Residential Multi-Function Heat Pump Laboratory Testing Final Report

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Source: Publicly available Villara AquaThermAire sales brochure

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Disclaimer

Executive Summary

Background

Residential heat pump space-conditioning and water-heating products are more efficient than existing electric resistance and natural gas-combustion options. For retrofit customers considering a heat pump for space conditioning and/or hot water heating, potential requirements for electrical service upgrades can add cost and installation delays (Outcault, et al. 2021).

Residential Multi-Function Heat Pumps (MFHPs) use one efficient compressor and outdoor heat exchanger coil to provide space conditioning and domestic hot water heating. MFHPs can significantly reduce the maximum power requirements compared to the typical separate space conditioning heat pump and standalone heat pump water heater. This means that MFHPs are more likely to fit on existing electrical panels and not require expensive upgrades.

Air-to-air MFHPs use refrigerant to bring thermal energy into and out of the building. Some models, including the MFHP tested in this project, can recover waste heat from space cooling to heat hot water for significant energy savings.

Objectives

This project's objective was to complete laboratory testing of the residential single-speed MFHP equipment. The tests measured its capacity and energy consumption for space cooling, space heating, water heating, waste heat recovery using simultaneous space cooling and water heating, and defrost across a range of outdoor conditions that match California climate zones. The measured results were used to develop performance curves that will be used by future projects to estimate annual energy savings in residential buildings across climate zones.

Methodology: Multi-Function Heat Pump Laboratory Testing

Test

This project tested the residential single-speed MFHP in the University of California, Davis Western Cooling Efficiency Center environmental chambers. The equipment was tested across a range of outdoor conditions, including those that match the federal space conditioning heat pump test standard, AHRI 210.240-2023, and the ASHRAE MFHP test standard, ASHRAE 206-2024 (AHRI 2023, ASHRAE 2024). Additional test conditions were selected to better cover the California climate zones and to provide enough test points to enable regressions to develop accurate performance curves. For water heating operation, an additional set of first-hour rating tests were completed, informed by the Department of Energy's Uniform Test Method for Measuring the Energy Consumption of Water Heaters (U.S. DOE 2024).

Findings

The laboratory test results show that the single-speed MFHP unit capacity and coefficient of performance (COP) or both space heating and space cooling modes match the 3rd party lab test results for AHRI-rating manufacturer-published data for the mass produced split system heat pump before it is modified with additional refrigerant valves to become the MFHP. This alignment suggests that discrepancies observed in previous field test were not inherent issues with the MFHP design but were instead likely due to differences between lab tests and field installation, suboptimal installation, or defective components.

Space Cooling

The measured space cooling capacity and COP of the 4-ton single-speed MFHP in laboratory tests matched the manufacturer's data for the conventional heat pump before being converted into a MFHP, within experimental uncertainty. At 95°F outdoor temperature and 80°F indoor temperature, the space cooling mode had a capacity of 45.3 kBTU/h at a COP of 3.57.

Space Heating

The measured space heating capacity and COP of the MFHP in laboratory tests matched the manufacturer's data for the conventional heat pump, within experimental uncertainty. The space heating mode capacity and COP were consistent with manufacturer-reported data, and the MFHP performed reliably, even at low outdoor temperatures. At 47°F outdoor temperature and 70°F indoor temperature, the space heating mode had a capacity of 49.9 kBTU/h at a COP of 3.48.

Water Heating

As expected for all heat pump water heating equipment, the laboratory test-measured water heating capacity and COP at a given outdoor air temperature decreased as water tank setpoint temperatures rose. At 47°F outdoor temperature and 115°F water tank temperature, the dedicated water heating mode demonstrated a capacity of 36.1 kBTU/h at a COP of 2.67. The water heating mode had a higher COP than electric resistance water heating, except at the extreme high water tank setpoint temperature of 135°F paired with very low outdoor temperatures below 17°F. The MFHP achieved a first-hour rating of 82.0 gallons, with calculations based on water draws meeting test standards and accounting for delays in valve actuation and test conditions. The team noted that the refrigerant heat exchanger for water heating was not adequate for the compressor size. Increasing the refrigerant to water heat exchanger heat transfer capability is expected to increase water heating capacity, increase COP, and increase first hour rating.

Simultaneous Space Cooling and Water Heating

In simultaneous space cooling and water heating mode, the COP generally outperformed the separate space cooling and water heating modes. The simultaneous mode saved an average of 38% of electrical energy compared to performing space cooling and water heating separately. Even under extreme conditions with water tank setpoint temperatures above 135°F and outdoor temperatures below 75°F, the COP was higher for simultaneous mode than two separate modes. Increasing the refrigerant to water heat exchanger size or heat transfer capability is also expected to increase performance for simultaneous mode.

Defrost

Defrost operation for the MFHP uses the refrigerant compressor to move heat from the hot water tank to the outdoor coil to melt accumulated frost. Defrost of the outdoor coil was completed in one to three minutes with a peak compressor power of three kilowatts and a system peak power of three kilowatts. Compared to typical single-speed split-system heat pumps, this MFHP completes defrost faster, at lower peak compressor power, and much lower system power since it does not use resistance heaters. A much lower system defrost maximum power than typical single-speed heat pump systems makes the MFHP more likely to fit on existing electrical panels without needing an upgrade. An unexpected lock-out period following defrost mode was identified, which the manufacturer is fixing in a software update.

Outreach

The project team has contacted the California Energy Commission CBECC-Res developers to share the MFHP performance curves and promote their use in code compliance and efficiency credit calculations.

The project team has reached out to the California Technical Forum (CalTF) and to San Diego Gas and Electric (SDG&E), the California statewide lead for heating, ventilation, and air conditioning (HVAC) efficiency programs, to start the measure development process.

Recommendations and Next Steps

The project team's next steps include:

- Continue to engage with the California HVAC program administrator, SDG&E, to determine what standards and requirements the equipment needs to meet to be included in the program as a new efficiency measure. Follow up with the MFHP manufacturer to support meeting those requirements.
- Continue to engage with the CalTF to present the MFHP lab and field test results to the CalTF Deemed Initiative Subcommittee to begin the measure development process.
- Continue to recruit and prepare the team to do the next step measure development. The Western Cooling Efficiency Center will contribute to, or lead, the measure development efforts to ensure the performance curves can represent equipment efficiency in the required modelling tools to predict energy savings.
- Continue to engage with EnergyPlus and California Energy Commission CBECC-Res software developers to promote the use of the performance curves for cost-benefit analysis and for code compliance.

Abbreviations and Acronyms

Acronym	Meaning
A	Amps
AC	Air Conditioner
AESC	Alternative Energy Systems Consulting, Inc.
AHR	Air Conditioning, Heating, and Refrigeration
AHRI	Air Conditioning, Heating, and Refrigeration Institute
AHU	Air Handler Unit
ASHRAE	American Society of Heating, Refrigerating and Air-Conditioning Engineers
ASIHP	Air-Source Integrated Heat Pump
CalFlexHub	California Load Flexibility Hub
CalTF	California Technical Forum
CEC	California Energy Commission
COP	Coefficient of Performance
DAC	Disadvantaged Communities
DB	Dry Bulb Temperature
DEF	Defrost
DHW	Domestic Hot Water
DOE	U.S. Department of Energy
DP	Dew Point Temperature
EE	Energy Efficiency
EIR	Electric Input Ratio/Energy input Ratio
EPRI	Electric Power Research Institute

Acronym	Meaning
ESP	External Static Pressure
ET	Emerging Technology
FHR	First Hour Rating
GAL	Gallon
GPM	Gallons Per Minute
HDPE	High-Density Polyethylene
HP	Heat Pump
HPWH	Heat Pump Water Heater
HTR	Hard-to-Reach
HVAC	Heating, Ventilation, and Air Conditioning
IA	Indoor Air
IEA	International Energy Agency
IOUs	Investor-Owned Utilities
kBTU/h	Kilo-BTU per Hour
kW	Kilowatt
kWh	Kilowatt-hour
LL	Liquid Refrigerant Line
MFHP	Residential Multi-Function Heat Pump
NEEA	Northwest Energy Efficiency Alliance
NEMI	National Energy Management Institute
OA	Outdoor Air
ODU	Outdoor Unit

Acronym	Meaning
OEM	Original Equipment Manufacturer
Pa	Pascal
PG&E	Pacific Gas & Electric
PID	Proportional-Integral-Derivative
PLR	Part Load Fraction
RA	Return Air
RH	Relative Humidity
RT	Refrigeration Ton
SA	Supply Air
SC	Space Cooling
SSC	Simultaneous Space Cooling
SCE	Southern California Edison
SDG&E	San Diego Gas & Electric
SEER	Seasonal Energy Efficiency Ratio
SH	Space Heating
SHR	Specific Heat Ratio
SIM	Simultaneous Space Cooling and Water Heating
SMACNA	Sheet Metal and Air Conditioning Contractors' National Association
SS	Steady State
SWH	Simultaneous Water Heating
TVA	Tennessee Valley Authority
UC	University of California

Acronym	Meaning
U.S.	United States
VL	Vapor Refrigerant Line
WB	Wet Bulb Temperature
WCEC	UC Davis Western Cooling Efficiency Center
WH	Water Heating
WT	Water Tank
WTH	Water Tank Storage Capacity

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Final Report

Introduction

Residential heat pump (HP) space-conditioning and water-heating products are more efficient than existing electric resistance and natural gas-combustion options. For retrofit customers considering HPs for space conditioning and/or hot water heating, requirements for electrical service upgrades add cost and installation delays (Outcault, et al. 2021). Around 30 to 50% of all homes in California are expected to need electrical-service-panel upgrades to fully electrify (Efficiency First California 2020, C. Merski 2021, Murphy 2022, Lindsey 2023, Zhao 2022). The cost of such an upgrade in California is typically around $5,000, but can range from $2,000 to $30,000. This is potentially a prohibitive additional cost for electrification retrofits.

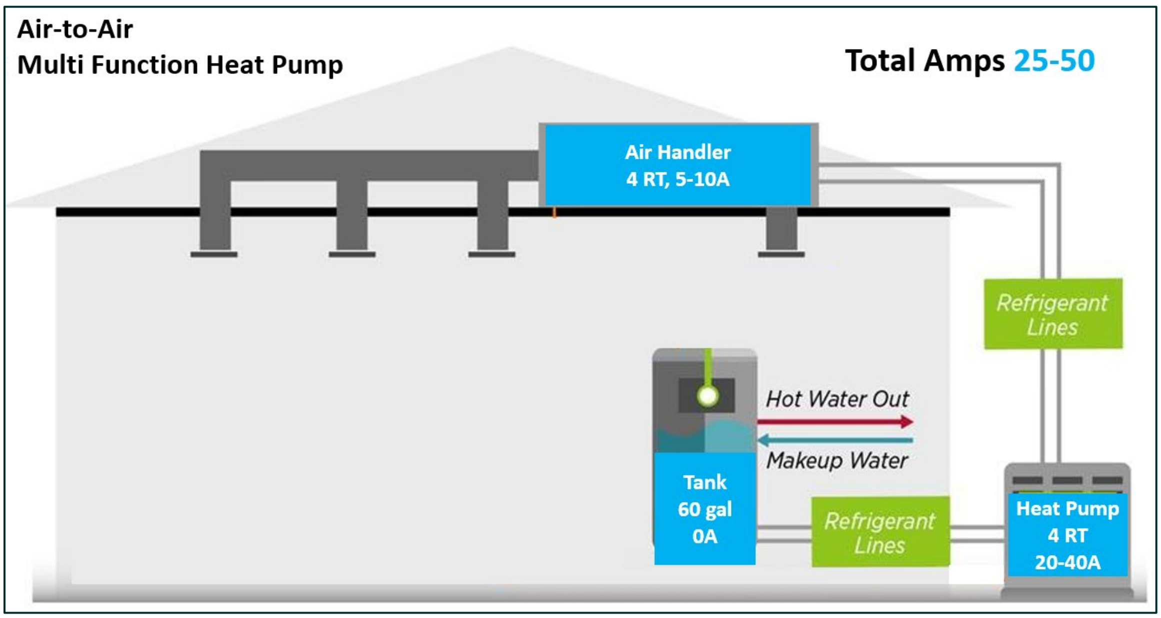

Figure 1. Air-to-air MFHP system diagram showing the outdoor unit and refrigerant lines serving both the air handler and the indoor hot water tank.

Source: Adapted from original image provided by MFHP manufacturer

Residential multi-function heat pumps (MFHPs) use one efficient compressor and outdoor heat-exchanger coil to provide space cooling, space heating, and domestic hot water (DHW) heating. These systems offer many energy-efficiency (EE) benefits. Air-to-air versions of MFHPs use refrigerant to provide heating and cooling services. They have the potential to eliminate the need for electric-resistance backup heaters, reducing the maximum power requirements for full-size capacity systems. For retrofits in buildings with existing air conditioning (AC), this means that full-size capacity air-to-air MFHPs can use existing AC electrical circuits without modification. Air-to-air MFHPs will have lower peak power consumption compared to separate space conditioning and standalone heat pump water heater (HPWH) equipment, so they are less likely to trigger the need for a service-breaker-panel or service-wire upgrade (Outcault, et al. 2021, Pena, et al. 2022). This air-to-air MFHP technology allows a full-capacity HP that matches the building heat demand in most California climates to use an existing split-system AC electrical circuit.

Historically, to avoid the need for electrical service panel upgrades, HPs could be undersized. Undersizing HPs is not recommended because they will not be able to meet the peak loads and may use electric-resistance strip heaters for auxiliary heating, which reduces efficiency and increases energy consumption. Oversizing of single-speed HP equipment causes short cycling with reduced efficiency and increased energy consumption.

Disadvantaged communities (DACs) and hard-to-reach (HTR) customers are more likely to live in older, single-family homes and apartment buildings with smaller electrical service panels (30-100 amps), smaller-gauge building electrical distribution wires, and smaller-capacity utility step-down transformers (Lindsey 2023, Pena, et al. 2022). This means that DAC and HTR customers are more likely to need electrical service upgrades, resulting in more costly projects that present a large barrier to electrification. MFHPs have the potential to significantly reduce electrification costs and installation times, particularly for these customers.

