially available coil that uses a standard fin-tube heat exchanger design. Isolated thermal testing of the full-scale heat exchangers was performed prior to testing in combination with an ATW heat pump. Improved hydronic heat exchanger design can improve heat pump efficiency by reducing the temperature lift required in the refrigeration cycle. Since the hydronic coil and heat pump work as a system, it was important to characterize the overall impact of the improved heat exchanger in terms of total system benefit. The ATW heat pump used for testing is commercially available and uses a low-GWP refrigerant (R-32). The heat pump is inverter-driven and capable of variable capacity, so the unit was tested at full and part-load conditions.

Isolated Testing

For isolated testing, a testing loop similar to the small-scale thermal testing was constructed and used to evaluate both heat exchangers (HX), using air and water as the working fluids. Photos of the testing loop are shown in Figure 6. Key outputs from the air side included pressure drop, inlet and outlet temperatures, humidity, and airflow. Main outputs from the water side included inlet and outlet temperatures as well as flow rate.

Figure 6: Heat Exchanger Testing Loop Configuration



(a) Duct blaster fan location and the start of the air duct; (b) Air duct extension and connection to the corresponding HX, as well as the acrylic shrouds and some sensor placements; (c) Front view of the HX, highlighting some sensor locations; (d) Chiller connection to the HX and intermediate components.

Source: UC Davis

Air-to-Water Heat Pump Testing

Testing of the performance of the MPHX and standard hydronic coil coupled to an ATW heat pump in the laboratory was conducted following the Air-Conditioning, Heating, and Refrigeration Institute (AHRI) Standard 210/240 test protocol that is used for air-to-air heat pumps. While another AHRI standard exists (550/590) for testing ATW heat pumps, it does not allow for a direct comparison with traditional heat pumps often used today. Table 7 and

Table 8 show the test conditions that were used to test the ATW heat pump with both hydronic coils.

Table 7: Different Cooling Capacity Ratio Performance Evaluation Test Conditions

Test ID	Compressor Speed	Outdoor Dry-Bulb (°F)	Indoor Dry-bulb/Wet-bulb (°F)
CCR2	58 Hz	75	80/63
CCR3	58 Hz	85	80/63
CCR4	58 Hz	95	80/63
CCR5	58 Hz	105	80/63
CCR8	58 Hz	75	80/67
CCR9	58 Hz	85	80/67
CCR10	58 Hz	95	80/67
CCR11	58 Hz	105	80/67

Source: UC Davis

Table 8: Part-Load Cooling Performance Evaluation Test Conditions

Test ID	Compressor Speed	Outdoor Dry-Bulb (°F)	Indoor Dry-bulb/Wet-bulb (°F)
PL1	58 Hz	95	80/67
PL2	52 Hz	95	80/67
PL3	46 Hz	95	80/67
PL4	42 Hz	95	80/67
PL5	36 Hz	95	80/67
PL6	30 Hz	95	80/67

Source: UC Davis

CHAPTER 3:
Results

Near-Term Solution

This section describes the results from laboratory and field testing of the near-term solution, as well as performance modeling. Laboratory testing was conducted in both UC Davis and Rheem facilities, and the field testing was performed in 10 single-family homes in California.

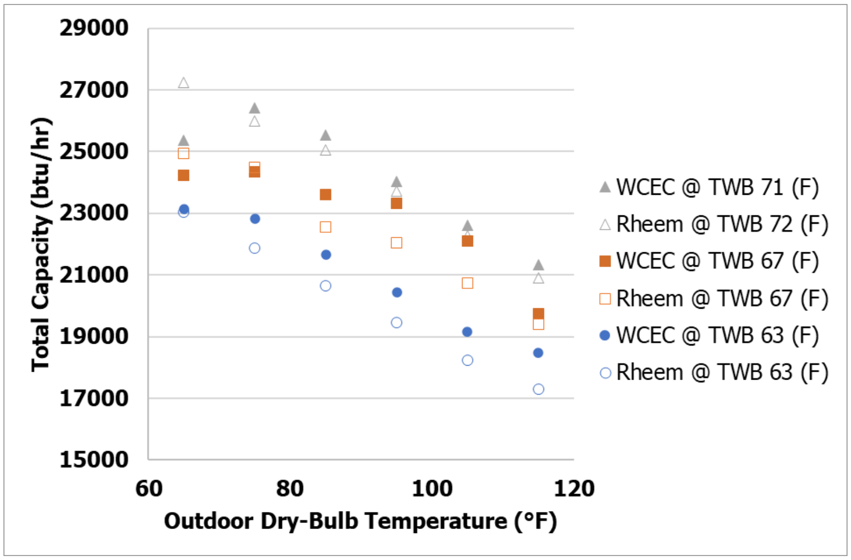
Heat Pump Laboratory Testing

Laboratory testing was conducted in both heating and cooling conditions to develop performance maps for energy modeling.

Full Speed Cooling Results

Figure 7 shows the total capacity of the WCEC and Rheem test points over the full range. The tabulated results are provided in Table A-1 and Table A-2 in Appendix A. The results show that the test points generally agreed with each other and had an average difference of only 2.4 percent. The largest difference was seen at the higher indoor wet-bulb condition and lowest outdoor dry-bulb condition with a 7 percent difference in capacity. Some of the variation between Rheem and WCEC lab data was likely due to a different expansion valve used in the two systems and different refrigerant line-set lengths. When the points were separated by common wet-bulb temperatures they showed a similar trend of decreasing capacity with increasing outdoor dry-bulb temperatures.

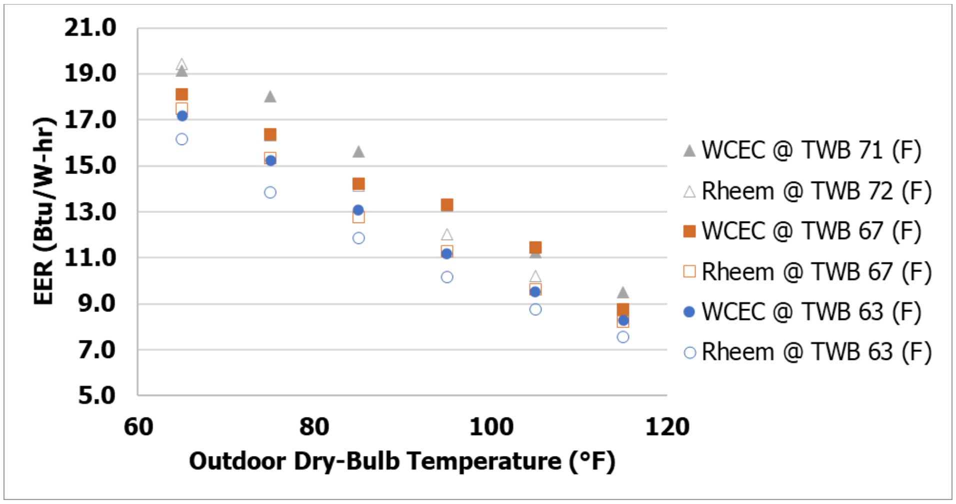
Figure 7: Total Cooling Capacity Over the Full Test Range



TWB=temperature wet bulb

Figure 8 shows the energy efficiency ratio (EER) for both data sets of the system over the full test range. In general, the cooling capacity and the EER of the system decreased with increasing outdoor air temperature, and with increasing indoor wet-bulb conditions, which is a typical trend for vapor-compression air conditioning systems. The test points generally agreed, but not as closely as the total capacity data with an average EER difference of 9.4 percent. This also may have been due to variations in the expansion valve, line-set length, and refrigerant charge used in each facility. The WCEC lab results showed a slightly higher overall EER.

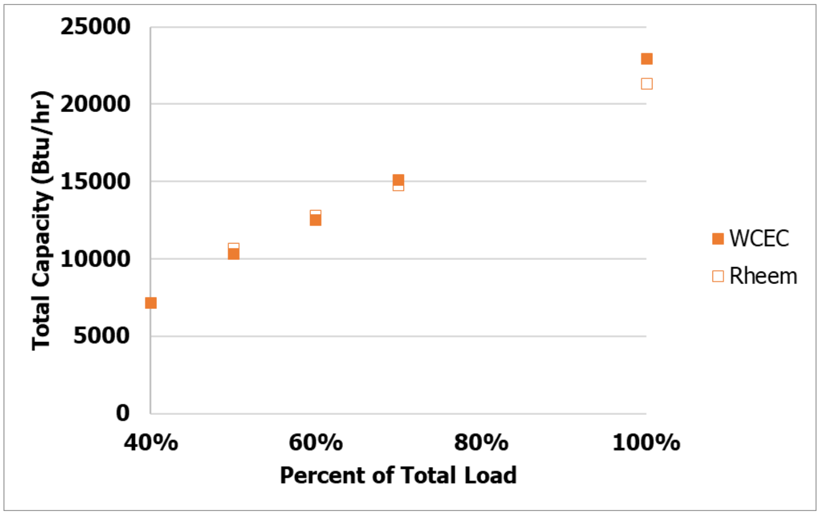
Figure 8: Cooling EER Over the Full Test Range



Part-Load Cooling Results

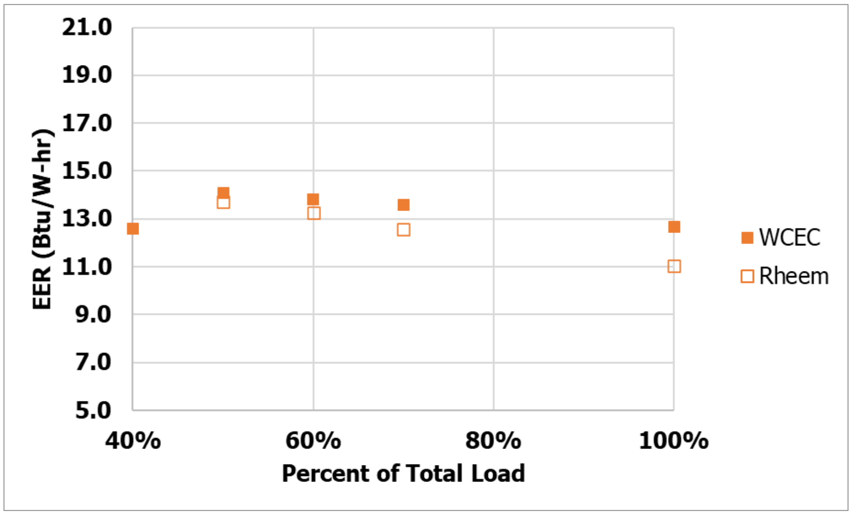
Figure 9 and Figure 10 show the total capacity of the system and the EER under partial load conditions for both WCEC and Rheem labs. These tests were conducted at a midpoint in the testing range (95°F [35°C] outdoor dry-bulb, 80°F [27°C] indoor dry-bulb, and 67°F [19°C] indoor wet-bulb). The tabulated results are provided in Table A-3 and Table A-4 in Appendix A.

Figure 9: Total Cooling Capacity Under Partial Load Conditions



Performed at 95°F (35°C) outdoor dry-bulb, 80°F (27°C) indoor dry-bulb, and 67°F (19°C) indoor wet-bulb.

Figure 10: Cooling EER Under Partial Load Conditions



Performed at 95°F (35°C) outdoor dry-bulb, 80°F (27°C) indoor dry-bulb, and 67°F (19°C) indoor wet-bulb.

Btu/W-hr=British thermal units per watt-hour

As expected, the total capacity decreased as the load percentage decreased. The EER showed improved performance as the speed of the unit decreased, with the exception of the 40 percent speed result, which was not a condition tested by Rheem. From the remaining three points, the total capacity measured by Rheem and WCEC agreed with an average difference of only 0.9 percent. The measured EER also generally agreed with an average difference of 5.0 percent. The discrepancy in efficiency was due to the relatively lower power measured by WCEC.

Full Speed Heating Results

Figure 11 and Figure 12 show the total heating capacity and EER for the full Rheem data set and the two WCEC test points. The tabulated results are provided in Table A-5 and Table A-6 in Appendix A.

Figure 11: Total Heating Capacity Over the Full Test Range

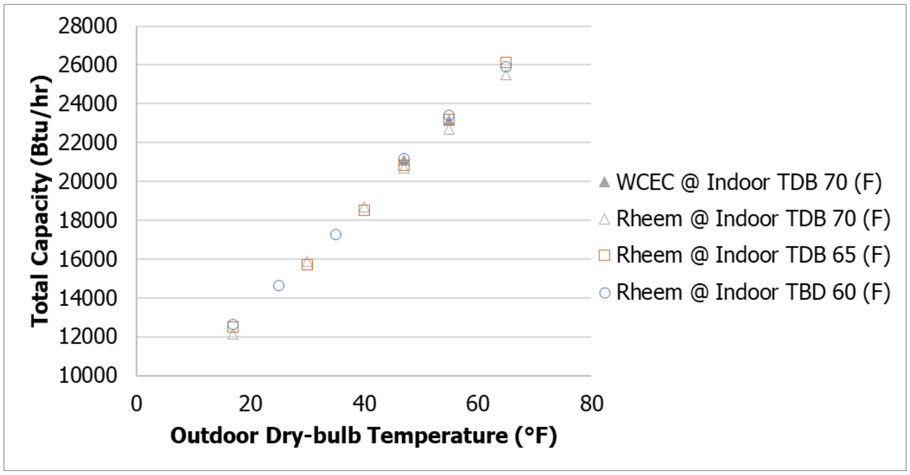
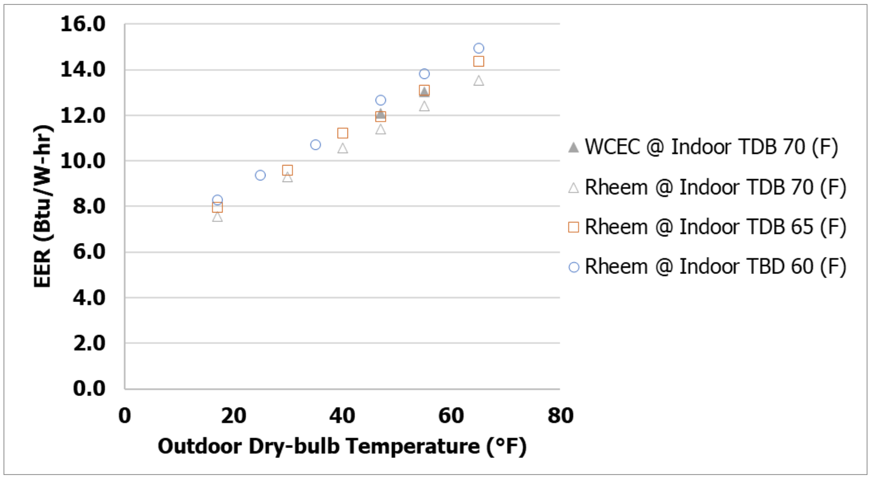