Multi-Function Heat Pump Products

The University of California (UC), Davis Western Cooling Efficiency Center (WCEC) previously completed a technical market characterization air-to-air MFHP product search in 2022 to identify MFHP products commercially available in the United States (U.S.) and globally (Vernon, Residential Multi-Function Heat Pumps: Product Search 2022). The Villara AquaThermAire is the only air-to-air MFHP commercially available in California that the project team was able to find as of November 2023. Panasonic offers an air-to-air MFHP product in southern Europe, but there is no announced date for offering the product in the U.S. market. In February 2023, at the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) Winter Conference's Air Conditioning, Heating, and Refrigeration (AHR) Expo, both LG and Samsung announced plans to offer a residential air-to-air MFHP product in the U.S. in 2023. A year later, in the 2024 AHR expo, the companies showcased their air-to-air MFHP products: LG (Multi V S Heat Recovery + Hydro Kit) and Samsung (DVM S Eco Heat Recovery) at their respective booths. Although both air-to-air MFHP products are now commercially available in the U.S., neither of the companies is providing the water heating and storage tank necessary for DHW production. As of April 2024, the Villara AquaThermAire is the only complete MFHP product with all equipment and controls commercially available in California, and it can provide heat recovery during simultaneous space cooling and water heating.

Laboratory Testing

This project tests the efficiency and capacity performance of a commercially available air-to-air MFHP in the WCEC environmental chambers across a range of outdoor air conditions to match California climate zones for space heating, space cooling, water heating, and for simultaneous space cooling with heat recovery water heating.

This Final Report details the laboratory testing methods, test plan, and sensors used. It describes how the test conditions were selected to match California climate zones and a comparison to the various test standards that apply to these types of equipment. The report also includes feedback from stakeholder engagement efforts and describes how this feedback has been incorporated into the project.

The lab test results were used to develop equipment performance curves for use with EnergyPlus and CBECC-Res to estimate energy savings in residential buildings. This project will enable future projects to estimate energy savings as part of measure development for custom and deemed utility efficiency programs. This project, and the future emerging-technology projects that it supports, will help to advance the residential air-to-air MFHP technology towards becoming a new measure in utility EE programs. Future field demonstrations will verify installation cost savings, real-world energy savings, and customer satisfaction, including among DAC and HTR customers.

Background

Residential MFHPs use one efficient compressor and outdoor heat exchanger coil to provide space conditioning and domestic hot water heating. Air-to-air MFHPs have several operating modes based on the heat exchangers used as the evaporator and condenser. The operational modes along with their respective evaporator and condenser combinations are listed in Table 1 below.

Table 1. MFHP manufacturer MFHP operational modes and respective evaporator and condenser combinations.

Mode	Evaporator	Condenser
Space Cooling (SC)	Indoor Coil	Outdoor Coil
Space Heating (SH)	Outdoor Coil	Indoor Coil
Water Heating (WH)	Outdoor Coil	DHW Tank Refrigerant-to-Water Coil
Simultaneous Space Cooling and Water Heating (SIM)	Indoor Coil	DHW Tank Refrigerant-to-Water Coil
Defrost (DEF)	DHW Tank Refrigerant-to-Water Coil	Outdoor Coil

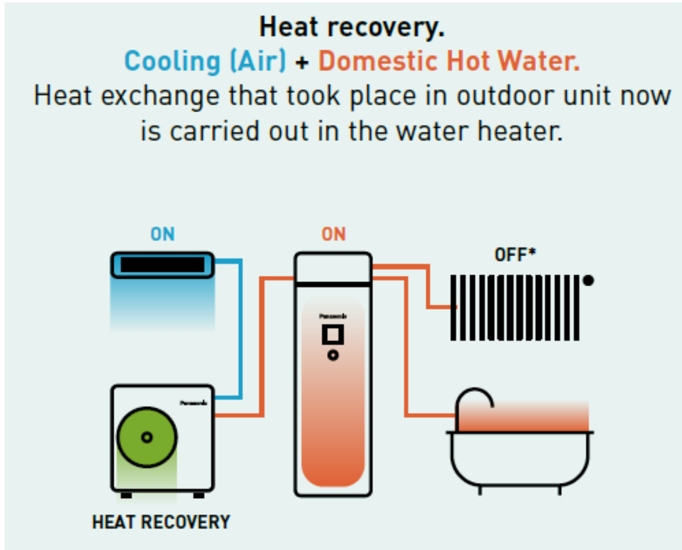

Figure 2. Simultaneous space cooling and domestic hot water heating waste heat recovery mode for an air-to-air MFHP system, showing the outdoor unit and refrigerant lines serving both the air handler and the indoor hot water tank.

Source: Publicly available Panasonic Aquarea EcoFleX sales brochure

WCEC recently completed a field test of a prototype version of the Villara AquaThermAire MFHP in a Pacific Gas & Electric (PG&E)-funded emerging technologies (ET) project in collaboration with Frontier Energy (Chally and Haile, Field Assessment of Residential Three Function Heat Pump Performance 2024).The completed field test of the prototype version of the MFHP showed good performance. Some of the observed issues in the first field demonstration include wiring defects that kept the outdoor unit (ODU) fan and indoor air handler unit (AHU) fan running. This made the DEF operation take longer than expected and resulted in cool-air delivery because no hot refrigerant gas was provided to the indoor coil while the AHU fan was running, which resulted in occupant discomfort. Villara resolved these issues by fixing the wiring defect and modifying the MFHP control board firmware. The field-measured coefficient of performance (COP) of the MFHP was found to be 22% lower in SC mode and about 15% lower in SH mode than the ideal original equipment manufacturer (OEM)-provided data of the HP before modification to MFHP. These differences were attributed to the fact that the OEM-provided data were from laboratory study, whereas the field study could have introduced uncertainties, such as varying return air conditions, longer refrigerant lines, different supply fan static pressure, and dynamic operation. Furthermore, at the time of the field-testing report, the MFHP manufacturer was still conducting refrigerant charge optimization studies and it is probable that the unit was not optimally charged. Currently, the manufacturer has found a reasonable trade-off between optimum refrigerant charge across all the operating modes.

MFHP SIM mode utilizes waste heat from space cooling to heat hot water and provides both services 36% more efficiently than using a separate SC and WH cycle, each exchanging heat with the outdoors. The results of the field test have also been presented at the International Energy Agency (IEA) Heat Pump Conference, May 2023 (Chally, Haile and Chakraborty, et al. 2023).

WCEC has two California Energy Commission (CEC)-funded projects in progress with the California Load Flexibility Hub (CalFlexHub), in which the center is developing load-shifting controls for HPWHs and for MFHPs. WCEC is developing control strategies to enable MFHPs to deploy SIM mode more often using multiple strategies. As part of the CEC-funded CalFlexHub project focused on the MFHP, WCEC continues to monitor the original field site, which is showing good performance, and is expanding to two additional field test sites in a central valley multi-family building. One control issue that was observed in both field sites was the overheating of the water tank (WT) beyond the setpoint in the T thermostat. Following a software upgrade, the Aquastat lost connection to the internet and the MFHP system got stuck in WH mode over the entire weekend, reaching very high temperatures. This was finally fixed by service technicians replacing the Aquastat and restarting the entire system. Since the event, the WCEC team requested a high-water temperature cut-off switch, which was installed in the WT to avoid the risk of scalding injuries.

WCEC has completed a study of HP market adoption that identified barriers that prevent residential homes and apartments from replacing broken AC equipment with HPs. One of the largest barriers to HP adoption is the need to upgrade the existing home electrical infrastructure: electrical circuit, service panel, and/or service wire and step-down transformer. These electrical upgrades are expensive and cause long delays in system installations. Very few owners or residents will choose to pay more and not have AC for an extended period to switch to a HP, so most chose lower-cost and faster-turnaround replacement of the AC.

A recent survey by the National Association of Home Builders found that the average age of a home in the U.S. is 39 years (Zhao 2022). Older homes have smaller electrical service panels because, at the time they were built, demand for electric service rarely exceeded 100 amps. A study by energy research firm Pecan Street estimates that more than 55% of all U.S. single family home electrical panels would need upgrades to allow full electrification (C. Merski 2021). Other electric panel capacity surveys and electrification capacity requirement analyses estimate closer to 30% of residential homes are expected to need electrical service panel upgrades to fully electrify (Efficiency First California 2020, Murphy 2022, Zhao 2022, Lindsey 2023). The cost of an electrical service panel upgrade in California is typically around $5,000, but can range from $2,000 to $30,000, representing a significant level of uncertainty and potentially a cost-prohibitive additional cost for electrification retrofits (Lindsey 2023, Pena, et al. 2022).

A review of publicly available equipment specifications compared the energy requirements for AC and HP equipment, based on systems with equivalent refrigeration capacities and Seasonal Energy Efficiency Ratio (SEER) ratings. The review shows that a typical three-ton capacity, 16-SEER central AC requires a 20–30-amp breaker for the ODU and a 5–10-amp breaker for the AHU, for a total of 25--40 amps. A typical three-ton capacity, 16-SEER central HP has similar requirements to the AC and adds a requirement for the backup electric resistance heaters (strip heat) of 30–45 amps, for a total of 55–85 amps. A typical stand-alone HPWH with a 50-gallon tank requires 30 amps. The typical combination of separate HPs for space conditioning and for hot water heating would require 85–115 amps. MFHPs do not require backup electric resistance heaters so they would require the same electrical service capacity as a comparable AC. They also do not need a separate electrical circuit for hot water heating. As a result, both space conditioning and water heating would require a total of 25–40 amps, a reduction of 60–75 amps compared to the typical equipment.

DAC and HTR customers are more likely to live in older single-family homes and apartment buildings with smaller electrical service panels (30–80 amps), smaller-gauge building electrical distribution wires, and smaller-capacity utility step down transformers. A recent study by the Electric Power Research Institute (EPRI) found that 39% of single-family detached houses built before 1960 have electrical breaker panels of 100 amps or less throughout the U.S. (Lindsey 2023). This means that DAC and HTR customers are more likely to need electrical service upgrades and, therefore, have a larger barrier to electrification. MFHPs have the potential to significantly reduce electrification costs and installation times particularly for DAC and HTR customers.

Some past work in combination HP space heating and water heating used equipment that was not capable of providing space cooling, such as CO2-refrigerant HPs (Eklund and Stephens 2018). These types of products require both a central heating HP and a central cooling AC. These studies show good energy savings but have high equipment and installation costs and face barriers to adoption from installers and customers.

The most relevant previous work in air-to-air MFHPs that provide space heating, space cooling, and water heating used a prototype cobbled together from a ductless HP ODU connected to an AHU and an adapted standalone HPWH tank with a hot water flow-through design and integrated refrigerant-to-water heat exchanger (Energy 350 2015). This prototype used backup electric resistance heaters in the WT and would require them to avoid blowing cold air during defrost cycles so that it would still have high peak power consumption and require similar electrical service upgrades as typical separate HP systems.

There are two providers of custom-engineered residential air-to-water MFHPs based on hydronic thermal energy distribution in Northern California called Harvest Thermal and Stow Energy (Harvest Thermal 2024, Stow Energy 2022). Harvest Thermal uses a separate AC for cooling, so it is expected to be significantly more expensive. Both companies use air-to-water HPs and connect them to AHUs with hydronic coils and one or more WTs. Because most existing residential buildings in California use direct expansion refrigerant systems, retrofits changing to hydronic systems are expected to be more expensive and to take longer to install. These systems are expected to have premium product pricing and to have higher installation costs making them a less than-ideal fit for DACs.

Objectives

The primary purpose of this laboratory testing project is to measure the energy performance of the MFHP in each operating mode across a range of outdoor conditions and produce equipment performance curves for future energy simulations. The steps of this project are:

- Develop a test plan to measure equipment performance of this commercially available air-to-air MFHP across a range of outdoor air conditions to match California climate zones for space heating, space cooling, water heating, and for simultaneous space cooling with heat recovery water heating, informed by ANSI/ASHRAE Standard 206-2024
- Test the efficiency and capacity performance of this commercially available air-to-air MFHP for space heating, space cooling, water heating, and for simultaneous space cooling with heat recovery water heating

- Use lab test results to develop equipment-performance curves for use with EnergyPlus and CBECC-Res to estimate energy savings in residential buildings
- Engage in stakeholder outreach to ensure that performance curves are compatible with EnergyPlus energy-simulation tools and with code-compliance tools including CBECC-Res
- Disseminate the Final Report, including performance results, equipment performance curves for use in EnergyPlus and CBECC-Res to the target audience. The target audience includes CEC CBECC-Res developers, EnergyPlus developers, HVAC equipment performance curve repositories, investor-owned utilities (IOUs) and groups that will use the performance curves developed in this project to make energy savings estimates for measure package(s), including UC Davis WCEC, CalNEXT partners, IOU staff, and relevant engineering consulting firms, as well as other stakeholders including Heating Ventilation and Air Conditioning (HVAC) manufacturers (Villara, Panasonic, Samsung, LG, Mitsubishi), and emerging technologies groups and researchers from other states.

Methodology & Approach

The core expertise of the WCEC is using environmental chambers and data acquisition infrastructure to measure HVAC equipment performance. WCEC staff and facilities have deep expertise and history measuring performance of vapor compression HVAC and WH equipment.

Test Plan

WCEC engineers developed the SH and SC test plan informed by the MFHP ASHRAE test standard 206 (Table 2), and informed by the Air Conditioning, Heating, and Refrigeration Institute (AHRI) standard 210/240 for the indoor conditions (ASHRAE 2024, AHRI 2023). Additional outdoor air (OA) conditions and indoor air (IA) conditions beyond those in the test standards were selected to better match California climates and to enable better performance curve generation. The selected test conditions are listed in Table 3 with air condition dry bulb temperature (DB) and wet bulb temperature (WB).

The ASHRAE MFHP standard calls for SC test points at 95DB/75WB and 82DB/65WB. Considering this, AHRI test points were chosen close to the ASHRAE conditions, 95DB/75WB and 85DB/72WB. To get performance curves accurate in California's hot summer climates, a third OA condition of 105DB/80WB was added. These selected outdoor conditions cover 98% of the California population based on 1% hottest design day data from Title 24 2022 and the 2022 American Community Survey (ACS).

Testing used the ASHRAE MFHP-stated cooling indoor condition of 80DB/67WB and an additional Tennessee Valley Authority (TVA) test indoor condition of 75DB/63WB that more closely matches residential return air conditions for typical California thermostat setpoints. To accurately predict performance in California's dry climate regions, where IA can be lower humidity and only sensible cooling occurs, an additional dry IA condition was added at 80DB/57WB.