Figure 12: Heating EER Over Full Test Range



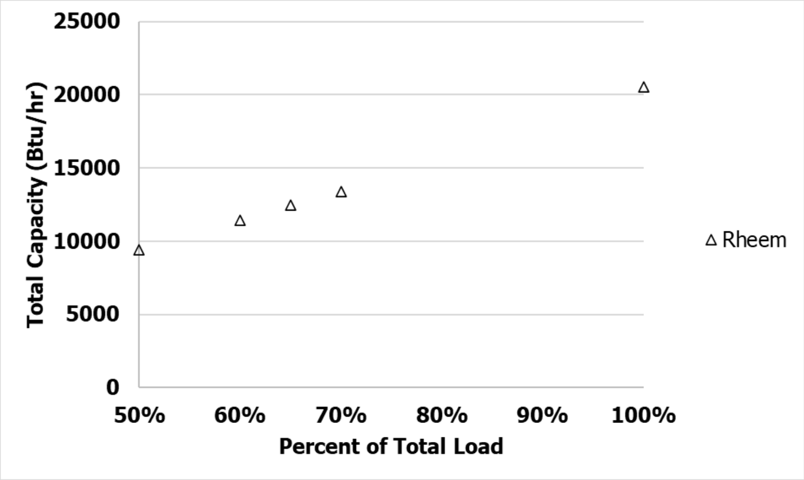
TDB=temperature dry bulb

Comparable points of heating data also found good agreement between the Rheem and WCEC data. Total capacity was measured at an average difference of 1.9 percent. EER had an average difference of 5.4 percent, likely due to the different expansion valve and line-set length. The total capacity showed a strong relationship with outdoor dry-bulb temperature with minimal variation due to indoor dry-bulb temperature. EER also showed a strong relationship with outdoor dry-bulb temperature with a small dependance on indoor dry-bulb temperature. The efficiency tended to increase at lower indoor dry-bulb temperatures.

Part-Load Heating Results

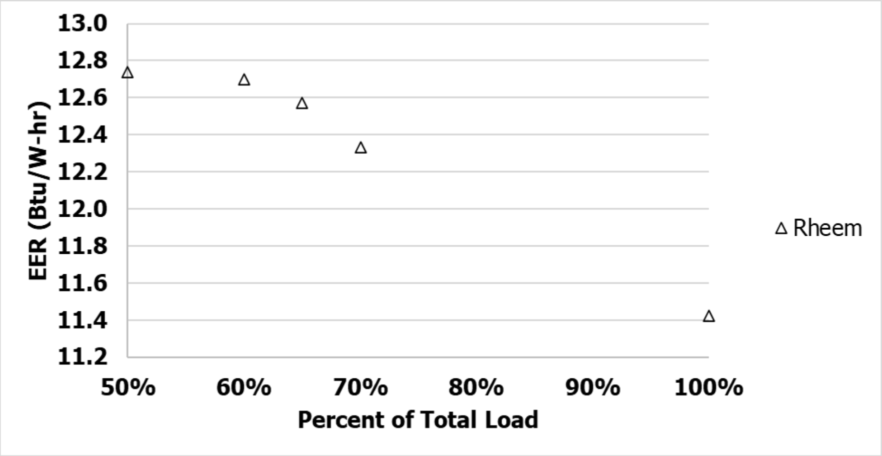
The part-load heating test data at the midpoint of the test range (47°F [8°C] outdoor dry-bulb, 43°F [6°C] outdoor wet-bulb, and 70°F [21°C] indoor wet-bulb) is shown in Figure 13 and Figure 14. The tabulated results are provided in Table A-7 in Appendix A. No part-load heat tests were conducted at WCEC.

Figure 13: Total Heating Capacity Under Partial Load Conditions



Performed at 47°F (8°C) outdoor dry-bulb, 43°F (6°C) outdoor wet-bulb, and 70°F (21°C) indoor. wet-bulb.

Figure 14: Heating EER Under Partial Load Conditions



Performed at 47°F (8°C) outdoor dry-bulb, 43°F (6°C) outdoor wet-bulb, and 70°F (21°C) indoor wet-bulb.

Heat Pump Laboratory Test Discussion

This project tested the performance of the near-term heat pump in the laboratory and validated the data collected by Rheem. The results were used to determine the seasonal energy efficiency rating (SEER) as well as to provide the data necessary to develop performance maps of the system. The heat pump was tested under a wide range of conditions at full compressor speed in both cooling and heating modes. It was also tested at lower compressor speeds to characterize the performance under partial load. The total capacity data showed a good correlation between the Rheem and WCEC points with an average percent difference of 2.4 percent for cooling full-speed points, 0.9 percent for part-load cooling points, and 1.9 percent for heating full-speed points. The EER measured showed larger differences between the Rheem and WCEC data with an average of 9.4 percent in full speed cooling points, 5.0 percent in part-load cooling points, and 5.4 percent in full speed heating points. In general, the WCEC tests showed higher efficiency with similar capacity at lower power.

Heat Pump Field Testing

The following section describes the results from field testing of the near-term solution.

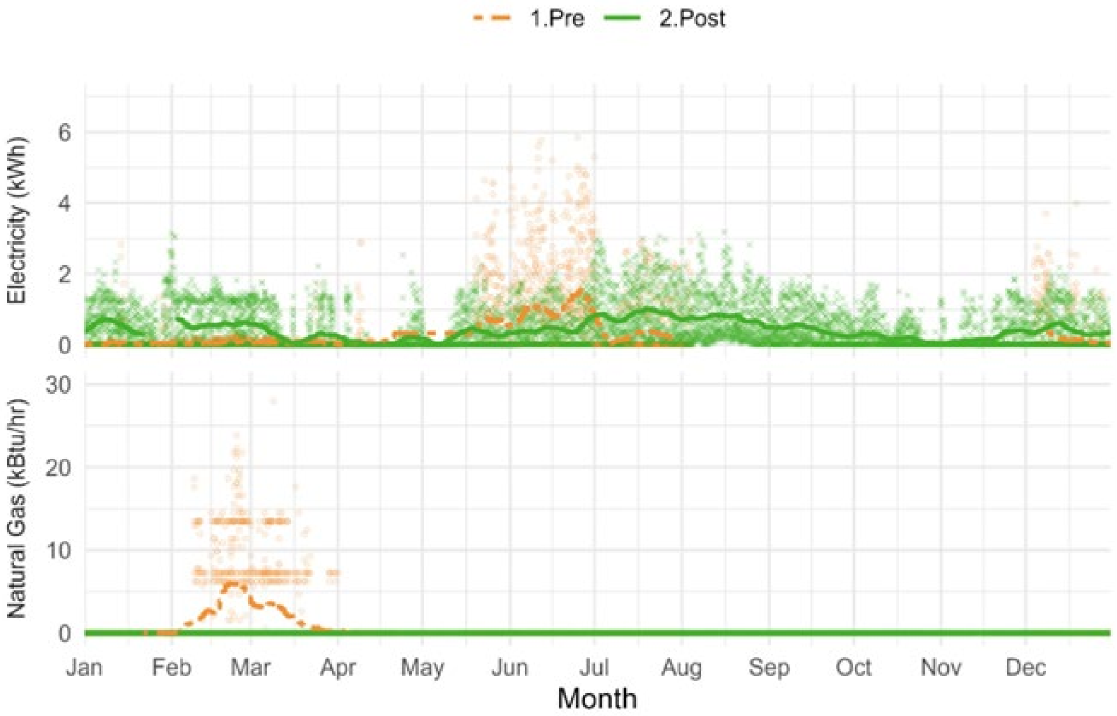
Measured HVAC Operation

The research team assessed HVAC energy use for both pre- and post-retrofit periods. In the context of this paper, winter spans from December to February, while summer encompasses July to September. Example data from Site 7 is presented in detail in Figure 19 with similar analysis for other sites available upon request.

Example Data for Site 7

Figure 15 presents example data from Site 7 depicting the hourly electricity and natural gas usage of the HVAC system, presenting a simultaneous comparison between the pre- and post-retrofit monitoring periods. The graph includes a line representing a 10-day rolling average of the data. During the cooling season, the electricity consumption profile for both the pre- and post-retrofit systems was similar comprising of the outdoor compressor and fan, as well as the indoor supply fan. Conversely, in the heating season, the electrical usage for the pre-retrofit systems, with exception of the baseline heat pump system, was comprised only of the supply fan, which closely aligned with the natural gas consumption of the furnace. The post-retrofit heat pump systems exclusively used electricity for both cooling and heating, which resulted in increased electricity use and no gas use in the winter. It is noteworthy that there were instances of missing gas usage data for December during the pre-retrofit period, underscoring the significance of using normalized energy data for direct comparison.

Figure 15: Measured Hourly HVAC Energy Use at Site 7



kBtu/hr=thousand British thermal units per hour

Switching heating fuels from natural gas used in the existing furnace to electricity for the retrofit heat pump created challenges in evaluating energy use directly. To compare the heating operation of the existing furnace with the retrofit heat pump, the hourly profiles of electricity and natural gas usage were analyzed before and after the retrofit. Boxplots,[3] like those shown in Figure 16 and Figure 17, were used to visualize the impact on energy use and thermal comfort for the retrofit system. The retrofit heat pump appeared to commence heating earlier in the morning compared to the furnace. This was likely a result of changes in thermostat schedules during the post-retrofit period due to the lower heating capacity of the heat pump relative to the existing furnace. It was also found that no significant operation of the electric heater kit in the retrofit heat pumps occurred.

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[3] The lower and upper limits of each box in the boxplot represent the 25th and 75th quartiles, and the middle of the box is the median. The thin vertical lines, or "whiskers," show the range of temperatures from minimum to maximum, excluding outliers. Points were considered outliers if they were not within 1.5*IQR (inter-quartile range) from the lower and upper limits of the box, where the IQR is the distance between the 25th and 75th percentiles.

Thermal comfort conditions before and after the retrofit at each site were assessed by comparing the measured zone temperature data. Although the retrofit heat pumps effectively maintained reasonable thermal comfort levels across all sites, changes in zone temperature distribution were observed during both summer and winter seasons at several demonstration sites. These changes were likely attributable to variations in occupant behavior, such as alterations in operating hours and thermostat adjustments for both cooling and heating. For Site 7, the variable-speed heat pump appeared to maintain the space temperature within a narrow range for both cooling and heating operations, as depicted in Figure 17. This finding aligns with the results of the occupant survey, wherein more than half of the respondents (n=7) indicated that the new heat pump better maintained setpoints than the old system, while the remainder reported no discernible difference. Figure 17 also shows that the occupant generally had lower zone temperatures in the summer and warmer zone temperatures in the winter. This would lead to increased energy use since there would be higher conditioning loads to meet these setpoint. The regression modeling used to evaluate energy use did not account for changes in indoor conditions so this behavior would tend to make the model predictions skew toward higher energy use for the retrofit heat pump.

Figure 16: Boxplot of Measured HVAC Energy Use Hourly Profile During Winter at Site 7

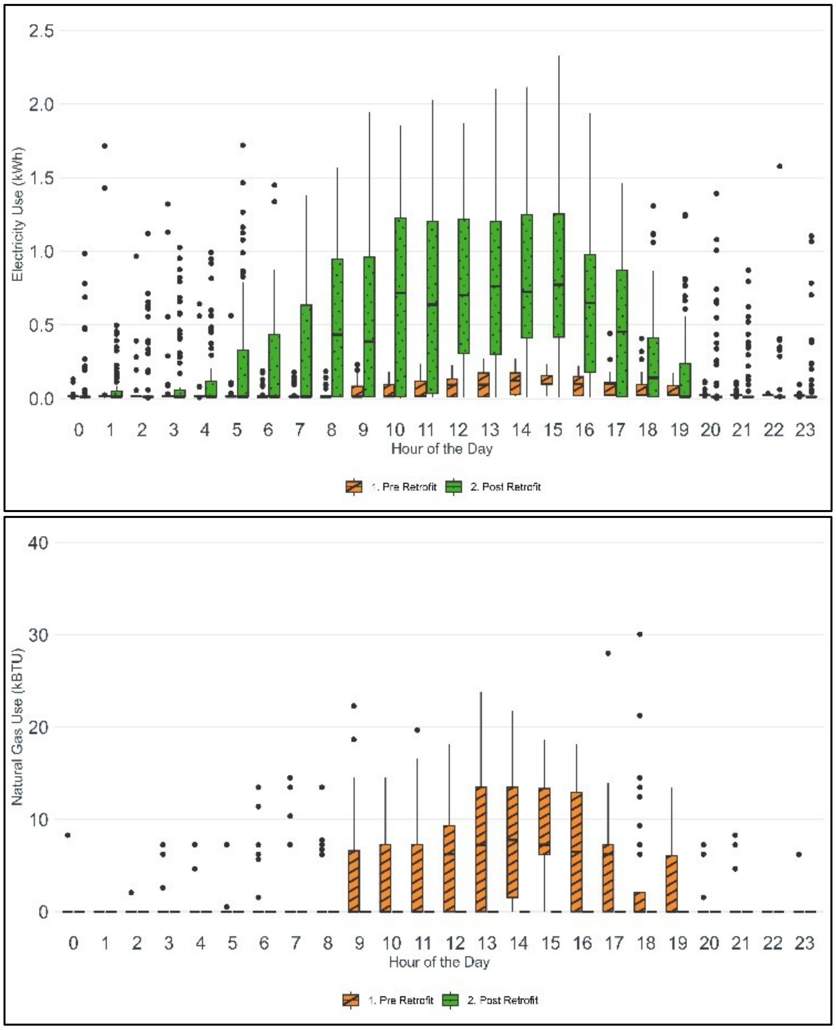
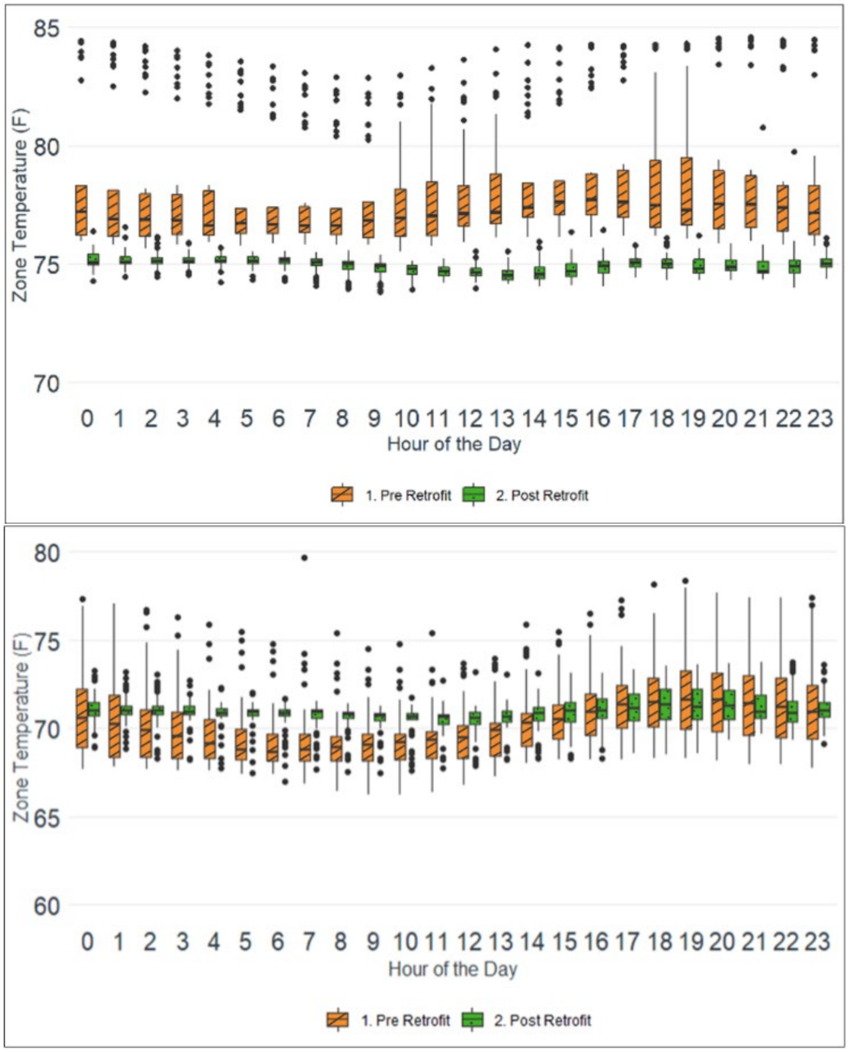


Figure 17: Boxplot of Measured Zone Temperature Hourly Profile at Site 7

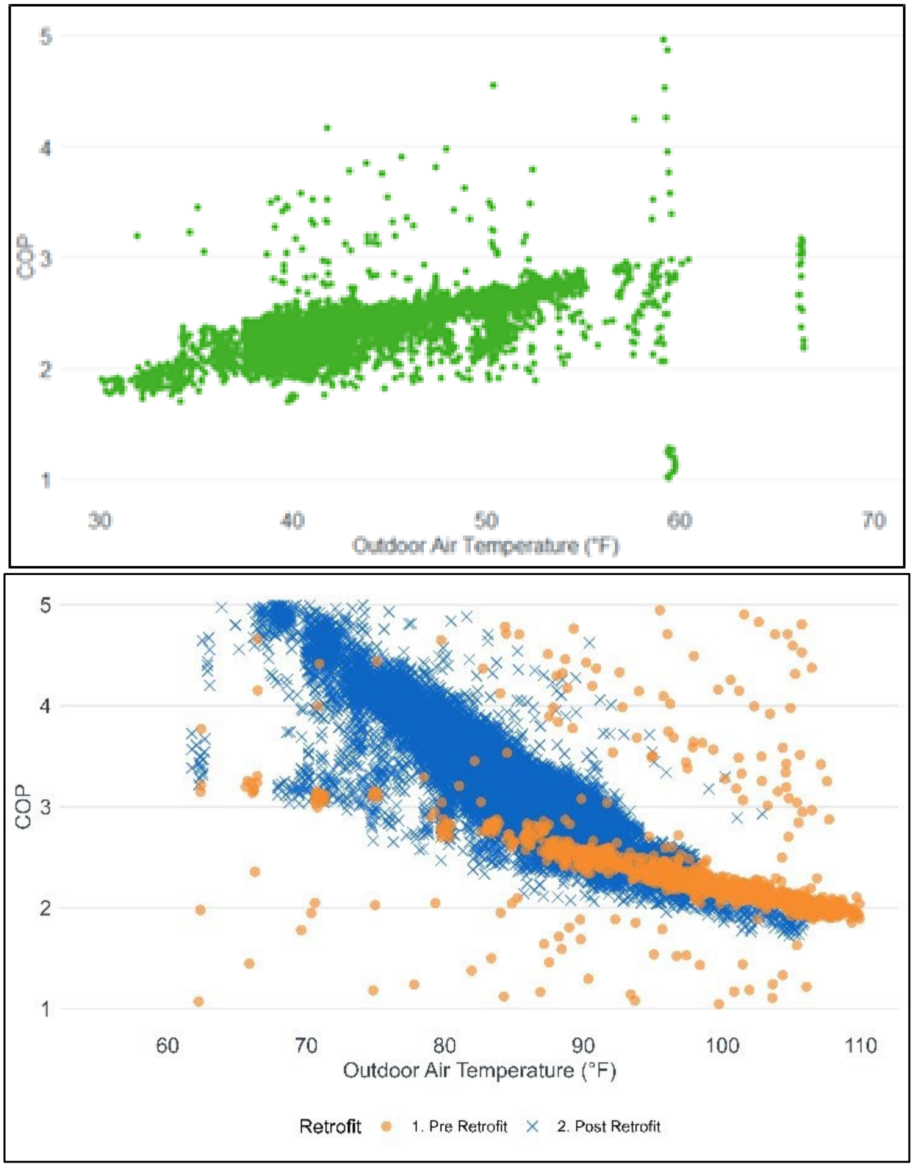


Summer (left) and Winter (right).

kBtu=thousand British thermal unit

Another way to evaluate the energy impacts of the retrofit is to compare the efficiency of the systems under similar operating conditions. This analysis was performed for Site 7 during steady-state conditions and the results are provided in Figure 18. During heating operation, the heating COP was observed to range between 2 and 3 for most installations, particularly when outdoor air temperatures fell within the range of 30°F (–1°C) to 60°F (16°C). The baseline gas furnace efficiency was between 0.5 and 0.8. Conversely, for cooling, the retrofit heat pumps exhibited COPs ranging from 2 to 5. The part-load efficiency of the retrofit heat pump during mild outdoor conditions showed the largest improvement in performance relative to the baseline equipment. As temperatures increased, the retrofit heat pump ran at full speed and the performance was reduced. It was observed that the retrofit system had slightly lower performance compared to the baseline systems at the hottest conditions. This behavior can be expected since variable speed systems were optimized for seasonal efficiency at part-load; whereas, the baseline single-speed equipment was optimized at the full speed condition.

Figure 18: Example HVAC Efficiency for Site 7



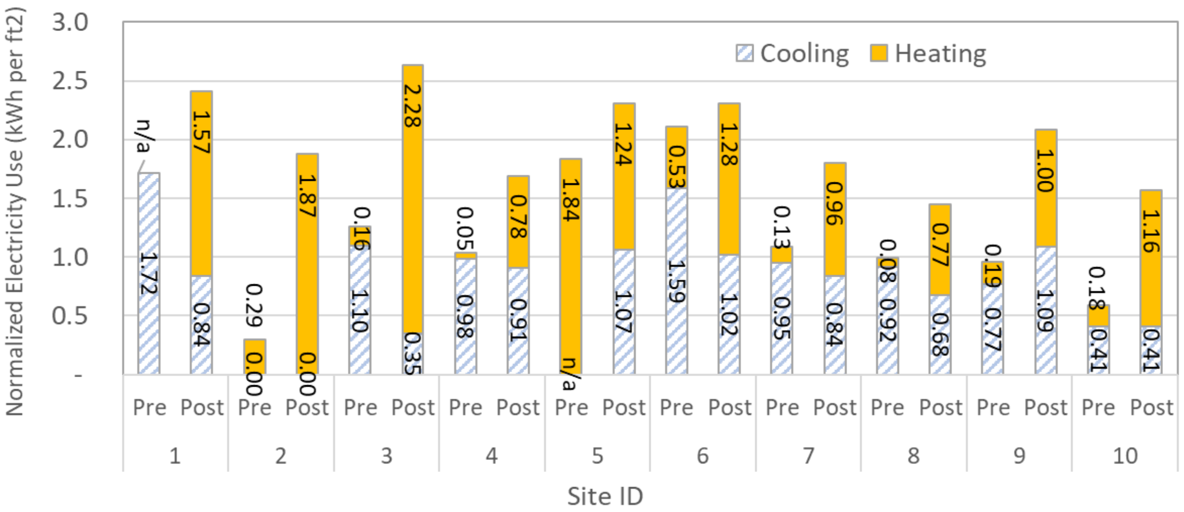
Next Generation HP Heating COP (left) and Existing AC Unit and Next Generation HP Cooling COP (right) at Site 7.

Normalized Energy Use

Annual HVAC energy usage varied significantly across sites due to factors such as occupant behavior, home vintage, and equipment efficiencies. Figure 19 and Figure 20 present the annual electricity and gas usage density, normalized against typical meteorological year 3 weather data, for both the existing and retrofit HVAC systems. It is evident that heating energy usage dominated for sites in CZ2 (sites 1 to 3), while the distribution between heating and cooling usage was more balanced for sites in CZ12 (4 to 10).

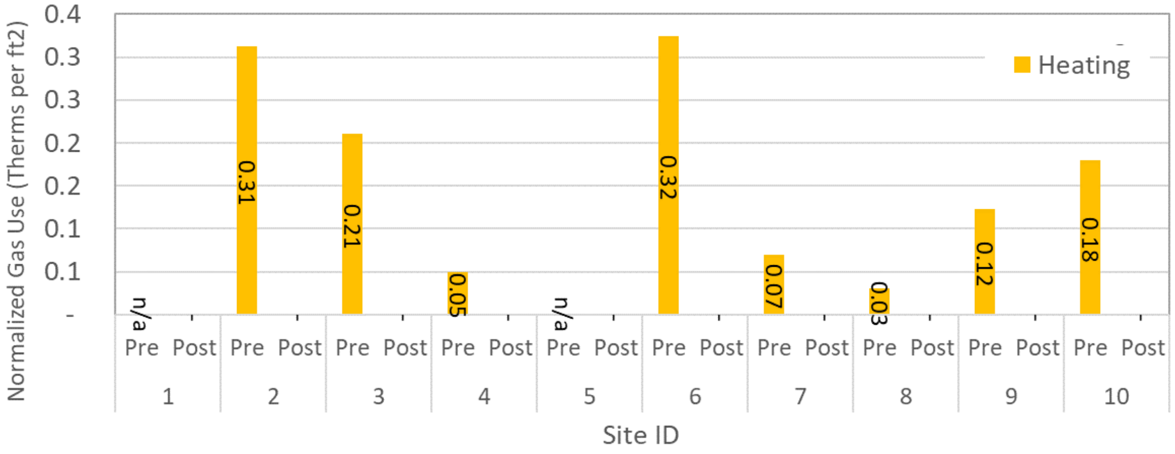
Upon comparing the pre- and post-retrofit periods, it was observed that electricity savings for cooling ranged from -41 percent to 68 percent across the sites. Although the heat pump systems demonstrated similar or higher efficiency compared to the old systems, the substantial variance in energy impacts among the homes can be attributed to occupant behavior. This includes changes in operating hours and thermostat adjustments, as reported in the Participant Survey.

Figure 19: Normalized Annual Electricity Use Density for Pre- and Post-Retrofit Periods



kWh per ft^2=kilowatt-hours per square foot

Figure 20: Normalized Annual Gas Use Density for Pre- and Post-Retrofit Periods



therms per ft^2=therms per square foot

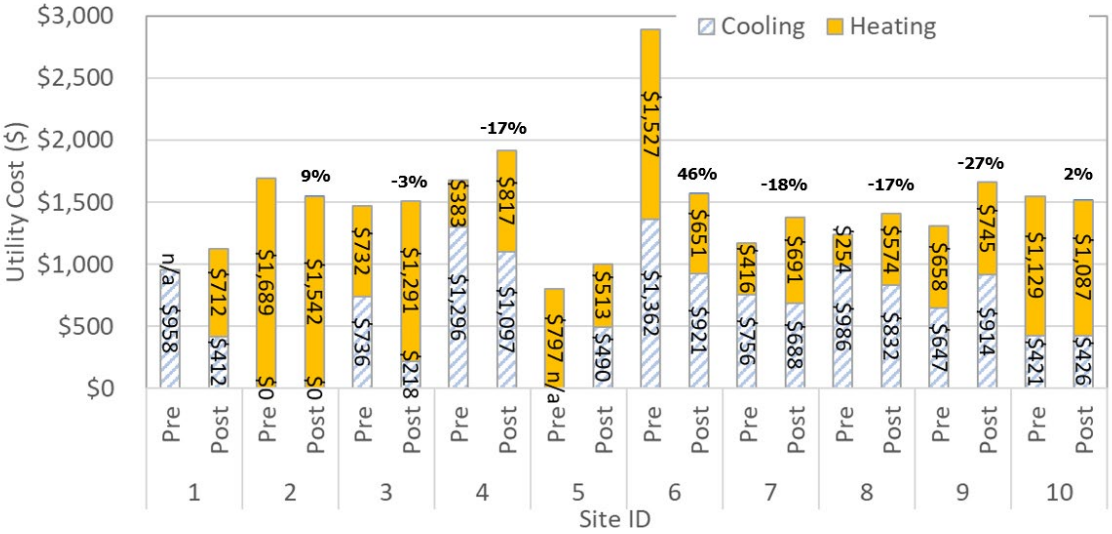
Utility Cost and Environmental Impact

The research team evaluated the energy costs associated with the post-retrofit system, as well as the calculated baseline for all homes, using the PG&E TOU-C rate (PG&E, 2024). Figure 21 illustrates the calculated annual costs for space heating and cooling in both the pre-retrofit gas scenario and the post-retrofit heat pump scenario. The percentage figures displayed above the post-retrofit bar indicate the differences between the two scenarios, where a negative number signifies an increase in cost. The findings indicate that, based on the analyzed rate structure, operating the heat pump likely incurred higher costs compared to the baseline gas system.