A similar methodology for selecting SH points was used. The OA conditions selected are 47DB/43WB, 37DB/34WB, and 17DB/15WB. These three OA conditions match the ASHRAE and AHRI standards, with the middle point (37DB/34WB) likely leading to frost build-up followed by a defrost cycle. This test point enabled characterizing the DEF mode logic, operation and impact on the equipment performance. For the IA conditions, the ASHRAE-stated condition of 70DB/60WB was used. An additional indoor condition of 75DB/63WB was also selected to provide more data for various indoor home conditions performance map. This indoor condition (75DB/63WB) was chosen as it was the only common point between SH and SC according to the OEM datasheets. This should enable clearer visibility into differences of performance between operating conditions when using the different MFHP modes. The OA conditions in SH and SC tests cover a broad range and closely match the boundaries of the California climate zone conditions. These OA and IA conditions were simulated by the environmental chambers to measure equipment performance.

WH test conditions were based on the ASHRAE standard 206, providing two OA points 67DB/53.5WB and 47DB/43WB. WH operation by the MFHP is required by a household throughout the year across the entire range of outdoor conditions. The WCEC test plan was expanded to cover up to the hottest OA condition (105DB/77WB) and down to the lowest OA condition (17DB/15WB). Although it is expected that the MFHP unit would not be able to provide adequate WH at such low temperatures, the limits of the compressor were pushed in these low-ambient tests. WH tests included a water heating cycle starting from the minimum 100°F and heating to the 140°F setpoint to capture the efficiency degradation at the higher tank temperatures for broad operating range. Separate from the performance map, a first-hour rating (FHR) test was performed with 3 GPM water draws for the dedicated WH mode to compare with other water heaters.

The SIM mode, where the MFHP recovers heat from the indoor space and puts it into the WT, was also tested. The SIM mode performance was characterized for different indoor conditions and WT setpoint temperature conditions. Although there is no heat transfer with the outdoors in this mode and outdoor conditions were set to 95DB for all SIM tests. Similar to WH tests, the SIM-mode tests were analyzed for a minimum WT temperature of 100°F heated to a maximum 140°F setpoint.

Table 2. Test points from MFHP ASHRAE test standard 206.

MFHP ASHRAE Standard 206 Tests	Outdoor Air Conditions		Indoor Air Conditions	
Temperature (°F)	**Dry Bulb**	**Wet Bulb**	**Dry Bulb**	**Wet Bulb**
Space Cooling	95	75	80	67
	82	65	80	67
Space Heating	47	43	70	60
	17	15	70	60
Defrost	35	33	70	60

MFHP ASHRAE Standard 206 Tests	Outdoor Air Conditions		Indoor Air Conditions	
Water Heating	67	53.5		
	47	43		
Simultaneous Space Cooling + Water Heating (Heat Recovery)	67	53.5	80	67
	95	75	80	67

Source: ANSI/ASHRAE Standard 206-2024

Table 3. Test plan for single-speed MFHP testing in the WCEC environmental chambers

Test #	Outdoor Air Conditions		Indoor Air Conditions	
Temperature (°F)	Dry Bulb	Wet Bulb	Dry Bulb	Wet Bulb
A.C.1	85	72	80	57
A.C.2	95	75	80	57
A.C.3	105	77	80	57
A.C.4	85	72	75	63
A.C.5	95	75	75	63
A.C.6	105	77	75	63
A.C.7	85	72	80	67
A.C.8	95	75	80	67
A.C.9	105	77	80	67
A.H.1	67	53.5	70	60
A.H.2	47	43	70	60
A.H.3 (defrost)	37	34	70	60

Test #	Outdoor Air Conditions		Indoor Air Conditions	
A.H.4	17	15	70	60
A.H.5	67	53.5	75	63
A.H.6	47	43	75	63
A.H.7 (defrost)	37	34	75	63
A.H.8	17	15	75	63
A.WH.1	105	77		
A.WH.2	85	72		
A.WH.3	67	53.5		
A.WH.4	47	43		
A.WH.5 (defrost)	37	34		
A.WH.6	17	15		
A.SIM.1			80	67
A.SIM.2			80	57
A.SIM.3			75	63

Laboratory Testing Methods

Setup

The MFHP equipment was purchased, delivered, set up in the environmental chambers, and commissioned. The necessary sensors were identified, purchased, and installed on the equipment.

The WCEC laboratory is equipped with adjoining environmental chambers – an outdoor chamber and an indoor chamber – with independent ducting networks and controls to hold each chamber at different environmental conditions for HVAC testing. The AHU and ODU were installed in the outdoor chamber to mimic the environmental effects of being in an unconditioned space, while the WT was installed in the indoor chamber to mimic the environmental effects of being placed in a conditioned space. Return air (RA) and supply air (SA) from the AHU were ducted to the indoor chamber using insulated ducts. Copper refrigerant lines, return and supply water hoses, and miscellaneous instrumentation wiring were run from the outdoor chamber to the indoor chamber via a through-hole that was sealed and insulated to mitigate the influence of one chamber's conditioning on the other.

The unmodified ODU was instrumented with a pressure transducer array in the unit interior to measure cavity pressure. Initial startup of the unit assessed the unrestricted air flow through the ODU (running with open chamber doors and no restriction on the unit outlet) to establish baseline cavity pressure values, which represented the unit's free flow as generated by the integral fan.

To provide insight into coil frosting behavior, an initial investigation of the effects of coil occlusion was made by monitoring cavity pressure values while progressively blocking the ODU outer surface. Additionally, a bell-mouthed hood was used to extract air from the outdoor chamber directly above the ODU to allow control over recirculation of the outdoor air. The hood was placed on 18" standoffs to ensure the outdoor chamber's circulation fans would have an insignificant effect on the ODU's cavity pressure values during operation and to maintain free flow through the unit, regardless of chamber fan settings.

Pressures across the AHU were monitored and used to establish appropriate indoor chamber "booster fan" settings to maintain target operating external static pressure (ESP) according to testing standards (AHRI 2023). Pressurized duct leakage tests were performed on the AHU, indoor chamber, and intermediate ducting to quantify and correct for leakage between environmental chambers and conditioned air.

Figure 3. MFHP ODU and AHU being installed and instrumented in the outdoor chamber.

Figure 4. AHU instrumented and ducted (left). MFHP ODU instrumented and exhaust duct connected with a stand-off (right).

Figure 5. WT of the MFHP manufacturer system installed in the WCEC indoor chamber (left). WT refrigerant-to-water and water-to-water heat exchangers with temperature sensors (right).

Testing

For each space conditioning test, the team ran the MFHP until they observed a 30-minute duration of steady state (SS) operation. For the WH tests, the team held the environmental chambers at steady ambient conditions for the duration of a full water heating cycle.

Two chamber “booster fans” and actuated dampers allow controlled chamber testing of HVAC equipment to make up for the flow resistance caused by the chamber conditioning equipment and to achieve the expected pressure drops of real-world building installations. The temperatures of the chambers were controlled by water-to-air heat exchanger coils modulated by proportional-integral-derivative (PID)-controlled valves and a supplemental walk-in freezer unit for near and below freezing conditions. Humidity was controlled by PID-controlled dampers that send part of the air flow through a dryer and the other part through a humidifier while maintaining a constant pressure drop.

To control ESP for the MFHP, the team adjusted the indoor and outdoor chamber “booster fans” and actuated dampers to match a determined setting based on supply air flow rate, using a fan affinity law square exponential relationship. The outdoor chamber “booster fan” and actuated damper settings were also used to maintain a minimum positive outdoor chamber pressure. The bell-mouthed hood that was added to the equipment to aid in air recirculation was positioned in such a way as to avoid any suction on the ODU.

Space Cooling (SC)

For SC mode testing, the team ran tests at the environmental conditions selected in the test plan. The test plan grid combined three OA conditions and three IA conditions (dry bulb and wet bulb temperatures). In total, 9 different SC tests were run for the MFHP.

Space Heating (SH)

For SH mode testing, the team ran tests at the environmental conditions selected in the test plan. The test plan grid combined four OA conditions and two IA conditions. In total, 8 different SH tests were run for the MFHP. SH runs often used an open-loop configuration on the indoor chamber to maintain SS operation, given that the indoor chamber conditioning equipment has less cooling capacity. Chamber "booster fans" and dampers were adjusted at the beginning of each test to compensate for any changes in conditions due to this configuration.

Water Heating (WH)

For WH mode testing, the team ran tests at the environmental conditions selected in the test plan. However, less focus was placed on controlling the indoor chamber conditions, since the primary conditions affecting MFHP operation during the WH calls are WT temperature and OA conditions. Since no space conditioning was happening simultaneously and the tank is well insulated, there was little concern for the effects of slight deviations in the indoor chamber conditions on MFHP performance during WH tests.

The WT was preheated from about 80°F to 100°F, then a full WH cycle was observed from 100°F until the Aquastat terminated the heating call when it reached its 140°F control setpoint temperature. WH presented additional challenges, as the range of WT temperatures resulted in a wide variance of compressor capacity and a varied rate of heat exchange and refrigerant condensation. These tests featured considerable dynamics that posed challenges to maintaining SS conditions. To help maintain outdoor environmental conditions (dew point, in particular) water draws were used to hold WT temperatures steady while environmental conditions stabilized. The team initiated a draw at the start of each WH test when the WT temperature neared the 100°F starting point, allowing the chamber conditions to stabilize before proceeding with the run. Subsequent draws were used on an as-needed basis if chamber conditions drifted out of range. Water draws were used to bring the WT temperature back below the point at which SS conditioning was lost, the conditioning controls reestablished, and the draw was terminated to continue the WH run.

Simultaneous Space Cooling and Water Heating Mode (SIM)

For SIM mode testing, the team ran tests at the environmental conditions selected in the test plan. SIM mode tests required an emphasis on maintaining the IA conditions due to SC requirements but were less demanding regarding OA conditions, since heat transport primarily occurred between the indoor conditioned space and the WT. The team observed MFHP behavior over a full WH cycle from 100-140°F, while using the MFHP's SC capacity in conjunction with the environmental chambers' conditioning equipment to maintain SS environmental conditions.

SIM mode operation included similar dynamic behaviors to WH, again due to the wide range of WT temperatures observed across each test and were run in the same manner including water draws to reestablish environmental conditions when they drifted out of bounds.

Defrost (DEF)

DEF events occurred during SH and WH tests. These events were evident by increasing ODU cavity pressure values, indicating an accumulation of frost occluding the condenser coils. The DEF events occasionally interrupted these tests since the timing of the DEFs would overlap with the SS test segment and destabilize chamber conditions. It was possible to modify DEF timing or to manually initiate a DEF event via the unit control settings on the MFHP control board, though it was difficult to uniformly time the DEF events, as these controls did not seem to be completely reliable. Initially, the DEF timer on the unit was set to 60-minutes. However, not much frost formation was observed at this setting. For impact characterization, a good amount of frost formation is required; thus, the DEF timer on the unit was increased to 120-minutes to allow more time for frost to accumulate on the coils before the unit initiated a DEF cycle.

The outdoor chamber “booster fan” settings were used to control cooling of 37DB tests but were insufficient for conditioning temperatures below this point due to glycol loop limitations. The coldest DEF tests were run with the outdoor chamber isolated from its ducting loop, with supplemental cooling from a walk-in freezer unit set about 5°F above target DB. With this strategy, the cooling was provided primarily by recirculation from the MFHP ODU. This prevented the hysteresis loop of the walk-in freezer unit from interfering with the SS environmental test conditions, and also prevented chamber temperatures from getting too out of control during MFHP DEFs and shutdowns.

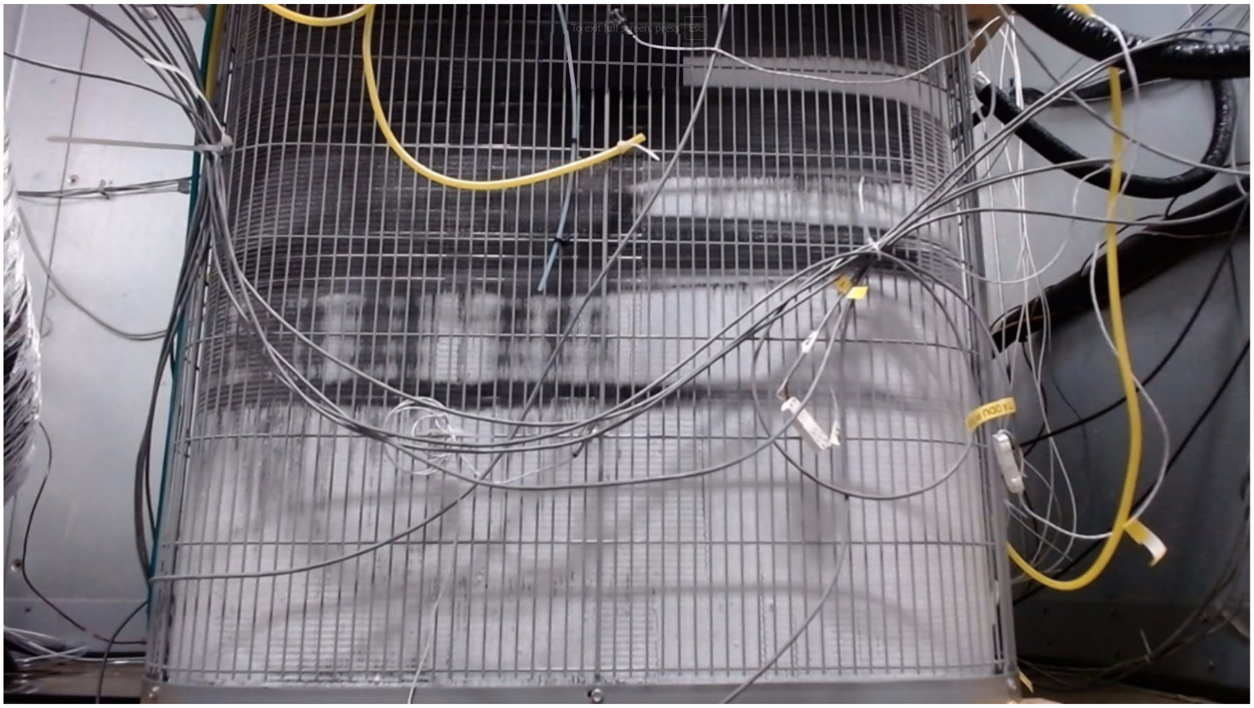

Figure 6. ODU midway through a defrost cycle.

Water Heating First-Hour Rating Testing

After concluding environmental chamber tests to measure the performance of each of the MFHP's operational modes, we performed a FHR test on the system. The FHR is a standard metric used to estimate the maximum volume of hot water a water heater can supply in an hour, assuming it begins with a fully preheated WT. We performed the test according to the Department of Energy's (DOE's) Uniform Test Method for Measuring the Energy Consumption of Water Heaters (U.S. DOE 2024). Table 4 below summarizes the conditions required to be maintained while performing the test.

Table 4. Test conditions for FHR test from the U.S. DOE's Uniform Test Method for Measuring the Energy Consumption of Water Heaters.

	Test Parameters	Notes
Ambient Air Temperature (DB)	67.5 ± 1°F, during an active WH call 67.5 ± 2.5°F, after a WH call terminates and before another begins 67.5 ± 5°F, throughout the test	For HPWHs; ambient conditions for HP unit (outdoor chamber)

	Test Parameters	Notes
Ambient Air Relative Humidity (RH)	50 ± 2%, during an active WH call 50 ± 5%, throughout the test	
Supply Water Temperature	58 ± 2°F	–
Outlet Water Temperature	125 ± 5°F	–
Supply Water Pressure	P_{supply} > 40 psig	When water is not actively being drawn
Flow Rate	3.0 ± 0.25 GPM	For WHs with storage volumes ≥ 20 gals

Source: U.S. DOE Title 10, Chapter II, Subchapter D, Part 420, Subpart B, Appendix E

To determine the appropriate WH temperature setpoint to target the desired outlet water temperature range, we first performed a calibration test according to Section 5.2 of the test standard, on water heater preparation. The calibration involved heating the water tank to some setpoint temperature and observing the maximum outlet WT for a five-minute water draw at a 1.7 ± 0.25 GPM flow rate, starting immediately after the Aquastat terminated the unit's call for WH. If the maximum outlet temperature observed fell outside the allowable temperature range (125 ± 5°F), then the team was to repeat the procedure for a different setpoint temperature until one satisfied the condition for the FHR test. By applying this procedure, the team determined 115°F was an appropriate Aquastat setpoint for the FHR test.

The team then performed the FHR test according to the procedure outlined in Section 5.3.3 of the test standard. The Aquastat was programmed to a 115°F setpoint with a 10°F deadband. Once the WT was fully preheated and the Aquastat terminated the call for WH, the team initiated the first water draw upon the WT reaching a maximum mean tank temperature. The start of the first water draw marked the beginning of the one-hour test period for the FHR test. The end of WH calls was observed by the MFHP compressor power cutting out, and the mean water tank temperature was calculated from the array of temperature sensors installed at interval heights within the WT. The team estimated the maximum outlet water temperature of the draw, beginning fifteen seconds after the draw was initiated, and terminated the water draw once the outlet water temperature dropped 15°F below the maximum observed. Subsequent water draws were initiated after the WT had reheated and the Aquastat stopped calling for WH, and were terminated when the outlet water temperature dropped 15°F below the maximum outlet temperature observed starting fifteen seconds after the start of the draw. At the one-hour mark, the WT was in an active WH cycle, so the team forced a final draw, as per the test procedure. The power to the compressor was cut off and the draw was initiated. The final draw was terminated when the outlet water temperature dropped below the cut-off temperature used for the previous water draw.

Instrumentation

Data Collection

Each test condition was run for several hours until the chamber temperature and humidity, and equipment operation reached steady state to start the 30-minute test. During each 30-minute test duration, 10-second-averaged data was saved for the MFHP, and one-minute-averaged data were saved for the environmental chamber. The data was collected using National Instruments Compact DAQ data acquisition system and LabVIEW software. These data sets were aligned and averaged over the 30-minute steady state duration for summary calculations of all the tests. These average values were used to calculate desired capacity and operational metrics. If there were irregularities in any readings during a test, then the irregularity was fixed and that test was repeated.

Temperatures and Humidity

Indoor and outdoor chamber air temperatures and dew points were monitored and controlled with RTD probes and OptiSonde chilled mirror hygrometers positioned at the SA and RA ducts close to the connection with the air handler unit. 4-point humidity sampling arrays were utilized to improve sampling quality at each of these points and an RTD probe measured DB temperature. The outdoor air temperature was monitored using 8 RTDs positioned around the ODU condenser coil inlet. The averaged value of these outdoor air sensor arrays was taken as the effective outdoor DB and passed through to the chamber control loop.

These temperatures, dew point, and RH measurements were then used in a psychrometric calculator to derive any other psychrometric properties of desired air flows, including wet bulb temperature, humidity ratio, relative humidity, and air densities.

Air Flow Rates

Volumetric flow rates were measured and estimated by using two nozzle boxes on each of the total flows into each chamber. Mass flows rates were calculated from the volumetric flows and densities calculated from the psychrometric properties of the given air flow.

Power

Power was measured at the power connection to the MFHP ODU and AHU separately, using two Dent PowerScouts, which record the values of all three phase currents and voltages.

Water Flow Rate and Temperatures

The water flow rate was measured on the outlet side of the WT using an Omega flow meter. The inlet and outlet water temperatures were measured using Omega pipe plug RTDs on the inlet and outlet ports to the water tank. WT temperatures were measured by an array of seven hermetically sealed Omega RTDs positioned at interval heights of approximately 10” down the center of the WT.

Refrigerant Flow Rate, Temperatures, and Pressures

The MFHP unit features two refrigerant line sets, connecting the ODU to the AHU and to the WT. Omega surface-mount RTDs and ClimaCheck pressure transducers were used to measure refrigerant temperature and pressure, respectively, on the liquid refrigerant lines (LL) and vapor refrigerant lines (VL) at the AHU and WT. Refrigerant flow rate was measured from the ODU to the WT using a Micro-motion F-series Coriolis flow meter.

Pressures

ESP was measured at the curb interface of the MFHP AHU, at both the SA and RA plenums. This was accomplished using two of the channels on a DG8 differential pressure meter. Further static pressure measurements were recorded along the ODU condenser coil inlet, inside the ODU cavity, and at the ODU integral fan. All pressures were measured with an Energy Conservatory differential pressure gauge, and an atmospheric absolute pressure gauge in the outdoor environmental chamber.

The sensors used are detailed below (see Table 5), as is sensor placement (see Figure).

Table 5. Table of sensors and their model numbers.

Air Pressure Sensors	Channel	Model
ODU Inlet Air Pressure	P_A_ODU_IN	Energy Conservatory APT
ODU Internal Cavity Air pressure	P_A_ODU_CAV	
ODU Exhaust Air Pressure	P_A_ODU_EXH_1	
ODU Exhaust Air Pressure	P_A_ODU_EXH_2	
AHU Inlet Air Pressure	P_A_AHU_IN	
AHU Outlet Air Pressure	P_A_AHU_OUT_1	
AHU Outlet Air Pressure	P_A_AHU_OUT_2	
Indoor Chamber Air Pressure	P_A_IC	

Air Temperature RTDs #'d Top to Bottom Clockwise from Lines	Channel	Model
ODU Condenser Inlet Temp, Side 1 Upper	T_A_ODU_RTU 1	Omega 100Ω Class A Platinum RTD-805
ODU Condenser Inlet Temp, Side 1 Lower	T_A_ODU_RTU 2	
ODU Condenser Inlet Temp, Side 2 Upper	T_A_ODU_RTU 3	
ODU Condenser Inlet Temp, Side 2 Lower	T_A_ODU_RTU 4	
ODU Condenser Inlet Temp, Side 3 Upper	T_A_ODU_RTU 5	

Air Temperature RTDs #'d Top to Bottom Clockwise from Lines	Channel	Model
ODU Condenser Inlet Temp, Side 3 Lower	T_A_ODU_RTU 6	
ODU Condenser Inlet Temp, Side 4 Upper	T_A_ODU_RTU 7	
ODU Condenser Inlet Temp, Side 4 Lower	T_A_ODU_RTU 8	

Water Temperatures	Channel	Model
WT Water Temp In	T_W_WT_In	Omega RTD-NPT-72-E-1/4
WT Water Temp Out	T_W_WT_Out	
WT Temps 1-7 (Bottom to Top)	T_W_WT_1-7	HSRTD-3-100-A

Dew Point & Temperature Chilled Mirrors	Channel	Model
Outdoor Chamber / ODU Inlet	DP_A_ODU_IN T_A_ODU_IN	GE OptiSonde Chilled mirror Hygrometer
ODU Exhaust	DP_A_ODU_EXH T_A_ODU_EXH	
IC / AHU Return Air (Inlet)	DP_A_AHU_IN T_A_AHU_IN	
AHU Supply Air (Outlet)	DP_A_AHU_OUT T_A_AHU_OUT	

Electrical Power	Channel	Input	Model
ODU PowerScout	PS_ODU	RS-485 Converter	Dent PowerScout PS3037
AHU PowerScout	PS_AHU		

Water Tank Pressure & Flow Rate	Channel	Model
Water Pressure	P_W_WT	OMEGADYNE PX309-200A5V
Water Flow Rate	FV_W_WT	Omega FTB4607

Refrigerant Temperature	Channel	Model
WT Refrigerant Vapor Line Temperature	T_R_WT_VL	Omega SA1-RTD Surface-Mount RTD
WT Refrigerant Liquid Line Temperature	T_R_WT_LL	
AHU Refrigerant Vapor Line Temperature	T_R_AHU_VL	
AHU Refrigerant Liquid Line Temperature	T_R_AHU_LL	

Refrigerant Pressure	Channel	Model
WT Refrigerant Vapor Line Pressure	P_R_WT_VL	ClimaCheck Pressure Transducer 22S, 50 Bar, Teflon seal, 1-5 V
WT Refrigerant Liquid Line Pressure	P_R_WT_LL	
AHU Refrigerant Vapor Line Pressure	P_R_AHU_VL	
AHU Refrigerant Liquid Line Pressure	P_R_AHU_LL	

Refrigerant Flow Rate	Channel	Model
WT Refrigerant Flow Rate	F_R_WT	Micro-Motion F-series Coriolis flow meter

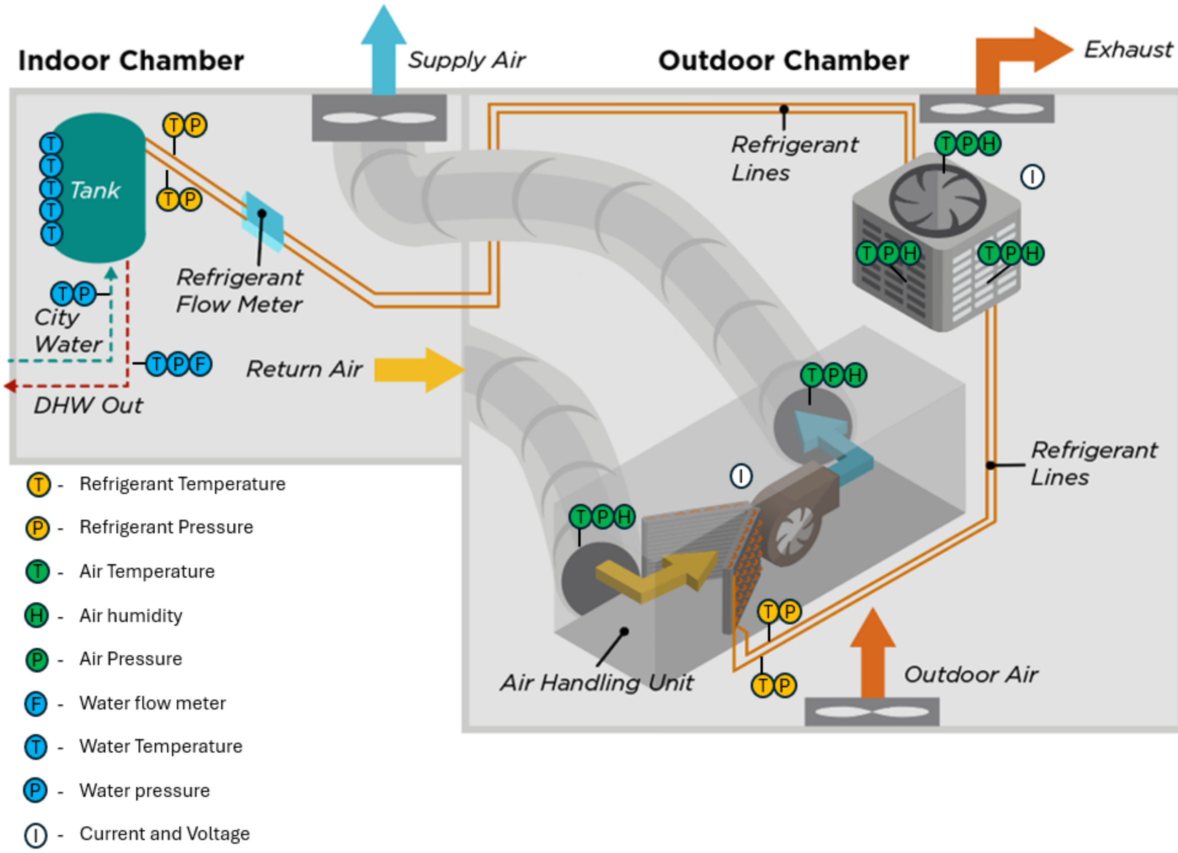

Figure 7. Schematic of the test setup showing the location of the MFHP unit in the environmental chambers and placement of instrumentation sensors.

Data Analysis Equations

Air Enthalpy

The mixed air refers to the air conditions entering the indoor coil (evaporator for SC). Due to difficulty directly measuring this condition, the mixed air enthalpy was calculated based on the known amount of RA and OA entering the indoor coil. Corrections in the SA and RA conditions were made according to measured pressure differentials and conditions of any air leaking into the system. The leak amounts were measured by isolating SA and RA sections of the AHU and mapping flow rates to various depressurization states. Mixed air corrections were calculated for any test conditions where supply or return side of the AHU were impacted. Enthalpies were calculated using methods from the ASHRAE fundamentals psychrometric calculations, where inputs of absolute pressure, dry bulb temperature, and dew point temperature are used to calculate enthalpy and any other psychrometric properties. As an example:

$$h_{SA} = f(p_a, T_{SA,DB}, T_{SA,DP})$$

Where:

h_{SA} is the supply air enthalpy
$p_{SA,a}$ is the supply air absolute pressure

$T_{SA,DB}$ is the supply air dry bulb temperature
$T_{SA,DP}$ is the supply air dew point temperature

Cooling and Heating Capacity

The SC capacity, $\dot{H}_{SC}$, of the system was calculated based on mass flow rate and the difference in specific enthalpy between the RA and SA. This is the net cooling produced by the MFHP in SC mode, including what is lost due to fan heat.