There was an increase in energy costs ranging from 3 percent to 27 percent for five of the eight sites for which there were valid data. The research team observed 9 percent and 2 percent cost savings for Site 2 and 10, respectively. Site 6, which had a 46 percent cost savings, had a tenant switch during the monitoring period, which may have impacted the result.

To put this in perspective, both gas and electricity rates and rate plans used in this analysis have experienced significant price increases. The average annual increase over a 15-year period has been 5.0 percent for gas and 7.2 percent for electricity, but in the past 5-year period the average annual increase has been much higher, at 9.6 percent for gas and 13.8 percent for electricity. Based on an equivalent unit of energy delivered to a home, electricity was 7.3 times more expensive than gas with the 2024 rate, thus exceeding the cost savings potential of electric heat pumps over natural gas heaters even though they are two to four times more efficient from an efficiency perspective. If electricity prices continue to increase at a higher rate than gas prices, it will significantly impact California's electrification goals and reduce the pace of market transformation to electric heat pumps, especially in existing residential homes.

Figure 21: Annual Utility Costs for Pre- and Post-Retrofit System

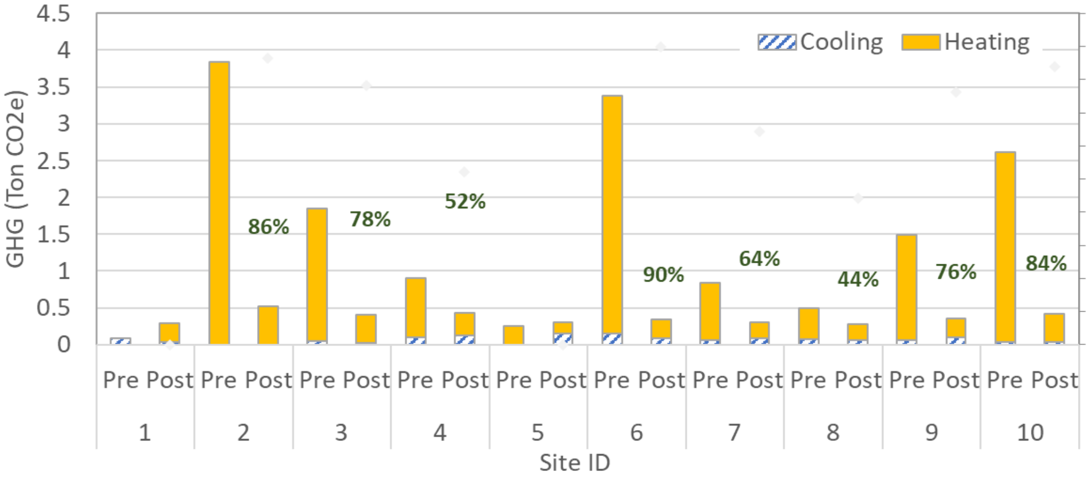


Note: reduction in cost shown as percentages atop post-retrofit result.

The California statewide grid hourly emissions factors, as provided by CEC, exhibit variability throughout the day and across seasons. This variability reflects the GHG emissions associated with electricity and gas usage, which are expected to change over time as the state's grid transitions towards cleaner energy sources. For instance, during the winter months, average GHG emissions from electricity use may be five to six times higher than in months such as May or June, when demands are lower and renewable generation is more abundant.

The GHG emission impacts of the retrofit were found to align with this trend. Figure 22 illustrates that GHG savings ranged from 44 percent to 90 percent across the sites, with larger savings observed for sites in CZ2 compared to those in CZ12. This discrepancy can be attributed to the fact that heating constitutes a larger portion of energy demand than cooling in CZ2, as opposed to CZ12.

Figure 22: Annual GHG Savings for Pre- and Post-Retrofit System



Note: reduction in GHG shown as percentages atop post-retrofit result.

Discussion of Field Results

The field results generally validated the performance of the next-generation heat pumps in terms of both space cooling and heating functionality, as well as their energy efficiency, which was found to be comparable to that of other efficient heat pumps on the market. Notably, at certain sites, improved space temperature control was observed attributable to the variable-speed capability of the new systems compared to the old single-stage systems.

During the cooling season, the COP for most installations exceeded 3 when outdoor air temperatures were below 90°F (32°C). However, as outside air temperatures increased to 110°F (43°C), the COP dropped to 2. For heating, heat pump COPs ranged between 2 to 3 when outside air temperatures were between 30°F (–1°C) and 60°F (16°C).

The assessment findings regarding energy usage, utility costs, and GHG emissions largely confirmed the anticipated impacts associated with replacing a split AC and gas furnace with a heat pump. With the analyzed rate structure, residents were likely to experience an increase in their utility bills. Moreover, regarding GHG emissions, the findings indicated a significantly positive impact showing reductions in GHG emissions at all sites with available data.

Upon reviewing the space temperature profiles, variations in occupant behavior were observed, particularly concerning changes in HVAC operational hours and thermostat setpoints. For instance, some owners adjusted their heating setpoints lower, while others increased them during winter. Such behavioral changes can influence energy consumption patterns and should be considered in assessing the overall impact of the retrofit.

Survey Responses

Participant surveys revealed that most were comfortable with the temperatures delivered by the heat pumps in both summer and winter. There were mixed reports on the speed with which the desired setpoint was reached, with nearly half indicating that the system was "somewhat slow" to heat up in winter. During the project, participants did not convey concerns about the flammability of the refrigerant used in the heat pumps.

Five participants observed a decrease in electricity bills compared to the previous summer, while three noticed an increase. Those who reported an increase noted that the summer had been hotter than previous years, making it challenging to attribute changes in electricity bills solely to the heat pump. Additionally, electricity rate increases in one utility territory complicated pre- and post-retrofit bill comparisons. The perceived difference in heating costs varied widely: five participants noticed a decrease, three observed an increase, and two saw no difference. Comparisons of heating costs were complicated by the switch from gas to electricity in 9 out of 10 cases.

At the project's end, participants were given the choice of a heat pump or furnace/AC to replace the experimental equipment. Nine of 10 chose a heat pump. Although units with R-454B were initially offered, Rheem was not able to get them ready in time for the replacement, so ultimately the replacement units had R-410A. One participant chose to return to a gas furnace because they had experienced bill increases and were concerned about the operating costs moving forward. Nine of 10 participants reported they would recommend the heat pump to others, a good indication that most participants were satisfied overall with their heat pumps.

Heat Pump Performance Modeling

Performance modeling was conducted as an alternative approach for estimating the impact of the retrofits without uncertainties related to uncontrolled variables in the field. The key results for the modeling analysis were related to cost and GHG emissions implications of the low-GWP heat pump system evaluated for this project. The two baseline systems were intended to represent other replacement options including a minimum efficiency air conditioner and gas furnace, and a minimum efficiency electric heat pump. Figure 23 shows the annual cost savings for the low-GWP heat pump relative to the two baseline systems. In most cases there was a clear reduction in utility costs relative to both baseline systems. Some significant cost increases were observed in the colder climate zones CZ1 and CZ16. In those cases, the relative cost of gas compared to electricity caused higher heating costs for the electric heat pump. Using the utility cost data previously presented, the cost of an equivalent unit of energy for electricity was five times more expensive than for gas.

One variable that had a significant impact on the operating cost for the heat pump was the sizing of the systems. The EnergyPlus autosize feature was used for modeling, which sizes the capacity based on a design load. However, it was found that the heat pump capacity was lower than the design loads and relied on electric resistance backup heat to satisfy the remaining load. A test was performed in CZ1 and CZ16 where the heat pump was manually sized to meet the full design load. Increasing the capacity of the heat pump reduced reliance on supplementary electric resistance heat and had a noticeable impact on heating energy use. With the larger sized heat pump the cost to operate the low-GWP heat pump was reduced by 14 percent for CZ1, and 8 percent for CZ16.

The average cost savings to operate the low-GWP heat pump compared to the baseline heat pump was 15 percent across all climate zones (Figure 23). By contrast, the average cost savings compared to the baseline AC with furnace was only 2 percent across all climate zones. The annual operating cost savings for the low-GWP heat pump compared to the AC with furnace system ranged from -$978 (a cost increase) in CZ16, to $113 in CZ8. For all climate zones with an annual operating cost reduction relative to the AC with furnace, the average reduction in energy costs was $68, and for the four climate zones that experienced a cost increase, the average annual operating cost increase was $113.

Figure 23: Annual Cost Savings for Low-GWP Heat Pump Relative to Baseline Systems

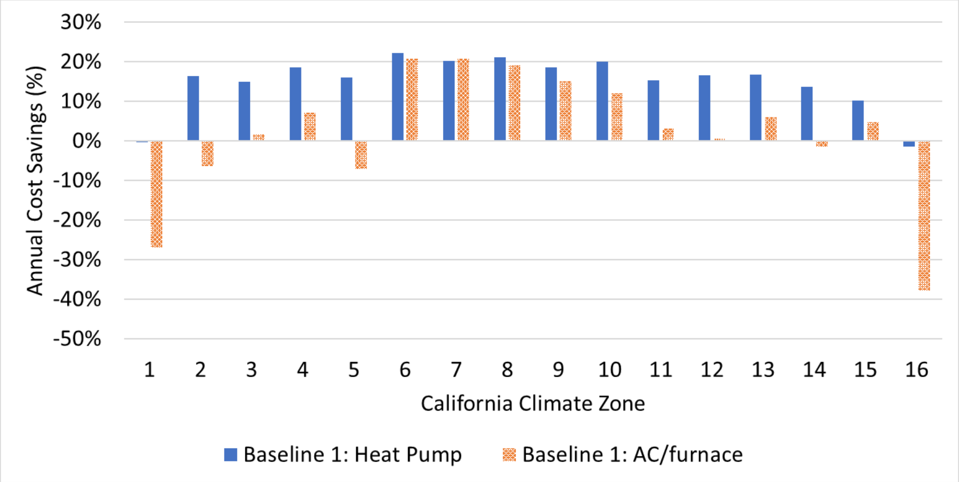
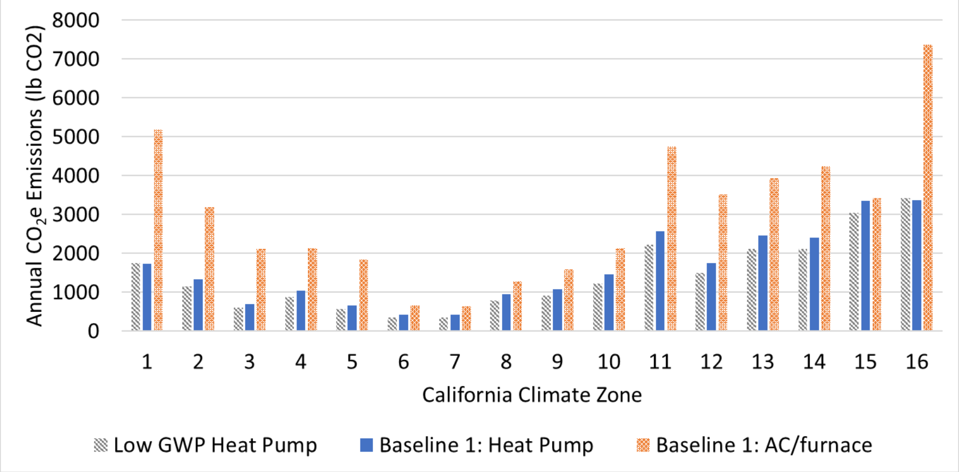


Figure 24 shows the annual GHG emissions related to operating the heating and cooling systems modeled. Substantial reductions in GHG emissions were observed for both heat pump systems, particularly in the climate zones with higher heating loads. The low-GWP heat pump achieved an average reduction in GHG emissions of 15 percent compared to the baseline heat pump and a 121 percent reduction compared to the baseline AC with furnace. The maximum reduction in GHG emissions relative to the AC with furnace system was in CZ3 with a 250 percent reduction in GHG emissions for the low-GWP heat pump.

Figure 24: Annual Greenhouse Gas Emissions from Each System Modeled



lb CO_2=pounds of carbon dioxide

Medium-Term Solution

The MPHX developed to improve the efficiency of the medium-term solution was tested on both small-scale and full-scale prototypes.

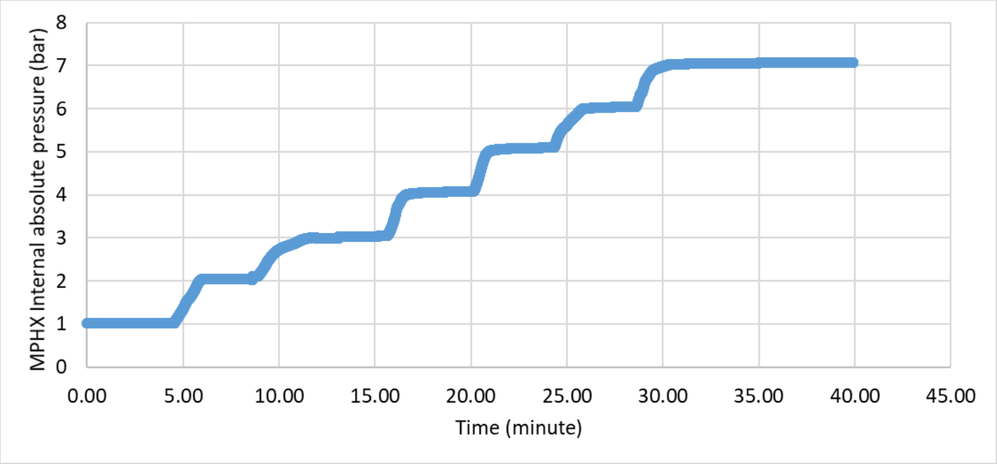
Heat Exchanger Testing

Heat exchanger testing consisted of mechanical and thermal testing to demonstrate durability and performance of the product.