Equation 1: Cooling Capacity

$$\dot{H}_{SC} = \dot{m}_{SA} \cdot (h_{RA} - h_{SA})$$

Where:
h_{SA} is the supply air enthalpy
h_{RA} is the return air enthalpy
$\dot{m}_{SA}$ is the supply air mass flowrate

The corresponding sensible capacity, $\dot{H}_{SC}^{sen}$, is determined by the air mass flow rate, the specific heat of air, and the temperature difference between the RA and SA stream.

Equation 2: Sensible Cooling and Heating Capacity

$$\dot{H}_{SC}^{sen} = \dot{m}_{SA} \cdot C_{p,air} \cdot \left(T_{RA,DB} - T_{SA,DB}\right)$$

Where:
$T_{RA,DB}$ is the return air dry bulb temperature
$T_{SA,DB}$ is the supply air dry bulb temperature
$C_{p,air}$ is the specific heat of air
$\dot{m}_{SA}$ is the supply air mass flowrate

Concomitantly, the latent cooling capacity, $\dot{H}_{SC}^{lat}$, was determined as the difference between the total and sensible capacities.

Equation 3: Latent Cooling Capacity

$$\dot{H}_{SC}^{lat} = \dot{H}_{SC} - \dot{H}_{SC}^{sen}$$

Similarly, when the HP runs in SH mode, the heating capacity, $\dot{H}_{SH}$, for the AHU was determined at any operating condition according to the SA mass flow rate and the specific enthalpy difference between the SA and RA stream. Note that both SC and SH capacities were calculated to typically yield positive values of capacity for their corresponding modes.

Equation 4: Space Heating Capacity

$$\dot{H}_{SH} = \dot{m}_{SA} \cdot (h_{SA} - h_{RA})$$

Water Tank Thermal Storage and Capacity

The water tank storage capacity (WTH) was determined using volume, density and specific heat properties of the primary WT components – water, copper, and high-density polyethylene (HDPE) liner – according to measured volumes and design documentation. The WT temperatures were measured using a vertical array of seven sensors to capture temperature stratification and differences in refrigerant and water draw coils in the WT. Total thermal energy stored in the WT was calculated as a sum over the vertically stratified temperatures, along with physical properties of the water and WT components.

Equation 5: Water Tank Heat Storage capacity

$$H_{WTH,tot} = \sum_{i=1}^{n} H_{WTH,i}$$

Where for each control volume i

$$H_{WTH,i} = (m_{WT,i} \cdot C_{p,water} + m_{Cu,i} \cdot C_{p,Cu} + m_{HDPE,i} \cdot C_{p,HDPE}) \cdot T_{WT,i}$$

$m_{WT,i}$ is the mass of water for control volume i
$m_{Cu,i}$ is the mass of heat exchanger copper for control volume i
$m_{HDPE,i}$ is the mass of the HDPE tank liner for control volume i
$C_{p,water}$ is the specific heat of water
$C_{p,Cu}$ is the specific heat of copper
$C_{p,HDPE}$ is the specific heat of HDPE
$T_{WT,i}$ is the water temperature measurement for control volume i

To equate and analyze these thermal energies as they relate to the water heating and supply water delivery capacities, we summed a first order time derivative (rate of change) for the temperatures in the WT. This yields a WT total rate of change in stored thermal energy which equates to the combination of water heating and supply water delivery capacities. During supply water draws and stagnant storage periods, this value will be negative and represents the rate at which thermal energy is removed from the storage tank. We can then sum these rates of change to quantify heat capacity gain of the entire control volume.

Equation 6: Water Tank Heating Capacity

$$\dot{H}_{WTH,tot} = \sum_{i=1}^{n} \dot{H}_{WTH,i} = \sum_{i=1}^{n} \left(m_{water,i} \cdot C_{p,water} + m_{Cu,i} \cdot C_{p,Cu} + m_{HDPE,i} \cdot C_{p,HDPE}\right) \cdot \dot{T}_{WT,i}$$

Simultaneous Water Heating and Space Cooling

For SIM mode for the MFHP, both the SC and WTH capacities were calculated and added together to get a total "combined" useful thermal energy capacity, $\dot{H}_{SIM}$

Equation 7: Simultaneous Space Cooling Water Heat Recovery Capacity

$$\dot{H}_{SIM} = \dot{H}_{SC,SIM} + \dot{H}_{WTH,SIM}$$

Where

$$\dot{H}_{SC,SIM} = \dot{m}_{SA} \cdot (h_{RA} - h_{SA})$$

$$\dot{H}_{WTH,SIM} = \sum_{i=1}^{n} \left(m_{water,i} \cdot C_{p,water} + m_{Cu,i} \cdot C_{p,Cu} + m_{HDPE,i} \cdot C_{p,HDPE} \right) \cdot \dot{T}_{WT,i}$$

Refrigerant-Side Water Heating Capacity

The refrigerant-side measurements (pressures, temperatures, and mass flow rates) were used to calculate the heat energy rate delivered by the refrigerant to the WT. Superheated refrigerant vapor is sent to the WT and comes out as sub-cooled liquid and enthalpies of the refrigerant in both the vapor line and liquid line are calculated as follows:

$$h_{R,VL} = f\left(P_{R,VL}, T_{R,VL}\right)$$

$$h_{R,LL} = f\left(P_{R,LL}, T_{R,LL}\right)$$

The water heating capacity for the MFHP in both WH and SIM mode was calculated as follows:

Equation 8: Refrigerant Water Heating Capacity

$$\dot{H}_{R,WH/SIM} = \dot{m}_{R,LL} \cdot \left(h_{R,VL} - h_{R,LL}\right)$$

Supply Water Heating Capacity

The DHW heating capacity, $\dot{H}_{HWS}$, was determined by the supply water mass flow rate, the specific heat of water, and the temperature difference between the supply and inlet water streams.

Equation 9: Supply Water Heating Capacity

$$\dot{H}_{HWS} = \dot{m}_{SW} \cdot C_{p,water} \cdot (T_{inlet} - T_{SW})$$

Where:

$\dot{m}_{SW}$ is the mass flow rate of supply water

$C_{p,water}$ is the specific heat of water

T_{inlet} is the inlet water temperature

T_{SW} is the supply water temperature exiting the tank

Water Tank Skin Heat Loss

Another piece needed to fully evaluate MFHP performance against any other collection of devices that provide DHW, is the WT heat loss. EnergyPlus models tank thermal losses to the environment using a uniform skin loss coefficient per unit area to ambient temperature [W/m^2-K] also known as U-value. This is detailed in section 1.24.3.1.31 of the EnergyPlus Input-Output documentation. Overnight tank heat loss tests were used to calculate the uniform skin loss coefficient for the WT using the equations below.

Equation 10: Newton's law of cooling heat loss

$$\dot{H}_{WTloss} = U\,A\left(T_{WT} - T_{surrounding}\right)$$

Equation 11: Conservation of Energy

$$\dot{H}_{WTloss} = \dot{Q} = m_{WT}\,C_{p,water}\,\dot{T}_{WT}$$

Equation 12: Skin Loss Coefficient (U factor)

$$U = \frac{m_{WT}\,C_{p,water}\,\dot{T}_{WT}}{A\left(T_{WT} - T_{surrounding}\right)}$$

Where

m_{WT} is the mass of water in the WT
$C_{p,water}$ is the specific heat of water
T_{WT} is the WT average temperature
$\dot{T}_{WT}$ is the rate of change of the WT average temperature with time
$\dot{Q}$ is the rate of change in thermal energy with time
A is the surface area of the WT
$T_{surrounding}$ is the air temperature surrounding the WT

Calculating Coefficient of Performance and Electric Input Ratio

Energy efficiency at any given operating condition is expressed as COP – the dimensionless ratio of useful thermal energy used to electrical power consumed – or, inversely, the electric input ratio (EIR). Both are commonly used, as COP intuitively increases as EE increases and EIR increases with power draw, which can be more intuitive when analyzing energy use. For this reason, both are used as parameters to quantify performance in EnergyPlus modeling curves.

Equation 13: Coefficient of Performance and Electric Input Ratio

$$COP = \frac{Thermal\ Energy\ Delivery\ Rate}{Electrical\ Energy\ Consumtion\ Rate} = \frac{\dot{H}}{\dot{E}}$$

and

$$EIR = \frac{1}{COP} = \frac{Electrical\ Energy\ Consumtion\ Rate}{Thermal\ Energy\ Delivery\ Rate} = \frac{\dot{E}}{\dot{H}}$$

The COP and EIR were calculated for each of the MFHP modes using their respective capacities (calculated previously) and power use. Thus, for the SIM mode, both the WH and SC capacities were summed, resulting in a combined COP when SC coincides with WH. As an example:

$$EIR_{SIM} = \frac{\dot{E}_{SIM}}{\dot{H}_{SIM}}$$

Water Heating First Hour Rating

The first-hour rating of a water heater is an estimate of how much hot water it can supply within an hour of operation. Given that the MFHP required a forced final draw, the below equation was used to calculate the FHR for the system:

Equation 14: First Hour Rating

$$FHR = V_n^* \left(\frac{T_{del,n}^* - T_{min,n-1}^*}{T_{del,n-1}^* - T_{min,n-1}^*} \right) + \sum_{i=1}^{n} V_i^*$$

Where:

V_i^* is the volume removed during the *i*th water draw

V_n^* is the volume removed during the *n*th (final) water draw

$T_{del,n}^*$ is the average outlet water temperature observed during the *n*th (final) water draw

$T_{del,n-1}^*$ is the average outlet water temperature observed during the *(n-1)*th water draw

$T_{min,n-1}^*$ is the minimum outlet water temperature observed during the *(n-1)*th water draw

Performance Curve Requirements

Future projects will perform whole building energy simulations using CalTF EnergyPlus prototypes to estimate MFHP energy savings across climate zones and residential building types. There are several available approaches to model the MFHP in EnergyPlus with advantages and drawbacks. The paths forward that we have Identified in EnergyPlus are using the Python Plugin, Plant/Energy Storage, Coil System as is, and Coil System with limited EnergyPlus Development.

Going the Python Plugin route enables quick application of detailed models already completed by our team to be indirectly used by EnergyPlus to aid in the simulation. This would effectively use the MFHP models defined in Python to hand the capacity and energy information to EnergyPlus as it hands outdoor and indoor conditions to the model. The major drawback of this is the lack of shareability with others that wish to model this type of system.

The Plant/Energy Storage options in EnergyPlus are numerous, given the large number of EnergyPlus objects available for commercial building heating and cooling systems. Our team expect this would be a viable route but would be complicated and require numerous assumptions to yield an operable model. This route would also limit shareability beyond our team.

The last approach is the Coil System object made for an air-source integrated heat pump (ASIHP) model developed by Shen et al. (Shen, New and Baxter 2017). The ASIHP is very similar to the MFHP in that they both have several operating modes and are both attached to a water tank. This coil system object in EnergyPlus can be used as is for matching modes, but other modes such as MFHP defrost from the WT will require some modification to the objects in EnergyPlus. In our opinion, the best route forward for the future projects will be to work with the EnergyPlus developers to tweak the ASIHP coil system object in EnergyPlus to be more expandable to incorporate additional operational modes. This would enable both our team and a wide range of other users to accurately simulate different kinds of MFHPs with various mode architectures. Because the changes necessary to EnergyPlus seem minimal, our team believes this to be the most prudent course forward.

The ASIHP coil system uses the following EnergyPlus objects, and the team has developed the required performance curves for each object based on the lab test data. These performance curves should also be sufficient to describe the MFHP performance if any of the other simulation approaches previously described be chosen.

The following lists the required data for each mode performance curve. EnergyPlus performance curves are benchmarked at a rating condition and the curve outputs a multiplication factor that is multiplied by the gross rating/capacity to yield the modified rating for those conditions (see Table 6).

The following are the coil objects referenced from the EnergyPlus Input-Output reference document informing the required curve types and format. The numbering has been retained for quick reference to the source document.

1.41.16 Coil:Cooling:DX:SingleSpeed (Dedicated Space Cooling Mode)

1.41.22 Coil:Heating:DX:SingleSpeed (Dedicated Space Heating Mode)

1.41.31 Coil:WaterHeating:AirToWaterHeatPump:Wrapped (Dedicated Water Heating Mode)

1.41.31 Coil:WaterHeating:AirToWaterHeatPump:Wrapped (But for Simultaneous Space Cooling and Water Heating Mode)

1.41.37 Coil:Heating:WaterToAirHeatPump:EquationFit (Not a defrost object, but used to inform the likely required performance equations for defrost)

A list of the required performance information is included in the appendix. Otherwise, the following information is required.

Table 6. Required data for EnergyPlus simulation (temperatures in °C, as specified by EnergyPlus).

Required Data	SH	SC	WH	SIM	DEF
Rated Heating/ Cooling Capacity	@ rated conditions	@ rated conditions	@ rated conditions	@ rated conditions	@ rated conditions
Rated COP	@ rated conditions	@ rated conditions	@ rated conditions	@ rated conditions	@ rated conditions
Rated Air Flow Rate	@ rated conditions	@ rated conditions	–	@ rated conditions	–

Required Data	SH	SC	WH	SIM	DEF
Rated SHR	@ rated conditions	@ rated conditions	@ rated conditions	@ rated conditions	–
Rated IA DB	21.11	–	–	19.7 (user defined)	–
Rated IA WB	15.55	19.44	–	13.5 (user defined)	–
Rated OA DB	8.33	35	19.4 (user defined)	–	19.7 (user defined)
Rated OA WB	6.11	–	13.5 (user defined)	–	–
Rated Water Temperature	–	–	57.5 (user defined)	57.5 (user defined)	57.5 (user defined)
Capacity Function of Temperature	f($T_{OA,DB}$ & $T_{IA,DB}$) | f($T_{OA,DB}$)	f($T_{IA,WB}$ & $T_{OA,DB}$)	f($T_{OA,DB}$ | $T_{OA,WB}$ & T_{water}) | f($T_{OA,DB}$ | $T_{OA,WB}$)	f($T_{IA,WB}$ | $T_{IA,DB}$ & T_{water}) | f($T_{OA,DB}$ | $T_{OA,WB}$)	f($T_{OA,DB}$ | $T_{OA,WB}$ & T_{water}) | f($T_{OA,DB}$ | $T_{OA,WB}$)
Capacity Function of Flow Fraction	f(V)	f(V)	–	f(V)	–
EIR/COP Function of Temperature	f($T_{OA,DB}$ & $T_{IA,DB}$) | f($T_{OA,DB}$)	f($T_{IA,WB}$ & $T_{OA,DB}$)	f($T_{OA,DB}$ | $T_{OA,WB}$ & T_{water}) | f($T_{OA,DB}$ | $T_{OA,WB}$)	f($T_{IA,DB}$ | $T_{IA,WB}$ & T_{water}) | f($T_{OA,DB}$ | $T_{OA,WB}$)	f($T_{OA,DB}$ | $T_{OA,WB}$ & T_{water}) | f($T_{OA,DB}$ | $T_{OA,WB}$)
EIR/COP Function of Flow	f(V)	f(V)	–	f(V)	–
PLR Correlation Curve	f(PLR)	f(PLR)	f(PLR)	f(PLR)	–

EnergyPlus Performance Modeling Equations

Biquadratic fits, typically using $T_{OA,DB}$ and $T_{RA,WB}$ as independent variables for SC, are used to model capacity and EIR. For SH, typically $T_{RA,DB}$ replaces $T_{RA,WB}$. Our team took measured and calculated values for capacity and EIR; normalized them for typical rated conditions; and finally used a least squares fit over the biquadratic terms of the two input variables. For example:

Equation 15: Capacity normalization for modeling

$$\dot{H}_{SC,norm} = \frac{\dot{H}_{SC}}{\dot{H}_{SC,rated}}$$

Where:
$\dot{H}_{SC,rated}$ is the rated SC mode capacity measured at 35°C OA DB and 19.44°C IA WB (95°F OA DB/67°F IA WB).