MPHX Mechanical Testing Results

Mechanical integrity testing was performed through static tests and limited cyclic tests. Static pressure testing of MPHX was performed up to 7 bar (100 pounds per square inch absolute, or psia). The MPHX internal channels were pressurized with 1 bar step increments and 2 minutes of holding time at each step (Figure 25). At the maximum tested pressure of 7 bar (100 psia) the hold time increased to 10 minutes to ensure that the MPHX was holding pressure without a leak. The MPHX core was inspected throughout the test with soapy water, and no leak was observed.

Figure 25: MPHX Static Pressure Testing



To test sudden pressurizing impact on the structural integrity, the MPHX core was subjected to more than 20 cycles of pressurizing and de-pressurizing events between 7 bar and atmosphere. As is seen in Figure 26 and, once the MPHX was pressurized, it was held for one minute and then it was purged, and the cycle was repeated after 15 seconds. At the end of cyclic pressure testing, as shown in Figure 27, the MPHX core held its structural integrity with no visible plate failure on the outer walls of the water plates or on the air channels (space between the water plates).

Figure 26: MPHX Cyclic Pressure Testing

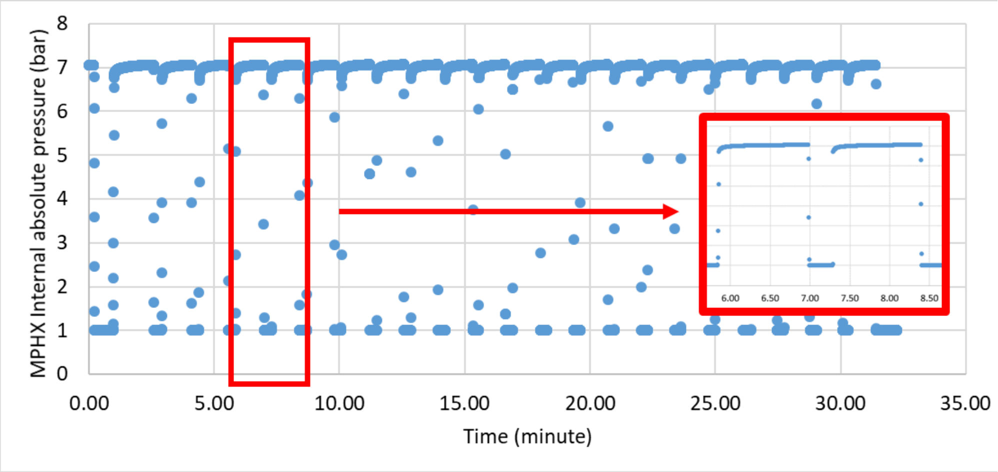
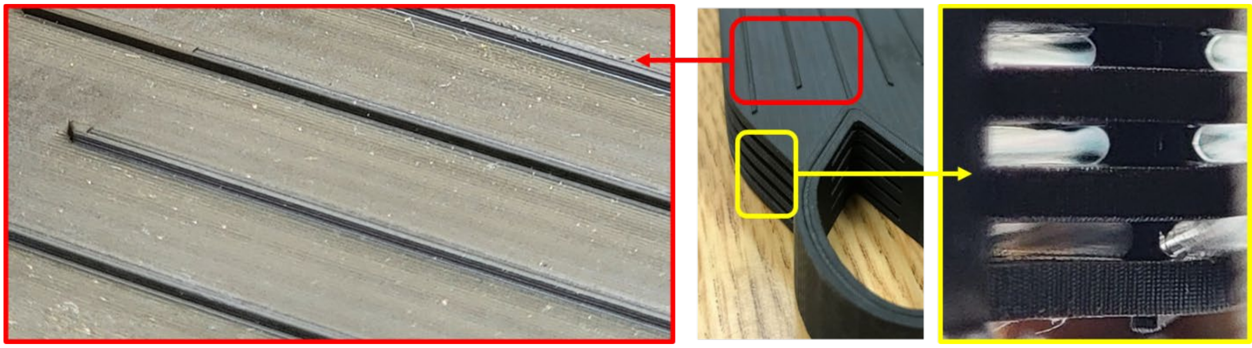


Figure 27: Photo of MPHX Outer Wall After Cyclic Testing



MPHX Core Outer Wall at the end of experiments, inset picture shows a few air channels with no visible sign of failure.

MPHX Thermal Testing Results

Figure 28 shows the experimental results and performance. Figure 28(a) plots the experimental heat transfers versus the air side Reynolds number. In general, the air side calculations predict a larger heat transfer than the water side calculations, with minimum, maximum, and average percent differences of 0.58 percent, 18.6 percent, and 9.1 percent, respectively. Figure 28(b) plots experimental effectiveness (with *Cr* values of around 0.9, 0.7, and 0.53) as well as theoretical effectiveness for a counterflow heat exchanger for these different *Cr* values. The effectiveness was high, with all exceeding 0.8. In general, the data points followed the expected trends, with effectiveness increasing as the Number of Transfer Units[4] (*NTU*) increased, and lower *Cr* value data points having higher effectiveness. Figure 28(c) plots experimental versus theoretical effectiveness, with the experimental effectiveness skewing higher. The minimum, maximum, and average percent differences between experimental and theoretical effectiveness were 4.4 percent, 13.8 percent, and 8.4 percent. One potential reason for this deviation could be the larger air side heat transfer values skewing the experimental effectiveness higher (and, as can be seen from Figure 28(a), the air

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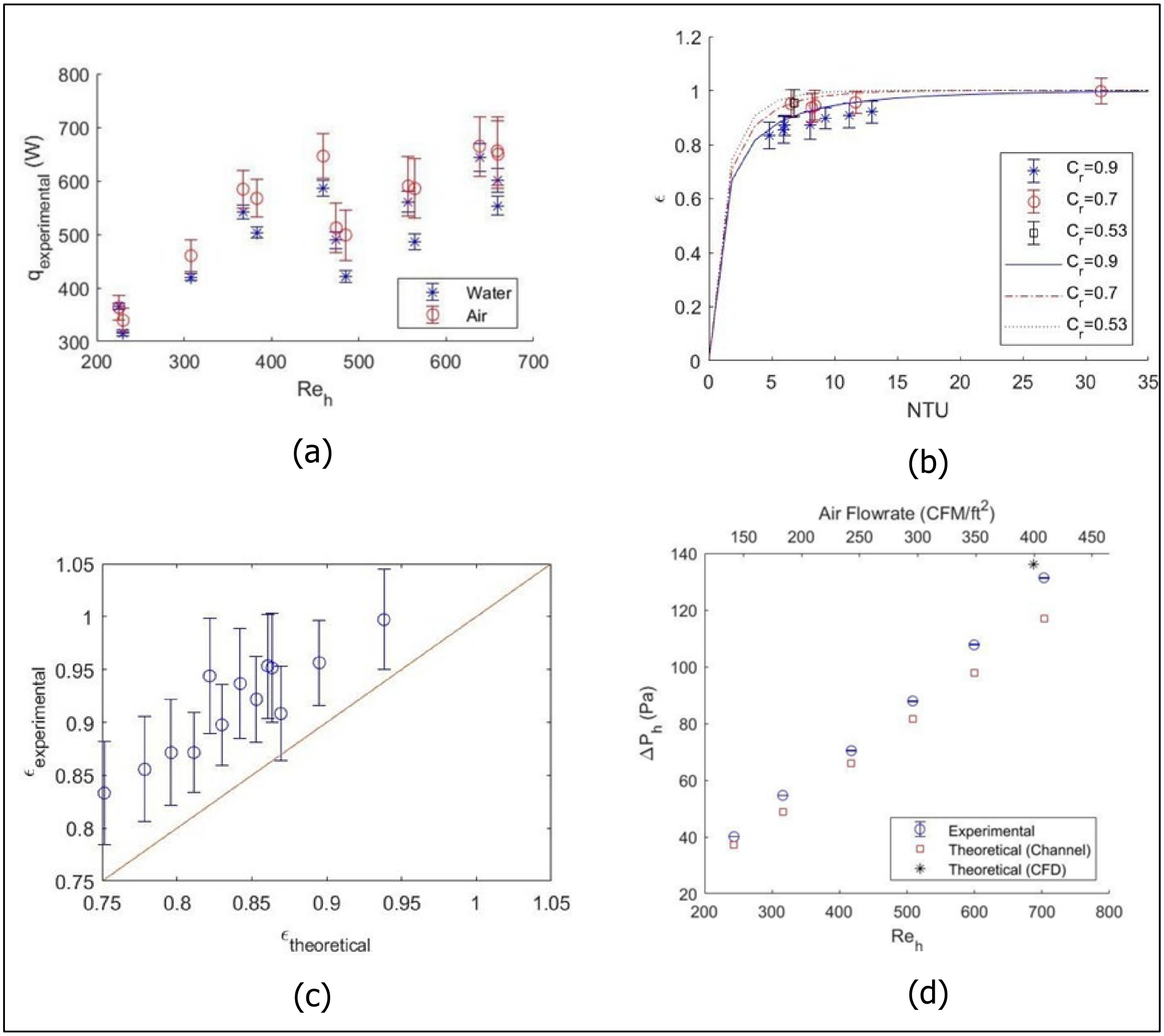
[4] Number of Transfer Units is a dimensionless value that expresses the effectiveness of a heat exchanger and defined as the heat transfer coefficient divided by the minimum heat capacity rate.

side heat transfers had greater uncertainties). It should be noted that while in Figure 28(b), effectiveness tended to skew lower than the prediction lines, not all of the data points plotted were at the exact *Cr* shown in the legend, and they were plotted assuming that they had the same *NTU*, while in reality, experimental and theoretical *NTU* values differed. Theoretical effectiveness and *NTU* were calculated with theoretical *UA*, which is the overall heat transfer rate multiplied by the heat transfer area, being obtained by the total thermal resistance. Air and water side resistances were calculated using heat transfer coefficients, which were obtained from the correlations for laminar flow in a duct for the air side, and correlations for flow in a pin array developed by the group for the water side.

Figure 28: MPHX Thermal Testing Results



(a)

(b)

(c)

(d)

(a) Experimental heat transfers for the MPHX versus air stream Re, (b) experimental effectiveness versus *NTU* and theoretical counterflow heat exchanger effectiveness, (c) experimental versus theoretical effectiveness, and (d) air side pressure drop versus air stream *Re* and CFM per square foot cross-sectional area.

CFM/ft2=cubic feet per minute per square foot; RE$_h$= air side Reynolds number

In addition to the heat transfer tests, air pressure drop tests were performed, with the results shown in Figure 28(d). For these tests, there was no water flow. The experimental results were plotted along with two theoretical pressure drops: one, the theoretical pressure drop predicted only accounting for passage of air through the MPHX channels, and the other, the predicted pressure drop based on a computational fluid dynamics (CFD) simulation, which included flow passage over the headers. As expected, the measured pressure drop was higher than the theoretical pressure drop when the liquid headers were neglected, with minimum, maximum, and average percent differences of 6.7 percent, 11.7 percent, and 9.0 percent, respectively. The measured values also ended up being slightly below the CFD-predicted pressure drop, with a percentage difference between this value and the nearest measured experimental pressure drop of 3.4 percent. Water pressure drop tests were also done, with a maximum pressure drop of 65 millibar (0.94 pounds per square inch) at a flowrate of 1,960 grams per minute.

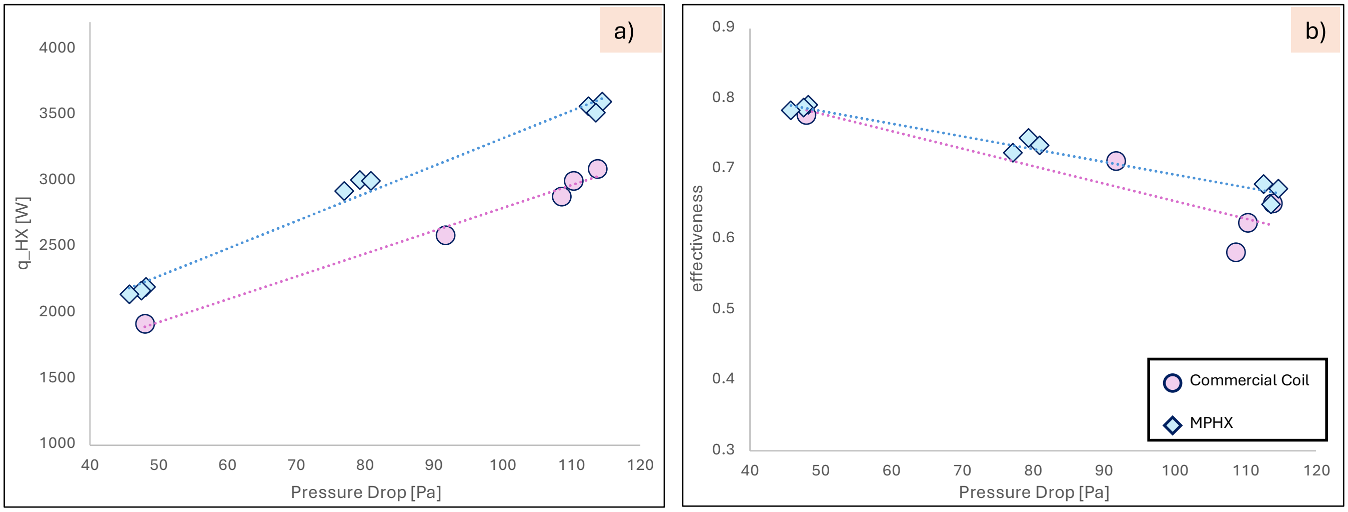
Full-Scale Testing

Full-scale testing of a 1.5-ton MPHX coupled to an ATW heat pump was conducted at WCEC and compared to a 1.5-ton commercial fin-tube coil. The MPHX used for testing was built with SLA 3D printing that did not allow for optimal channel spacing or length. This impacted the performance of the MPHX but allowed for validation of a coil model that was used to generate estimates of performance for an injection molded (IM) MPHX.