Now, using the equation form below, our team can calculate the constants $\boldsymbol{C}$ using a least square calculation, for various performance metrics of the system.

Equation 16: Biquadratic performance modeling over operational temperature range

$$\dot{H}_{SC,norm} = \overline{\boldsymbol{C}} \cdot \left[T_{OA,DB}^2 \quad T_{RA,WB}^2 \quad T_{OA,DB} \cdot T_{RA,WB} \quad T_{OA,DB} \quad T_{RA,WB} \quad 1\right]$$

Where:
$T_{OA,DB}$ is the outdoor air dry bulb temperature
$T_{RA,WB}$ is the return air wet bulb temperature
$\overline{\boldsymbol{C}} = [C_1 \quad C_2 \quad C_3 \ldots C_n]$ is an equal length vector of constants corresponding to the biquadratic vector of input terms to fit least squares.

This same form is used for other performance modeling inputs of EnergyPlus using appropriate input terms for each. For example, space heating modes use, $T_{RA,DB}$, the return air dry bulb temperature in place of the return air wet bulb because heating capacities are more sensitive to changes in the indoor dry bulb temp. For water heating capacities, appropriate water tank and water inlet temperatures are used. The input terms are chosen to be variables that would be known in the model with the highest correlation to predicting the desired output.

Lab Test Findings

Overview

The laboratory evaluation of this single-speed MFHP unit verified the AHRI rated performance of the single-speed base HP unit before conversion to MFHP.

The equipment performance for the SH and SC modes matched the manufacturer-published AHRI rated capacity and efficiency within experimental uncertainty. This shows that the discrepancy seen in the field test of the previous study was due to suboptimal installation or defective components.

Results

Space Cooling (SC)

Figure 8 and Figure 9 show the lab test measured SC capacity and COP for the 4-ton rated single-speed MFHP in SC mode. These figures also show the manufacturer published data for the conventional HP (Mftr.) before it was converted to a MFHP with the actuating valves and solenoids. As seen from the Figure 8, the manufacturer curve for the IA at 63WB aligns very well with the test data of the MFHP unit. Regression curve fits to the laboratory data align with manufacturer published data to within experimental uncertainty in the tested region of conditions. This verifies that the manufacturer data is reasonable. Curve fits using lab test data alone had a lower/flatter slope than manufacturer data which could overpredict COP at extremely high outdoor temperatures and lead to overestimates of Total System Benefit. For this reason, the project team included manufacturer published performance data for 115°F and 125°F outside temperatures to influence the curvature of the modeling fits and avoid over predicting COP. At 95°F OA DB and IA DB/WB of 80°F/67°F, the MFHP SC mode had a capacity of 45.3 kBTU/h at a COP of 3.57. This suggests the lower capacities seen in the previous field test of the MFHP prototype could be a result of poor commissioning in the field or a defective piece of equipment.

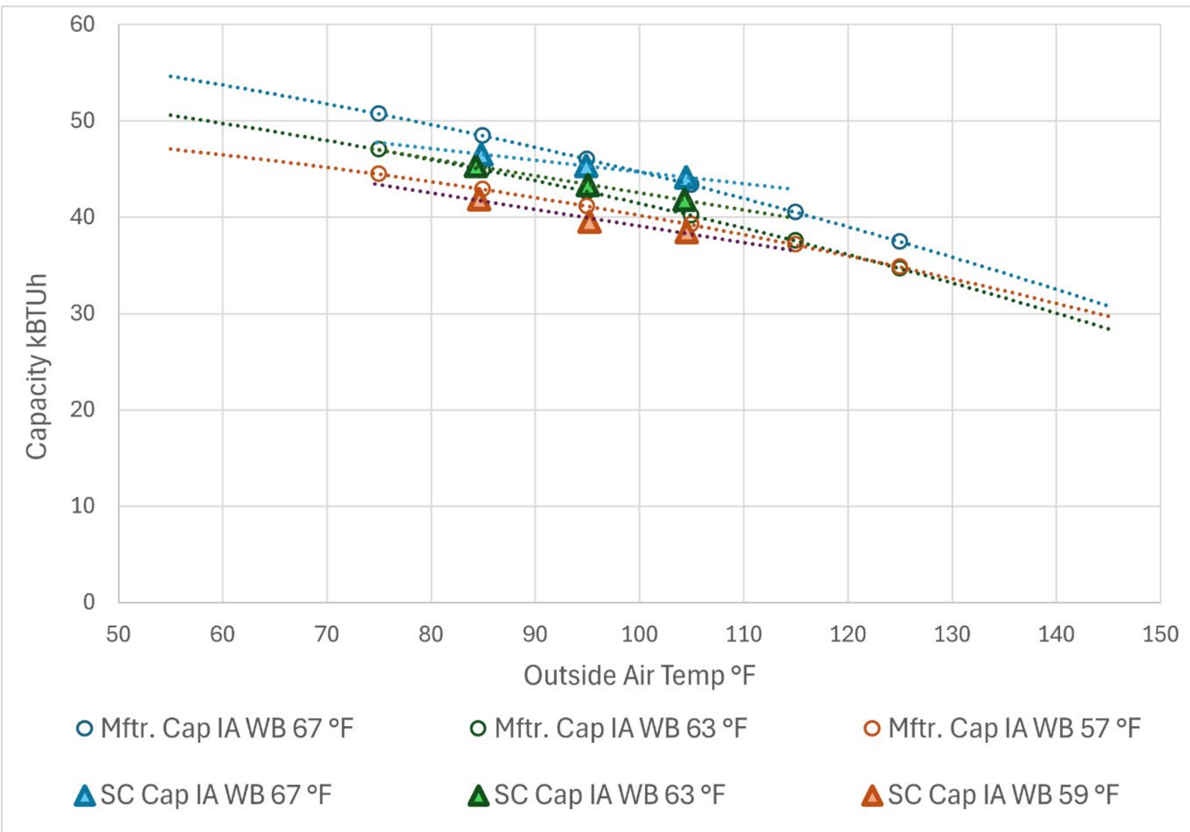

Figure 8. Measured MFHP SC capacity in each mode at different outdoor and indoor conditions, compared to the manufacturer-published performance data.

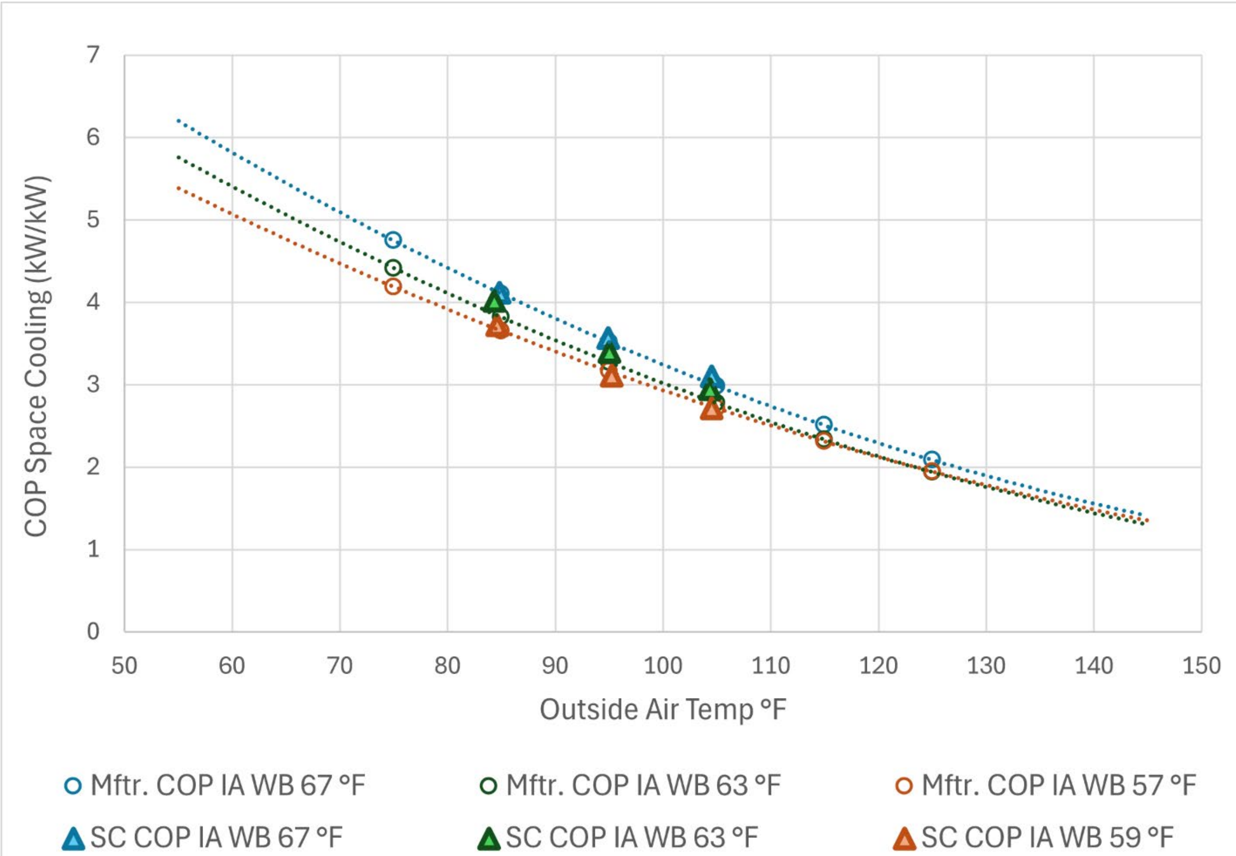

Figure 9. Measured MFHP SC COP in each mode at different outdoor and indoor conditions, compared to the manufacturer-published performance data.

Space Heating (SH)

Figure 10 and Figure 11 show the lab test-measured SH capacity for the 4-ton rated single-speed MFHP in SH mode compared to manufacturer data for the conventional HP before it was converted to a MFHP with the actuating valves and solenoids.

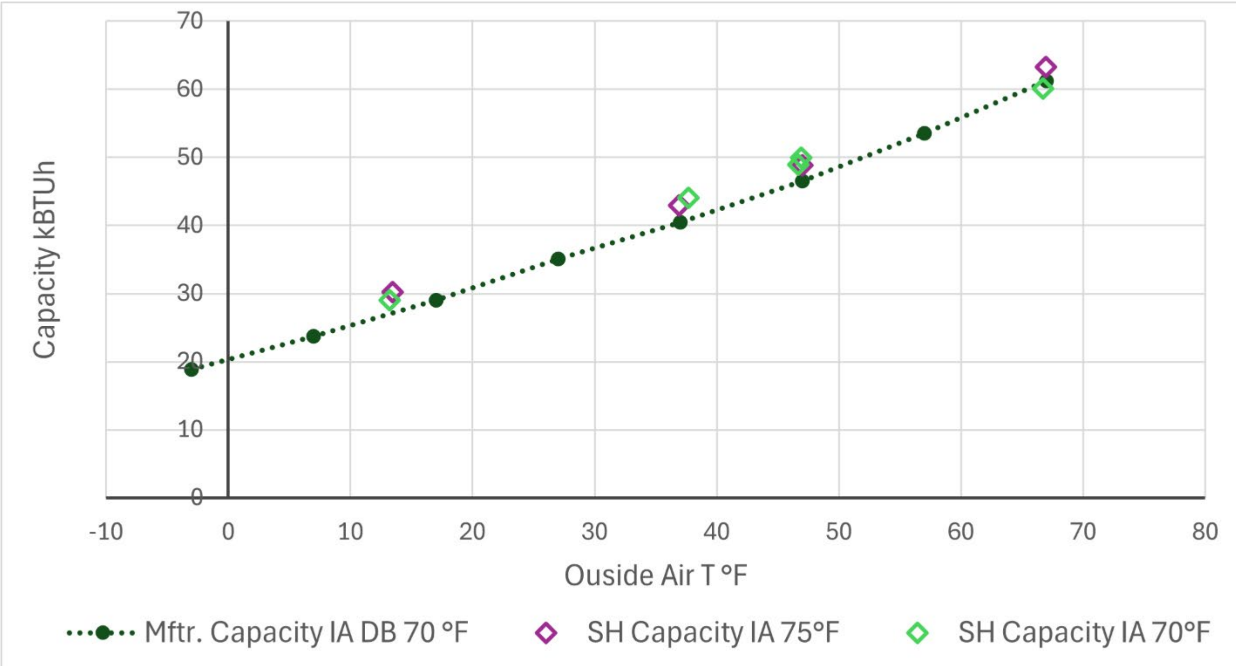

Figure 10. Measured MFHP SH capacity at different outdoor and indoor conditions, compared to the manufacturer-published performance data

The manufacturer curve for the SH capacity and COP aligns with the test data of the MFHP within experimental uncertainty. Two different IA DB conditions are shown for SH across a range of outdoor conditions. At 47°F OA DB and IA DB/WB of 70°F/60°F, the SH mode had a capacity of 49.9 kBTU/h at a COP of 3.48.

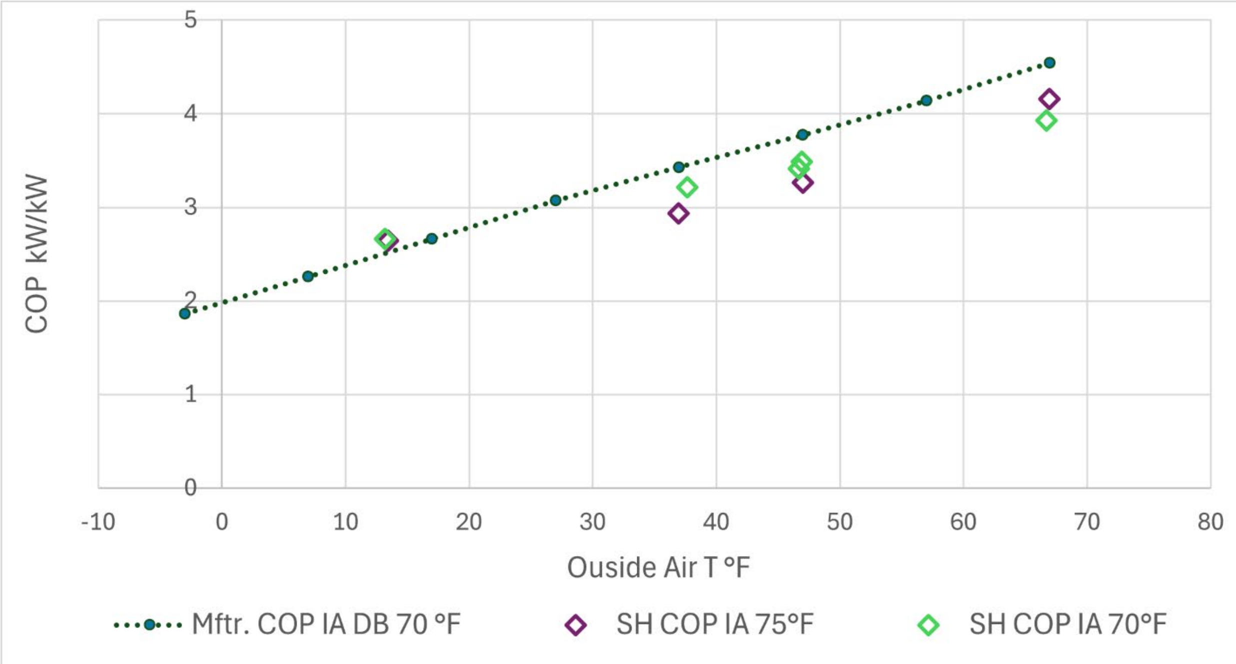

Figure 11. Measured MFHP SH COP at different outdoor and indoor conditions, compared to the manufacturer-published performance data.

For all tested indoor conditions, the MFHP maintained reliable performance at low outdoor temperature conditions. The MFHP switched between DEF and SH mode at the tested 37DB and 17DB outdoor conditions. The DEF operation is analyzed further in the DEF mode section.

Water Heating (WH)

Water heating capacity calculated from changes in the water tank thermal storage called the hot water tank capacity ($\dot{H}_{WTH,tot}$) was initially found to be significantly lower than the water heating capacity calculated from refrigerant flow rate and properties measurements ($\dot{H}_{R,SIM/SC}$). $\dot{H}_{WTH,tot}$ was then adjusted to account for the thermal mass of the copper, water volume in the water-to-water heat exchanger, and plastic liner of the tank for improved accuracy.

The water heating capacity presented in Figure 12 is based on these updated water tank thermal storage capacity calculations ($\dot{H}_{WTH,tot}$). The water heating capacities were measured at four WT setpoint temperatures (105, 115, 125, 135°F) to cover a range of conditions under which WH will operate. As expected for all heat pump water heating equipment, the water heating capacity decreases for WH mode as the WT setpoint temperature rises. At 47°F OA DB and WT setpoint temperature of 115°F the WH mode demonstrated a capacity of 36.1 kBTU/h at a COP of 2.67.

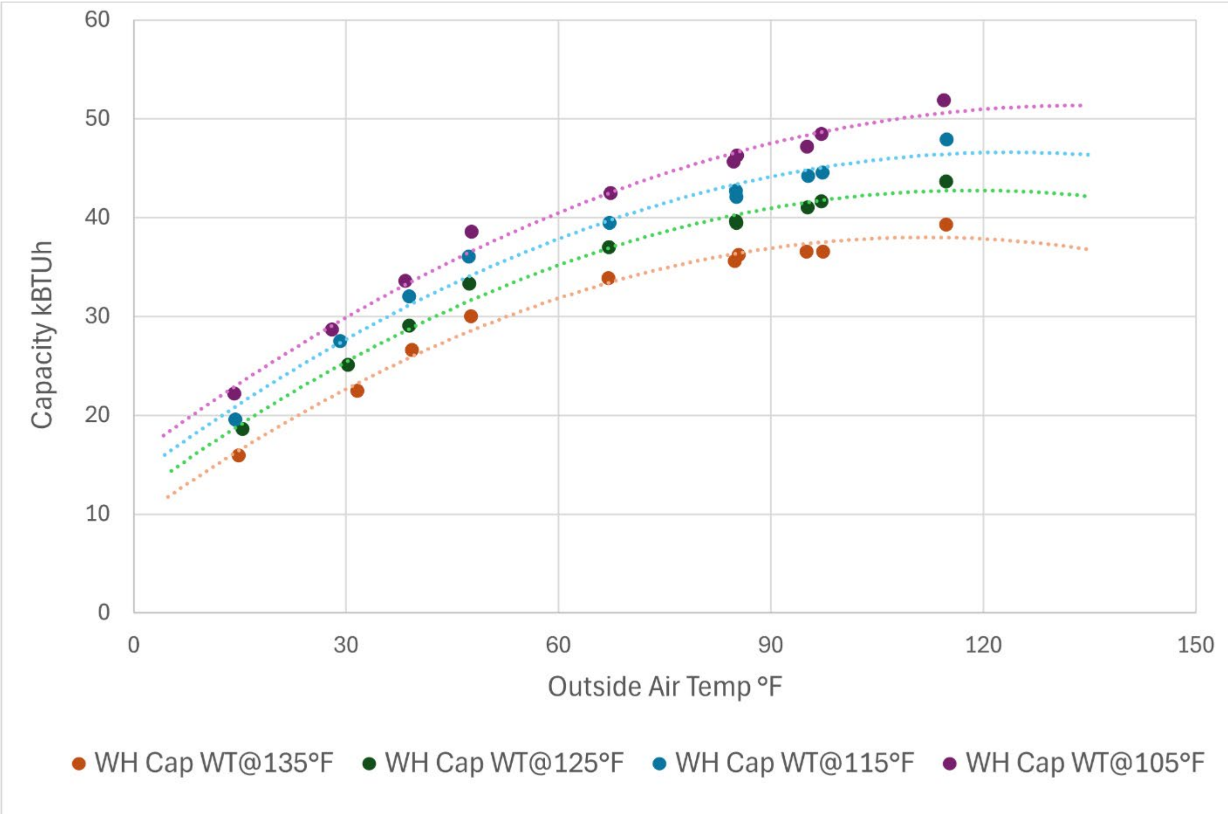

Figure 12. Measured MFHP WH capacity at different outdoor and WT setpoint conditions.

Figure 13 shows the measured WH COP with respect to outdoor air temperature. As expected, the highest COP occurs with low WT setpoint temperatures and higher OA temperatures and decreasing trend as WT setpoint temperatures increase and OA temperatures decrease. At the 67.5 °F outdoor condition the WH mode demonstrated a COP of 2.95 for WT temperature of 115 °F. This aligns very closely with the rated UEF of the 4-ref ton unit in Intertek labs. Notably with very high WT setpoint temperature of 135°F and very low OA temperature of 17°F DB the MFHP WH COP is equal to one and therefore the same efficiency as an electric resistance water heater. This very high WT setpoint and very low OA temperature is an unlikely combination for most residential buildings in California climates.

From overnight tank heat loss tests the uniform skin loss coefficient for the WT was measured as 0.73 $W/m^2 °K$ with an area of 2.56 m^2.

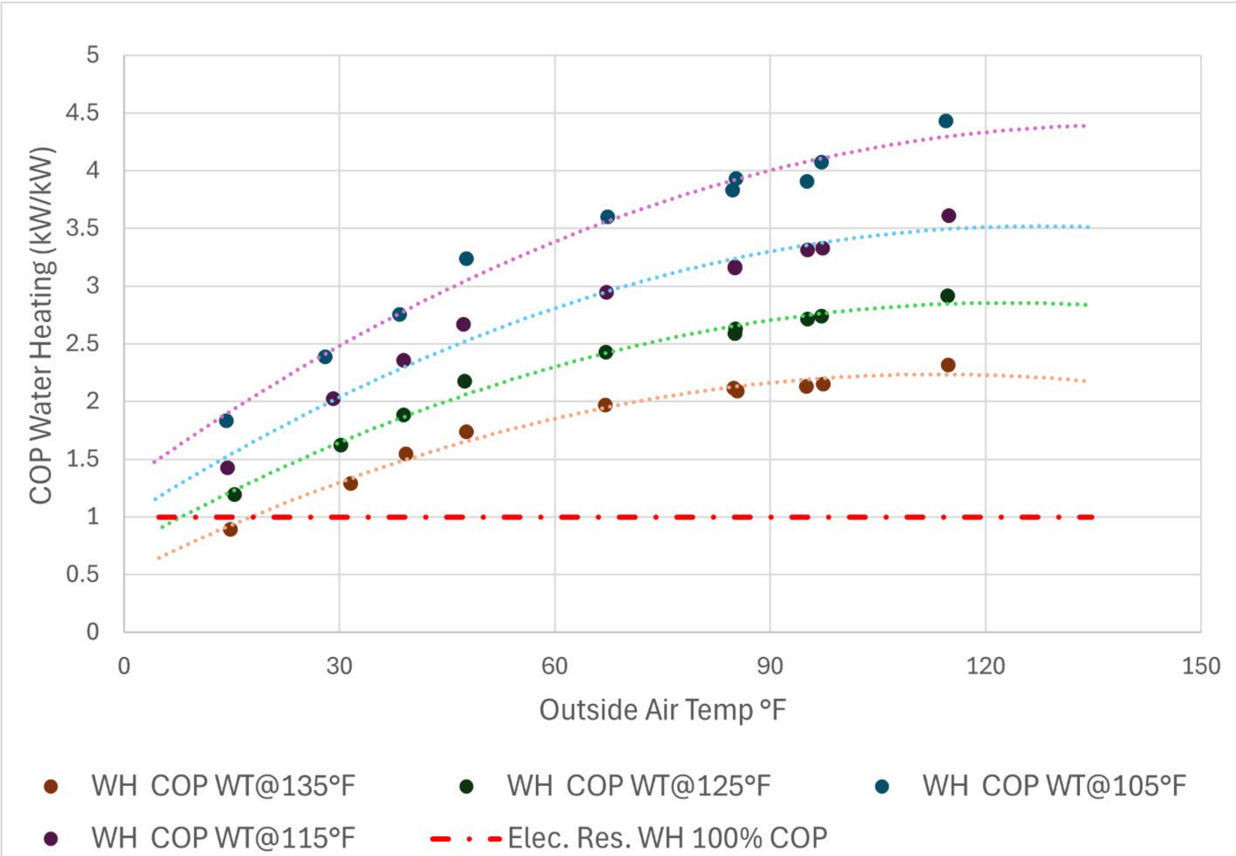

Figure 13. Measured MFHP WH COP at different DB and WT setpoint temperatures. For comparison, the electric resistance heater COP of 1.0 is shown as a dashed red line.

Simultaneous Space Cooling and Water Heating (SIM)

SIM mode cools IA and pushes the thermal energy into the WT, thereby recovering the heat that is typically wasted by HPs. The SC capacity of the SIM mode is shown in Figure 14 to compare with SC mode capacity. The SIM mode capacity is tested for a single outdoor temperature of 95°F DB, since in this mode the MFHP does not intentionally exchange heat with the outdoors. Refrigerant flows between the AHU and WT and bypasses the outdoor unit coil. As seen in Figure 14, the SIM mode has more dependency on the WT temperature than on indoor or return air conditions. This is likely due to the somewhat undersized refrigerant to water heat exchanger coil in the WT, which is likely limiting simultaneous mode cycle efficiency more than the AHU coil. The SIM mode SC capacity was seen to be reduced by about 35% compared to the SC mode at OA temperature of 95°F DB, WT setpoint temperature of 115°F, and IA temperature of 80°F DB.

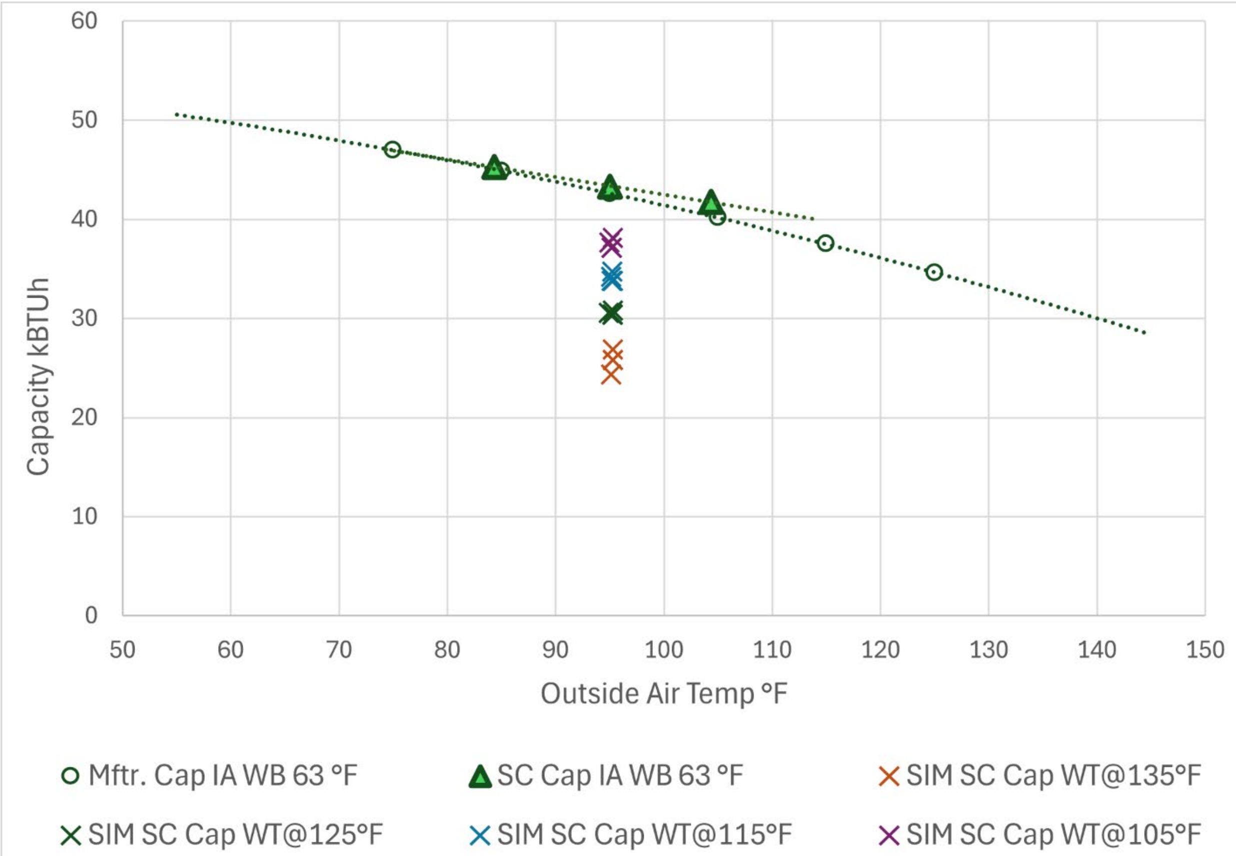

Figure 14. Measured MFHP SIM mode SC capacity at different OA DB temperatures with regular SC mode capacities for comparison.