Isolated Testing

Figure 29 illustrates the heat transfer rate [Figure 29(a)] and effectiveness [Figure 29(b)] in relation to air-side pressure drop for both the MPHX and the commercial coil. For equivalent pressure drops, the MPHX demonstrated a heat transfer rate that was 12.3 percent higher than the commercial coil with a pressure drop of 48 pascals (Pa) for both heat exchangers. At the higher pressure drop range of 112 to 115 Pa, the MPHX's heat transfer rate exceeded that of the commercial coil by 14 percent, with the commercial coil experiencing pressure drop of 114 Pa. Furthermore, the effectiveness of the MPHX was higher by 1.3 percent and 2.4 percent at these equivalent pressure drop ranges, respectively. Overall, the results indicated a close alignment in performance regarding effectiveness, with the MPHX consistently exhibiting marginally better performance. While these differences are modest, they suggest a slight advantage for the MPHX, especially at higher air pressure drops, where the performance differences begin to increase.

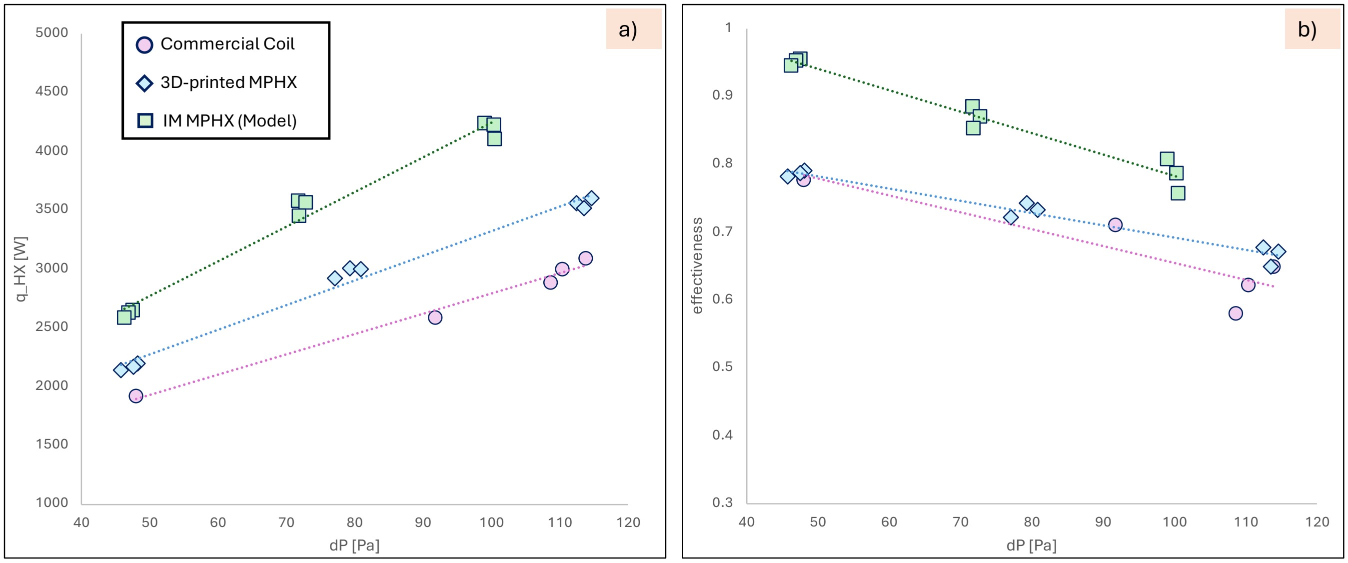
Figure 29: MPHX Heat Transfer Rate Compared to Commercial Coil



Heat transfer rate (a) and Effectiveness (b) vs. Air-Side Pressure Drop for MPHX and Commercial Coil.

Using the parameters for the IM version, the validated MPHX model was run to evaluate the performance of the IM version under the same test conditions as the current MPHX and the commercial coil. It was anticipated that the IM version would perform better than both heat exchangers, as illustrated in Figure 30, where heat transfer rate [Figure 30(a)] and effectiveness [Figure 30(b)] are compared for the three units against air-side pressure drop. The IM MPHX demonstrates effectiveness of 0.76 to 0.95, compared to the 3D-printed MPHX at 0.65 to 0.79 and the commercial coil at 0.58 to 0.78 under the specified conditions. Additionally, the heat transfer rates were as follows: 2580 W to 4230 W for the IM MPHX, 2150 W to 3600 W for the 3D-printed MPHX, and 1920 W to 3090 W for the commercial coil.

Figure 30: Expected Performance for an Optimized MPHX



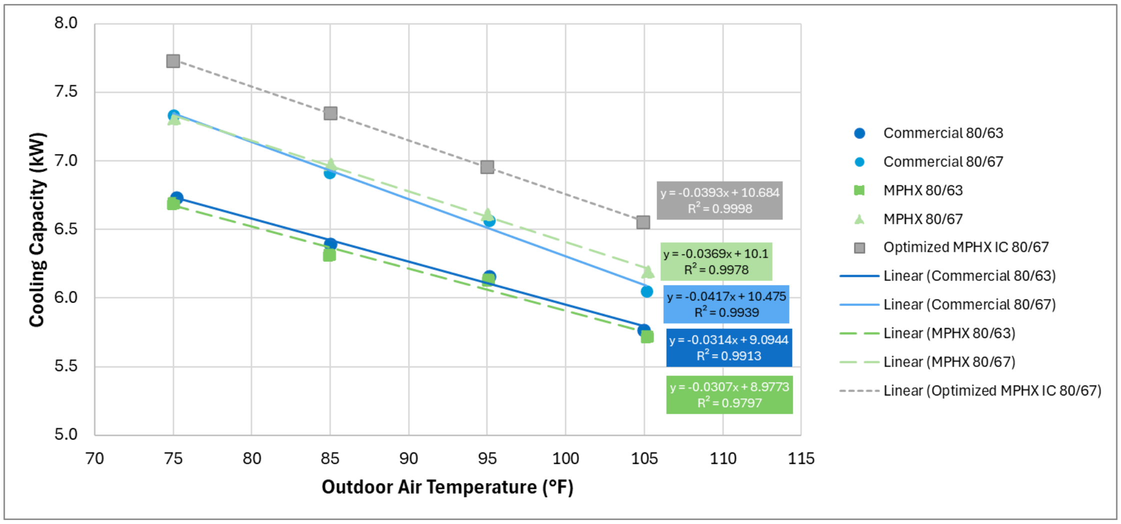
Comparison of an optimized MPHX using the injection-molding (IM) approach versus the current 3D-printed version of the MPHX and the commercial coil.

Air-to-Water Heat Pump Testing

The performance of the ATW heat pump coupled with both the commercial and MPHX heat exchangers was evaluated and compared. Improving the effectiveness of the heat exchanger in the secondary loop of the heat pump system will result in improved performance of the heat pump by reducing the temperature lift required by the compressor. Based on the results of the benchtop testing showing similar effectiveness between the commercial coil and the MPHX, an additional series of tests were performed to evaluate the impact of an improved heat exchanger design. This test included additional cooling capacity achieved with an in-line water heater to simulate the added effectiveness of the improved heat exchanger.

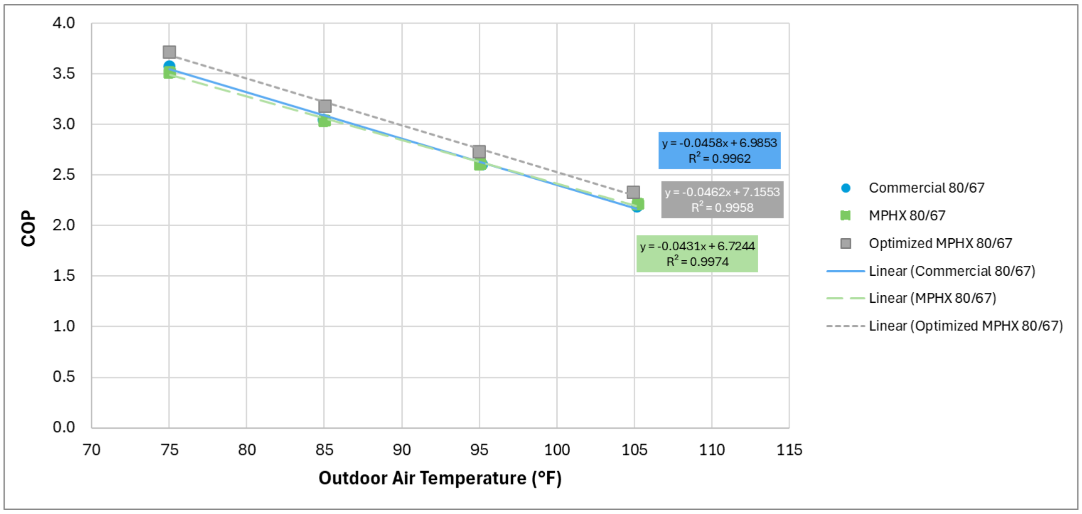
Figure 31 shows the water-side capacity measured in the laboratory for both the commercial coil and the MPHX. The results show very similar performance between the commercial coil and the MPHX, which confirms the findings from the benchtop testing. This is an impressive result considering the MPHX is constructed from a much lower conductivity plastic than the commercial coil made from high conductivity metals. Figure 31 also shows the additional capacity achieved from an improved heat exchanger design.

Figure 31: Cooling Capacity Measurements Comparing the Commercial Coil to the MPHX



The added cooling capacity to simulate the improved heat exchanger was 1,270 W, but interestingly the system capacity only increased by 345 W to 422 W compared to the other coils tested. This capacity increase did not have a significant impact on the power draw of the unit, and therefore, this translated to a similar average increase in efficiency of 5.3 percent for the overall ATW heat pump system. The COP for the improved heat exchanger along with the tested MPHX and commercial coil is presented in Figure 32.

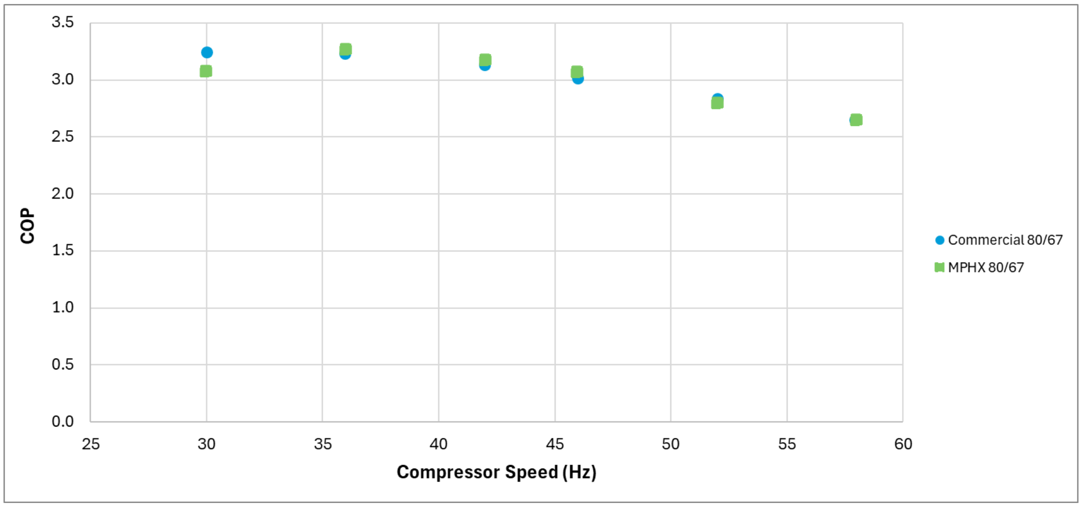
Figure 32: System COP for Air-to-Water Heat Pump Connected to Hydronic Air Handler



Further analysis determined that the improved heat exchanger capacity did not have the intended effect due to limitations in the ATW heat pump condenser capacity. The simulated capacity added by direct heating of the water resulted in increased secondary loop temperatures, reducing the capacity of the hydronic coil. A larger or more effective condenser would have been expected to maintain secondary loop temperatures and achieve the full capacity increase due to the improved heat exchanger.

Additional testing of the ATW system coupled with both the MPHX and commercial coils was conducted at part-load. These tests were all conducted at the 95°F (35°C) outdoor air condition and 80°F (27°C)/67°F (19°C) (dry-bulb/wet-bulb) indoor condition. The results showed that the heat exchangers performed similarly at part-load conditions as well (Figure 33).

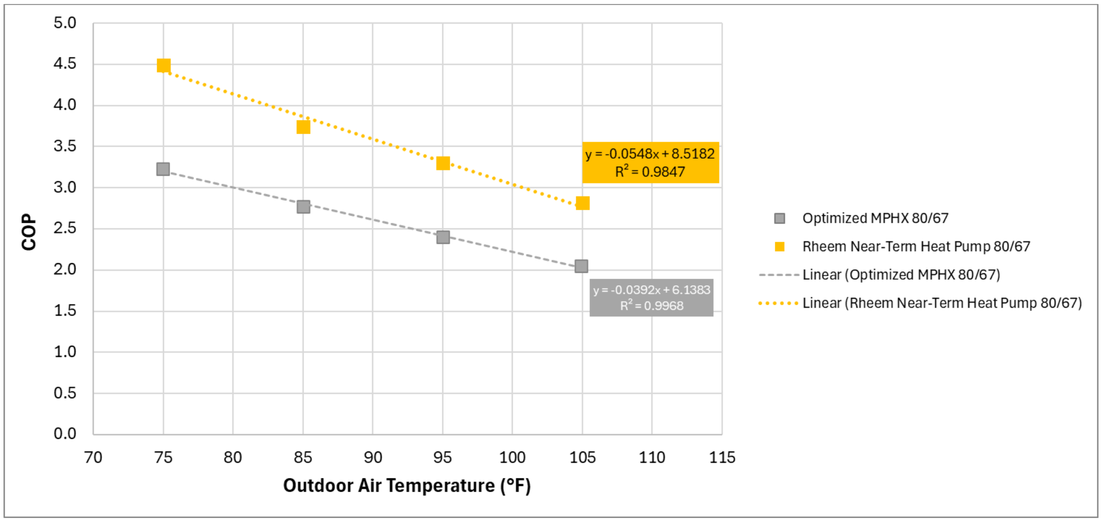
Figure 33: System COP for Air-to-Water Heat Pump Connected to Hydronic Air Handler



Heat pump operating at part-load at the 95°F (35°C) outdoor air condition and 80°F (27°C)/67°F (19°C) indoor air condition.
Hz=hertz

To evaluate the impact of the ATW heat pump system to a more typical air-to-air heat pump system, a comparison of the efficiency of both the medium-term solution with improved heat exchanger design and the near-term solution is provided in Figure 34. It should be noted that for Figure 34, the performance of the ATW system differed from earlier plots since it was based on the air-side capacity, which included the impacts of fan heating.

Figure 34: Near-Term vs. Medium-Term Solution Performance Comparison



Efficiency of air-to-water heat pump with central air handler and optimized MPHX compared to near-term solution air-to air heat pump

Figure 34 highlights the efficiency penalty of using ATW systems in combination with a central air handler. The performance of the air-to-air near-term solution tested for this project showed a 37 percent higher efficiency for the ATW heat pump with marginally improved hydronic heat exchanger. It was not an ideal comparison because the equipment components differed, but the result showed that replacing conventional forced-air distribution systems with an ATW heat pump has significant implications on efficiency.

CHAPTER 4:
Discussion

Transitioning to heat pump technology for space heating in California is a key step to achieving the state's ambitious climate goals. There are clear benefits to electric heat pumps relative to gas furnaces when considering the GHG emissions from residential heating systems. The cost to operate the heat pump, however, was demonstrated to be higher in some of the colder climate zones in California. Higher energy costs are expected to impact adoption rates and slow market uptake of heat pump technologies. Strategies to mitigate these impacts by providing different electricity fee structures for homes with electric heating systems in cold climates should be considered to support adoption. The cost of installing a heat pump often exceeds the cost of replacing a gas furnace, which can be exacerbated by the need to upgrade electrical services in a home. For these reasons, a wide-scale transition to heat pump technology for space heating will likely require incentives for ratepayers in the colder climate zones.

Near-Term Market Outlook

Users' and installers' experiences in the field demonstration project suggest that the use of A2L refrigerants in heat pumps does not pose significant challenges or barriers. Neither users nor installers had concerns about the flammability of the refrigerant, and the adaptations installers made to accommodate the refrigerant were minimal. This suggests that the market for heat pumps that use A2L refrigerants will be largely driven by refrigerant policy and the market for residential heat pumps in general. R-454B was permitted for all new residential heat pump installations as of July 1, 2024, and became required as of January 1, 2025. Further reductions in the allowable level of GWP for refrigerants, to near zero for example, are under discussion, but it is unclear whether such changes will occur.

Safety protocols may need to be established to address the increased flammability of A2L refrigerants. The latter may require special handling during transport, such as using compressed cylinder racks to hold the refrigerant on service vehicles. The Materials of Trade exceptions in the Code of Federal Regulations (49 CFR 173.6) exempt service vehicles transporting HVAC equipment from flammability signage requirements. Safety procedures related to installation also need to be developed, including whether standard "heat kits" used by HVAC contractors are adequate to mitigate the risk of fire.

Another change that is anticipated – on the equipment design side – is an update to the gauge hoses to allow for direct connection to heat pumps with A2L refrigerant. R-454B cylinders currently require special adapters to connect to the gauge hoses, adding a small amount of additional cost and effort, though no additional complication or risk.

While there were some concerns about working with A2L refrigerants at the outset of the study, this emerged as a non-issue. Neither contractor who worked on the project reported any problems or concerns about installing low-GWP heat pumps with an A2L refrigerant. One noted that the switch to low-GWP refrigerants is an inevitable part of the industry. The contractors observed that the trade has been subject to many policy-driven changes over the last few decades, and the move to low-GWP refrigerants is no different from earlier changes. They expressed no concerns about the safety or customer appeal of A2L refrigerants and expect consumer demand to increase as the products become commercially available (and required by CARB and U.S. EPA regulations).

The biggest supply-side challenge the installers foresee in a widespread transition to low-GWP heat pumps is the ability to train HVAC installers at a pace that will keep up with the expected growth in demand. Training programs will need to be made widely available to ensure that new graduates and existing installers are able to obtain the needed training and certification. At this stage, it is unclear what installer training may be required, who will provide it, and how it will be funded. One option is to require certification to work with A2L refrigerants. While this is arguably the best approach for ensuring safe installation given the mild flammability, the added burden on HVAC installers could result in too few installers undergoing the required training on installing heat pumps with A2L refrigerants, thus hampering widespread market adoption. To minimize the burden on companies that install HVAC systems, funding to offset the expenses associated with providing training for installers could be made available, perhaps through the Employment Training Panel, a state-run program established to support employers in upgrading the skills of their workers (https://etp.ca.gov/).

In addition to the particular challenges associated with transitioning to A2L refrigerants, the residential heat pump market generally faces several critical barriers, the primary being high installation costs. The initial cost of replacing a gas furnace with a residential heat pump is typically higher than installing a like-for-like replacement gas furnace and central AC.[5] Numerous financial incentives are available to California homeowners through utility programs, the statewide TECH Clean California program, and most recently through the Inflation Reduction Act, though its future is now uncertain. Much of the state funding has been earmarked for low-income customers, providing support worth up to 100 percent of project costs. Financial incentives for higher-income customers help defray the cost of heat pumps but do not typically create parity with gas furnaces. In the future, financial incentives could be used to encourage the adoption of heat pumps that use refrigerants that align with anticipated mandates, rather than current ones.[6]

The cost of operating heat pumps in California poses another barrier to adoption. The high cost of electricity compared to gas in California's investor-owned utility territories means that many customers would face higher operating costs moving from furnaces to heat pumps. The possibility of higher utility bills – and the uncertainty around that outcome – dissuades market adoption. Ensuring equipment efficiency – and favorable electricity rates for heat pump owners – can address the ongoing affordability of heating California homes with heat pumps.

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[5] The median total cost to install a natural gas furnace and 14 SEER Central AC in California is roughly $4,000 (Opinion Dynamics, 2022) and $5,000 to $6,000 (Bender, 2024). Estimates for installing a residential heat pump in California range from $10,000 (Opinion Dynamics, 2022) to $20,000 (TECH Clean California, 2024).

[6] There is precedent for this approach. The Self-Generation Incentive Program encourages the use of heat pump water heaters with "low global warming potential" refrigerants, offering an additional $1,500 incentive compared to standard equipment.

Finally, general awareness of heat pumps is low among California residents. This places an additional burden on installing contractors to deliver basic heat pump education to prospective customers. Workforce training may be required to ensure that HVAC contractors have the knowledge and communication skills required to effectively play this role. Additionally, consumer awareness campaigns, such as the one CAHPP plans, are needed to educate the public about heat pumps (CAHPP, 2025).

Medium-Term Commercialization Pathway

The MPHX has been demonstrated at a 1.5-ton scale in this project. Two pathways for fabrication were explored: (1) 3D printing of polymers using stereolithographic printing of resins, and (2) injection molding and joining. The performance of a 3D-printed version of the MPHX was demonstrated and showed that it can perform on par with commercial metallic heat exchangers. This 3D-printed version was not ideal geometry as it required larger liquid channels, and shorter heat exchanger length due to the fabrication limitations. Injection molding eliminated these issues and led to an improvement in performance, as MPHX test results showed. During this project, the first trial of IM was explored with joining using ultrasonic welding. The mold and joining method did not yield a leak-free unit. However, several lessons were learned, and a new design of the mold and two new joining methods were identified — laser welding, and solvent glueing — that show promise of creating leak-free MPHX liquid plates. Continued development of the IM method is planned as a part of two ongoing projects supported by the Advanced Research Projects Agency-Energy (ARPA-E) and U.S. DOE’s Industrial Efficiency and Decarbonization Office (IEDO). In the ARPA-E project, the MPHX concept will be scaled to a 100-kW demonstration aimed at dry coolers for the datacenter market. In the IEDO project, the MPHX will be explored for low-grade (less than 302°F [150°C]) industrial waste heat recovery opportunities in food processing. These two projects will allow continued development of the MPHX concept and expand the application space.

A patent for the MPHX (U.S. Patent No.: 12,066,197 B2) was issued in August 2024, and a continuation-in-part application was filed detailing several other applications and embodiments of the MPHX device. The patent holders are registered to attend a regional U.S. National Science Foundation's Innovation Corps workshop to engage in customer discovery.

A process-based cost modeling approach was used to estimate the cost of production of 1.5-ton MPHX units using two different methods: SLA 3D printing and injection molding. The latter is a more traditional approach to manufacturing that has been used in the industry for decades. The injection molding route results in a unit cost of $523 for a 1.5-ton MPHX. A commercial water coil (Hi-Velocity WCM-50) of similar size was recently procured for lab testing at a cost of $656. A coil from another manufacturer costs $570 (Precision Coils W102612N 1.65 ton). There are several process optimization scenarios that can be pursued to lower the cost of the MPHX unit such as reduction in the assembly cost of the sealed water plates, which has not been modeled in detail. Another cost reduction strategy is using an adhesive rather than laser welding for forming sealed water plates. Further production-scale cost reductions are to be expected when scaled up. Currently, a single mold insert is being used in the molding process. If one were to use multiple mold inserts in a larger machine, it is likely that the cost per molded part would be reduced. Further refinement will be performed as a part of a follow-up project.

Performance Metrics

Table 9 outlines the target performance metrics goals for this project.

Table 9: Performance Metrics Table

Performance Metric	Benchmark Performance	Current Performance	Low Target Performance	High Target Performance	Evaluation Method	Significance of Metric
HSPF2	7.0	8.1	8.2	9.4	AHRI Standard 210/240–2017 and Appendix M to Subpart B of 10 C.F.R. 430	Demonstrates improved heating efficiency over minimum efficiency heat pumps
SEER2	13.3	16.0	17.1	18.5	AHRI Standard 210/240–2017 and Appendix M to Subpart B of 10 C.F.R. 430	Demonstrates improved cooling efficiency over minimum efficiency heat pumps
Installed cost relative to competing technology	$14,000	$12,900 (-8%)	10% lower than competing technology	15% lower than competing technology	Commercial cost estimate	Demonstrates reduction in cost relative to competing technology (similar efficiency heat pump)
Global warming potential of refrigerant	2088	466	<750	<300	GWP based on refrigerant constituents	Reduces direct emissions related to refrigerant leakage
Secondary loop HX effectiveness at 900 CFM	0.7	0.8-0.95	0.8	0.9	Lab testing of heat exchanger effectiveness	Demonstrates improved heat exchanger performance relative to conventional water-to-air coils used in hydronic systems

HSPF2 Goal

The next-generation heat pump developed for this project had an 8.1 HSPF2, which nearly met the low performance target of 8.2.

SEER2 Goal

The next-generation heat pump developed for this project had a 16.0 SEER2, which was lower than the goal of 17.1.

Affordability Goal

The next-generation heat pump did achieve significant cost reductions due to the novel design, which reduced the production cost by 20 percent to 25 percent. The reduced equipment cost would be passed onto the consumer, making the product more affordable. Installation costs, however, account for most of the overall cost to the consumer, and while this unit is comparable to other heat pumps on the market, there were no cost savings relative to installation.

GWP Goal

The next-generation heat pump developed uses R-454B refrigerant, which has a GWP of 466. This is well below the limit of 750 proposed in the project showing this goal was achieved.

Heat Exchanger Goal

The MPHX heat exchanger developed for this project achieved much higher effectiveness than a commercial coil with an estimated effectiveness of 0.76 to 0.95. A sub-optimal design was tested for this project showing lower performance, but with injection molding strategies the heat exchanger can achieve the goal stated in the project proposal.

CHAPTER 5: Conclusion

Electrifying space conditioning systems is one of the first steps toward meeting California’s ambitious decarbonization goals. This project developed new technology aimed at reducing the cost of heat pumps that use lower GWP refrigerants while also improving performance. The near- and medium-term solutions considered the immediate need for heat pump technology that meets new refrigerant regulations while also advancing technology that would be needed if refrigerant regulations become even more stringent.

This project documented the impact of installing the near-term next-generation heat pump technology in 10 California homes. These installations highlighted some of the challenges of heat pump retrofits including electrical panel capacity limits and electrical upgrades required for the air handler. Notably, the new refrigerant, which is designated as an A2L, mildly flammable fluid, did not represent a significant market barrier for the heat pump from the perspective of installation or user acceptance. The measured performance of the heat pump showed higher efficiency than the baseline systems, yet still resulted in increased utility bills of 3 percent to 27 percent for five out of eight participating homes due to higher electricity use in the winter. Some of these increases may have been a result of changes in occupant behavior (that is, thermostat setpoints) as it was noted that the new variable-speed heat pump was able to achieve more stable temperature conditions in the home, and warmer zone temperatures were observed at some sites in the winter relative to the baseline. While in many cases there was an increase in utility costs, the GHG emissions were 44 percent to 90 percent lower for the heat pump system relative to the natural gas heating systems used in the majority of baseline systems showing significant progress toward decarbonization.

Rheem is preparing to commercialize the new hybrid inverter drive heat pump with R-454B refrigerant but must revise the control algorithm and re-test all models to comply with a recently proposed controls verification procedure for AHRI certification. The controls verification ensures that variable-capacity equipment modulates appropriately when installed and includes a load-based testing procedure. If the controls modifications are successful, it is not expected that these changes would impact the performance reported in this project.

The medium-term solution is aimed at advancing the performance of ATW heat pumps by developing new heat exchanger technology for the hydronic heat exchanger used in air distribution systems. The MPHX water-to-air coil developed in this project was tested and compared to a typical commercial fin-tube coil constructed from copper and aluminum. The construction of the MPHX using SLA 3D printing did not allow testing of the optimal geometry, and the results showed similar performance between the MPHX and the commercial coil. While the performance was similar, this is a major improvement compared to other plastic heat exchanger designs. The next iteration of the MPHX will be manufactured via injection molding, enabling the creation of geometries that are not feasible with 3D printing. Preliminary projections suggest that the injection molded version will likely surpass both the current MPHX and the commercial coil in terms of effectiveness (0.76 to 0.95) and heat transfer rate (2580

W to 4230 W). This translates to a 15 percent to 20 percent increase in effectiveness and capacity for the injection molded MPHX.

An improved heat exchanger design was simulated in the laboratory showing superior effectiveness compared to the commercial coil. The improved heat transfer resulted in 5 percent higher capacity and system efficiency for the ATW heat pump system. While performance was shown to improve, the efficiency of the ATW system was 37 percent lower than the near-term solution heat pump tested for this project. The injection molded versions would significantly improve heat exchanger performance, bringing the efficiency of the ATW system closer to that of the near-term solution heat pump.

Future research should continue to innovate around the development of efficient heat pumps at lower cost. Participants in this project were happy with the comfort provided by the heat pump but had mixed opinions about whether the operating costs were lower than their previous gas furnace. This is compounded by the fact that installation costs were higher than a gas furnace replacement due to electrical upgrades needed to support the heat pump indoor unit. There should also be research to validate the use of heat pumps without supplementary electric resistance heaters in California and encouragement of heat pump manufacturers to offer heat pump indoor units that operate on a standard 120V circuit for these cases. The electric resistance heaters installed for this project required upgrading the electrical service to the indoor unit but were seldom used, suggesting they are not necessary for maintaining comfort. Offering a 120V indoor unit solution would eliminate the installation barriers posed by electrical upgrades for the indoor unit, including potential panel upgrades, and reduce connected load on the grid. Manufacturers would need to continue developing strategies for dealing with defrost cycles without supplementary heating systems to reduce the potential for comfort issues related to this approach.

GLOSSARY AND LIST OF ACRONYMS

Term	Definition
$/kwh	dollars per kilowatt-hour
$/therm	dollars per therm
3D	three-dimensional
A2L	a class of mildly flammable refrigerants with low global warming potential
AC	air conditioning
AFUE	Annual Fuel Utilization Efficiency
AHRI	Air-Conditioning, Heating, and Refrigeration Institute
amp	ampere
ARPA-E	Advanced Research Projects Agency-Energy
ASHRAE	American Society of Heating, Refrigeration, and Air-Conditioning Engineers
ATW	air-to-water
Btu/hr	British thermal units per hour
Btu/W-hr	British thermal units per watt-hour
BUILD	Building Initiative for Low-emissions Development Program
°C	degrees Celsius
C_R	heat capacity ratio
CARB	California Air Resources Board
CAHPP	California Heat Pump Partnership
CEC	California Energy Commission
CFD	computational fluid dynamics
CFM	cubic feet per minute
CFM/ft2	cubic feet per minute per square foot
CFR	Code of Federal Regulations
CO2	carbon dioxide
CO2e/kWh	carbon dioxide equivalent per kilowatt-hour
CO2e/therms	carbon dioxide equivalent per therms
COP	coefficient of performance
CPUP	California Public Utilities Commission
CZ	California climate zones

Term	Definition
EER	energy efficiency ratio
EPIC	Electric Program Investment Charge
°F	degrees Fahrenheit
fenestration U-factor	a measure of how efficiently a window or door assembly (including the glass, frame, and spacers) transfers heat. A lower U-factor indicates better energy efficiency, meaning the window or door is better at preventing heat loss
ft^2	square feet
GHG	greenhouse gas
GWP	global warming potential
HSPF2	Heating Seasonal Performance Factor 2
HVAC	heating, ventilation, and air conditioning
HX	heat exchanger
Hz	hertz
IEDO	Industrial Efficiency and Decarbonization Office (U.S. DOE)
IM	injection-molded
IQR	inter-quartile range
kBtu	thousand British thermal unit
kBtu/hr	thousand British thermal units per hour
kBtu/yr	thousand British thermal units per year
kW	kilowatt
kWh	kilowatt-hour
lb	pound
lb CO2	pounds of carbon dioxide
lbm/min	pounds mass per minute
MPHX	microchannel polymer heat exchanger
NTU	number of transfer units
Pa	pascal
PG&E	Pacific Gas and Electric Company
PSC	permanent split capacitor
psia	pounds per square inch absolute
Re_h	air side Reynolds number
Rheem	Rheem Manufacturing Company

Term	Definition
SEER2	Seasonal Energy Efficiency Ratio2
SHGC	Solar Heat Gain Coefficient
SLA	stereolithography
TDB	temperature dry bulb
TECH	Technology and Equipment for Clean Heating
therms per ft^2	therms per square foot
TOU	time-of-use
TWB	temperature wet bulb
UC Davis	University of California, Davis
U.S. DOE	United States Department of Energy
U.S. EPA	United States Environmental Protection Agency
V	volt
W	watts
WCEC	Western Cooling Efficiency Center

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Project Deliverables

The deliverables for this project are listed below and provide more detail on the findings of this report. These deliverables are available upon request by submitting an email to pubs@energy.ca.gov.

- Development and Lab Testing of Near-Term Solution
    - Memo on Next Generation Heat Pump Development
    - Heat Pump Performance Test Plan
    - Heat Pump Laboratory Test Report
- Development of Pilot-Scale and Engineering-Scale of MPHX
    - Pilot-scale MPHX Fabrication and Mechanical Testing Memo
    - Pilot-scale MPHX Thermal Test Memo
    - Process-based Cost Model Memo
    - MPHX Development and Scale-up Report
- Engineering-Scale System Demonstration
    - Engineering-scale Medium-term Solution Test Plan Memo
    - Engineering-scale Medium-term Solution Testing Report
- Field Demonstrations
    - Field Measurement and Verification Plan
    - Memo on Analysis of Baseline Performance
    - Memo on Field Installation Process and Customer Feedback
    - Memo on Customer Satisfaction Survey Results
    - Field Test Results of Next Generation Heat Pump
- Modeling Impacts to California Ratepayers
    - Next Generation Heat Pump Modeling Results
- Market Barriers and Commercialization Assessment
    - Report on Obstacles and Opportunities on the Path to Market

In addition to the project deliverables, publications related to the work performed for this project are listed below:

Harrington, C., S. Outcault, J. Feng, and A. Frasier. 2024. *Field Evaluation of Affordable Low Global Warming Potential Residential Heat Pumps.* 2024 ACEEE Summer Study on Energy Efficiency in Buildings. Available at https://www.aceee.org/sites/default/files/proceedings/ssb24/pdfs/Field%20Evaluation%20of%20Affordable%20Low%20GWP%20Residential%20Heat%20Pumps.pdf.

Rasouli, E., E. Fricke, and V. Narayanan. 2022. *High Efficiency 3-D Printed Microchannel Polymer Heat Exchangers for Air Conditioning Applications*. *Science and Technology for the Built Environment*, 28: 289-306. Available at https://doi.org/10.1080/23744731.2022.2026693.

ENERGY RESEARCH AND DEVELOPMENT DIVISION

APPENDIX A: Near-Term Solution Laboratory Testing Data

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APPENDIX A:
Near-Term Solution Laboratory Testing Data

Table A-1: WCEC Full Compressor Speed Cooling Test Results

Test ID	Outdoor Dry-Bulb (°F)	Indoor Dry-Bulb (°F)	Indoor Wet-Bulb (°F)	Indoor Air Flow (CFM)	Total Capacity (Btu/hr)	Total Power (W)	EER (Btu/W-hr)
CCR1	65	80	63	767	23122	1347	17.2
CCR2	75	80	63	780	22814	1498	15.2
CCR3	85	80	63	781	21648	1656	13.1
CCR4	95	80	63	775	20434	1830	11.2
CCR5	105	80	63	782	19164	2010	9.5
CCR6	115	80	63	791	18487	2232	8.3
CCR7	65	80	67	776	24242	1338	18.1
CCR8	75	80	67	777	24349	1487	16.4
CCR9	85	80	67	774	23595	1658	14.2
CCR10	95	80	67	780	23308	1750	13.3
CCR11	105	80	67	780	22089	1928	11.5
CCR12	115	80	67	777	19740	2258	8.7
CCR13	65	80	71	789	25371	1324	19.2
CCR14	75	80	71	790	26409	1464	18.0
CCR15	85	80	71	792	25534	1637	15.6
CCR16	95	80	71	790	24023	1813	13.2
CCR17	105	80	71	793	22600	2008	11.3
CCR18	115	80	71	796	21340	2243	9.5

Btu/- hr=British thermal units per watt-hour
Source: UC Davis

Table A-2: Rheem Full Compressor Speed Cooling Test Results

Test ID	Outdoor Dry-Bulb (°F)	Indoor Dry-Bulb (°F)	Indoor Wet-Bulb (°F)	Indoor Air Flow (CFM)	Total Capacity (Btu/hr)	Total Power (W)	EER (Btu/W-hr)
CCR1	65	80	63	773	23038	1427	16.1
CCR2	75	80	63	772	21862	1581	13.8
CCR3	85	80	63	771	20653	1741	11.9
CCR4	95	80	63	775	19468	1915	10.2
CCR5	105	80	63	773	18237	2088	8.7
CCR6	115	80	63	775	17290	2292	7.5
CCR7	65	80	67	773	24945	1426	17.5
CCR8	75	80	67	778	24477	1596	15.3

Test ID	Outdoor Dry-Bulb (°F)	Indoor Dry-Bulb (°F)	Indoor Wet-Bulb (°F)	Indoor Air Flow (CFM)	Total Capacity (Btu/hr)	Total Power (W)	EER (Btu/W-hr)
CCR9	85	80	67	774	22560	1765	12.8
CCR10	95	80	67	776	22034	1955	11.3
CCR11	105	80	67	778	20735	2157	9.6
CCR12	115	80	67	780	19399	2362	8.2
CCR13	65	80	71	780	27247	1403	19.4
CCR14	75	80	71	771	25977	1593	16.3
CCR15	85	80	71	773	25041	1773	14.1
CCR16	95	80	71	772	23713	1970	12.0
CCR17	105	80	71	770	22271	2180	10.2
CCR18	115	80	71	771	20914	2396	8.7

Source: UC Davis

Table A-3: WCEC Part-Load Compressor Speed Cooling Test Results

Test ID	Compressor Speed	Outdoor Dry-Bulb (°F)	Indoor Dry-Bulb (°F)	Indoor Wet-Bulb (°F)	Indoor Air Flow (CFM)	Total Capacity (Btu/hr)	Total Power (W)	EER (Btu/W-hr)
PL1	100%	95	80	67	794	22938	1811	12.7
PL4	70%	95	80	67	565	15102	1110	13.6
PL5	60%	95	80	67	501	12496	905	13.8
PL6	50%	95	80	67	426	10319	733	14.1
PL7	40%	95	80	67	337	7137	567	12.6

Source: UC Davis

Table A-4: Rheem Part-Load Compressor Speed Cooling Test Results

Test ID	Compressor Speed	Outdoor Dry-Bulb (°F)	Indoor Dry-Bulb (°F)	Indoor Wet-Bulb (°F)	Indoor Air Flow (CFM)	Total Capacity (Btu/hr)	Total Power (W)	EER (Btu/ W-hr)
PL1	100%	95	80	67	773	21336	1936	11.0
PL4	70%	95	80	67	552	14737	1171	12.6
PL5	60%	95	80	67	499	12769	962	13.3
PL6	50%	95	80	67	426	10631	776	13.7

Source: UC Davis

Table A-5: WCEC Full Compressor Speed Heating Test Results

Test ID	Outdoor Dry-Bulb (°F)	Outdoor Wet-Bulb (°F)	Indoor Dry-bulb (°F)	Indoor Air Flow (CFM)	Total Capacity (Btu/hr)	Total Power (W)	EER (Btu/W-hr)
HCR14	55	49	70	761	23129	1777	13.0
HCR15	47	43	70	758	21106	1745	12.1

Source: UC Davis

Table A-6: Rheem Full Compressor Speed Heating Test Results

Test ID	Outdoor Dry-Bulb (°F)	Outdoor Wet-Bulb (°F)	Indoor Dry-bulb (°F)	Indoor Air Flow (CFM)	Total Capacity (Btu/hr)	Total Power (W)	EER (Btu/W-hr)
HCR1	65	57	60	774	25896	1731	15.0
HCR2	55	49	60	765	23434	1697	13.8
HCR3	47	43	60	765	21189	1674	12.7
HCR4	35	33	60	769	17250	1612	10.7
HCR5	25	23	60	771	14657	1564	9.4
HCR6	17	15	60	771	12653	1526	8.3
HCR7	65	57	65	774	26117	1818	14.4
HCR8	55	49	65	775	23183	1768	13.1
HCR9	47	43	65	778	20832	1743	12.0
HCR10	40	33	65	766	18506	1651	11.2
HCR11	30	23	65	769	15719	1638	9.6
HCR12	17	15	65	771	12510	1568	8.0
HCR13	65	57	70	775	25487	1882	13.5
HCR14	55	49	70	776	22722	1831	12.4
HCR15	47	43	70	782	20677	1811	11.4
HCR16	40	33	70	777	18683	1768	10.6
HCR17	30	23	70	769	15900	1708	9.3
HCR18	17	15	70	774	12144	1608	7.6

Source: UC Davis

Table A-7: Rheem Part-Load Compressor Heating Test Results

Test ID	Compressor Speed	Outdoor Dry-Bulb (°F)	Outdoor Wet-Bulb (°F)	Indoor Dry-Bulb (°F)	Indoor Air Flow (CFM)	Total Capacity (Btu/hr)	Total Power (W)	EER (Btu/W-hr)
HPL1	100%	47	43	70	821	20533	1797	11.4
HPL3	70%	47	43	70	560	13391	1086	12.3
HPL4	65%	47	43	70	533	12447	990	12.6
HPL5	60%	47	43	70	490	11418	899	12.7
HPL6	50%	47	43	70	410	9438	741	12.7

Source: UC Davis