Water heating capacities of the SIM mode showed very slight drop in comparison to the WH mode at outdoor condition of 95°F DB, Figure 15. Analysis of the refrigerant temperature and pressure measurements at the outlet of the WT refrigerant-to-water heat exchanger shows that the refrigerant not fully condensed for both SIM and WH tests. This means that the refrigerant-to-water heat exchanger is not achieving fast enough heat transfer to keep up with the compressor size. Increasing the refrigerant to water heat transfer capability is expected to increase water heating capacity, increase COP, and increase first hour rating.

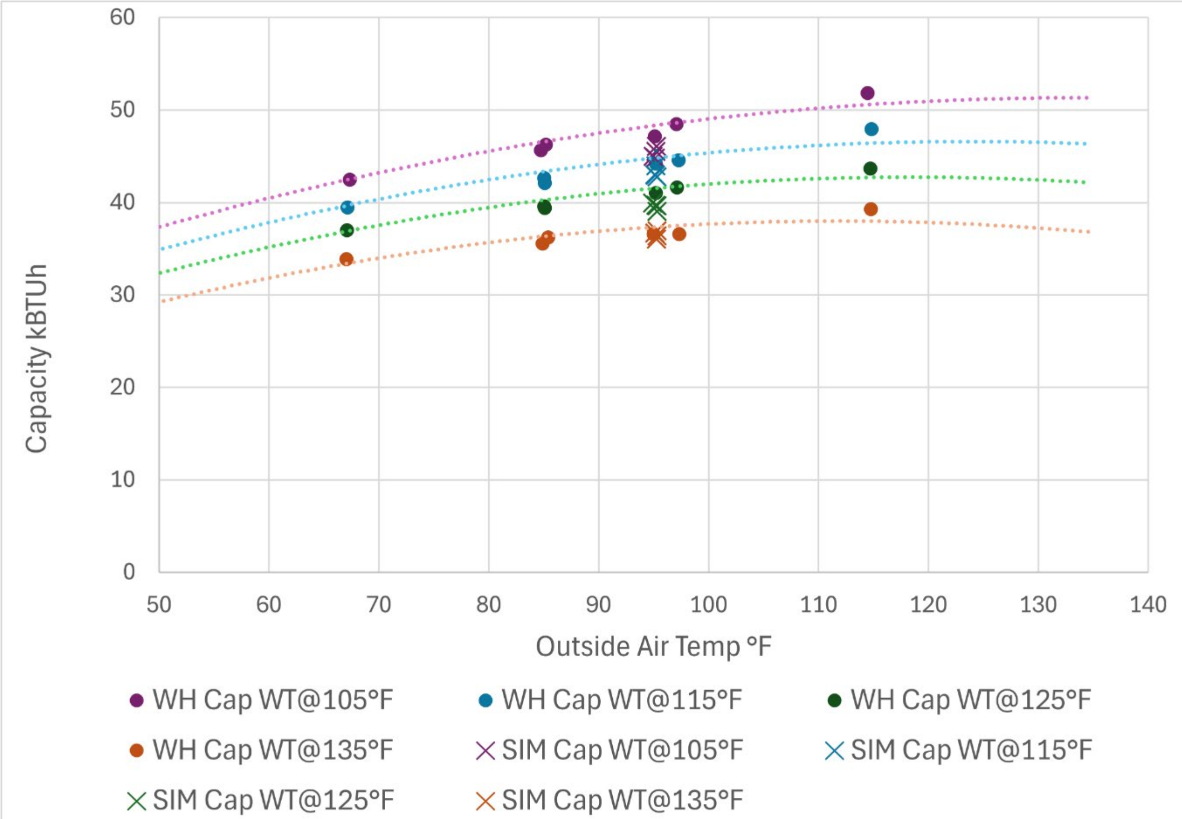

Figure 15. Measured MFHP SIM mode WH capacity at different OA DB and WT setpoint temperatures with regular WH mode capacities for comparison.

The electrical power draw (kW) of the MFHP for each operational mode is compared in Figure 16. The SIM mode only has a small increase in power use compared to the dedicated WH mode, meaning that when using SIM mode is used to heat water, the space cooling takes only a little bit more power. SIM mode power use is typically 5~10% higher than the WH power consumption, with the highest increases coming when the WT and OA temperatures are lower.

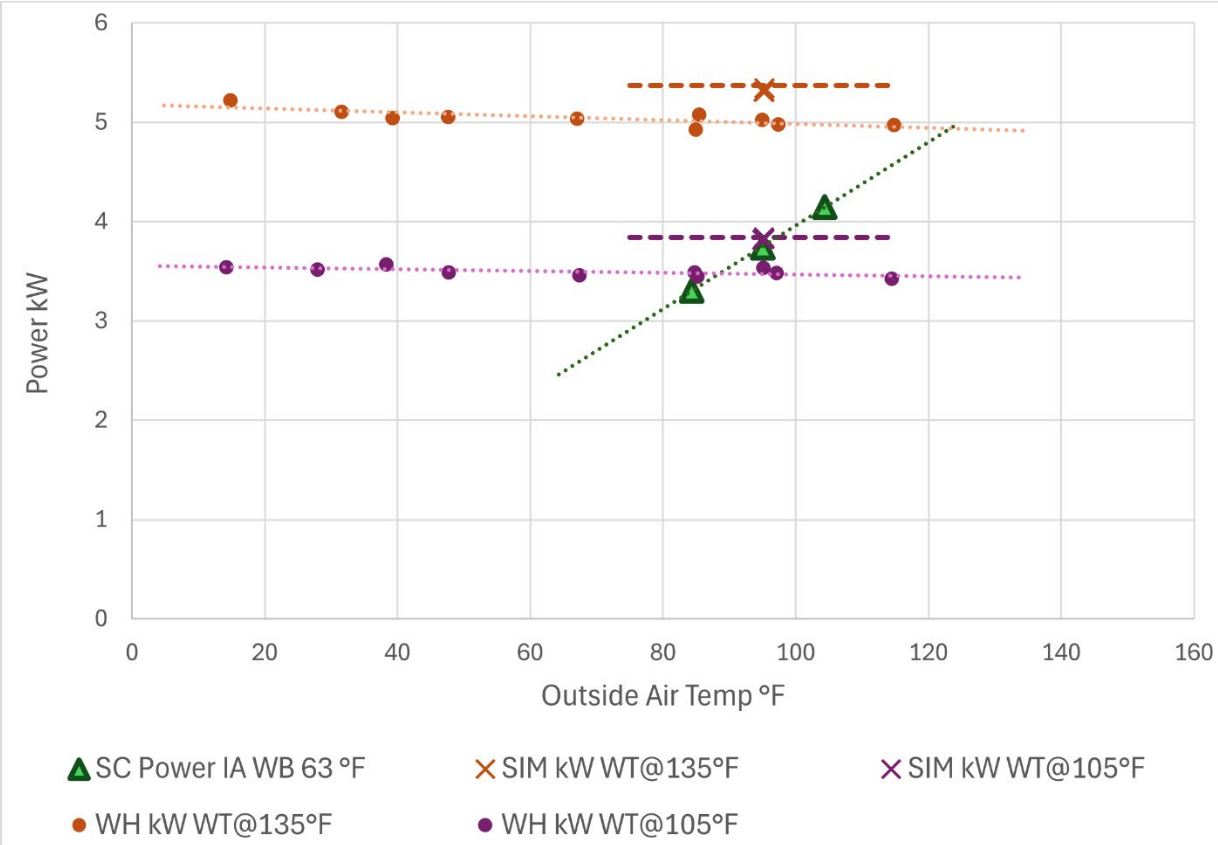

Figure 16. Electric power draw trends for SC, WH, and SIM modes.

To fairly compare the expected power consumption against a given delivered SH and SC capacity, the team used the COPs of the dedicated SC and WH modes to estimate equivalent power consumption to match the capacities (WH and SC) of the SIM mode, assuming a typical IA condition of 75DB for all comparisons. This analysis is performed for the four WT setpoint temperatures and shown in four plots in Figure 17. Like the findings when looking at COP alone, the SIM mode is found to use less power all conditions than providing the same services with separate SC and WH cycles. For a given WT setpoint temperature of 115°F, Figure 17, the SIM mode was found to consume 52% to 70% of the electrical power than two separate WH and SC modes with decreasing OA temperature conditions. This corresponds to an average 38% drop in power consumption than two separate modes across the range of conditions tested.

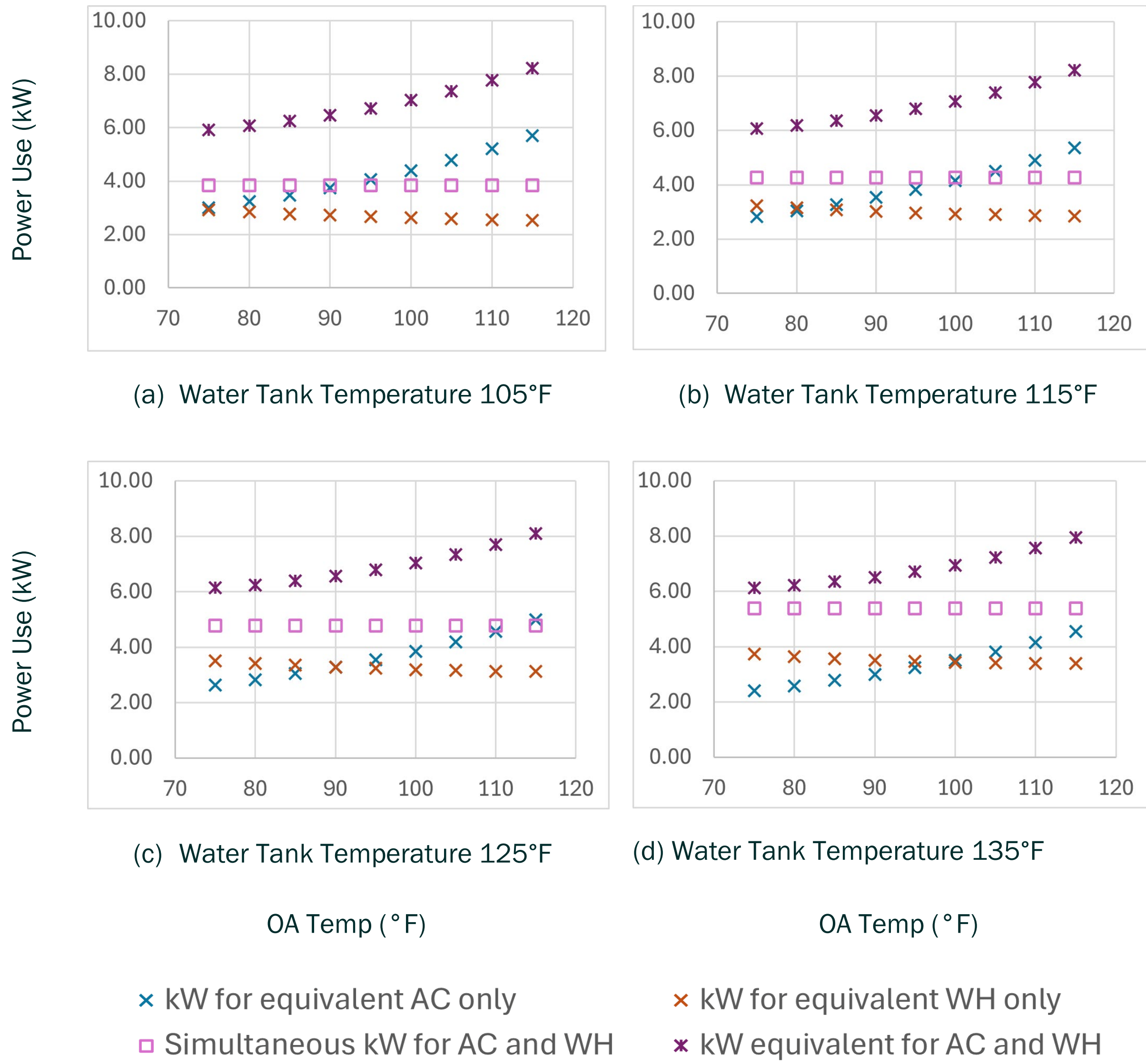

Figure 17. Equivalent power consumption estimate based on individual WH and SC capacities in SIM mode and equivalent SC and WH power consumption for equivalent capacity as scaled by SC and WH mode COPs, respectively.

Defrost (DEF)

To melt accumulated frost on the outdoor coil during a SH or WH cycle, the single-speed MFHP DEF mode reverses the refrigerant flow to transfer heat from the WT to the outdoor coil. The team successfully tested the MFHP in its unique DEF mode, keeping some of the variables and conditions the same. The DEF mode was triggered for WH and SH calls to understand the difference in MFHP operation and interruption created by the DEF mode in both cases.

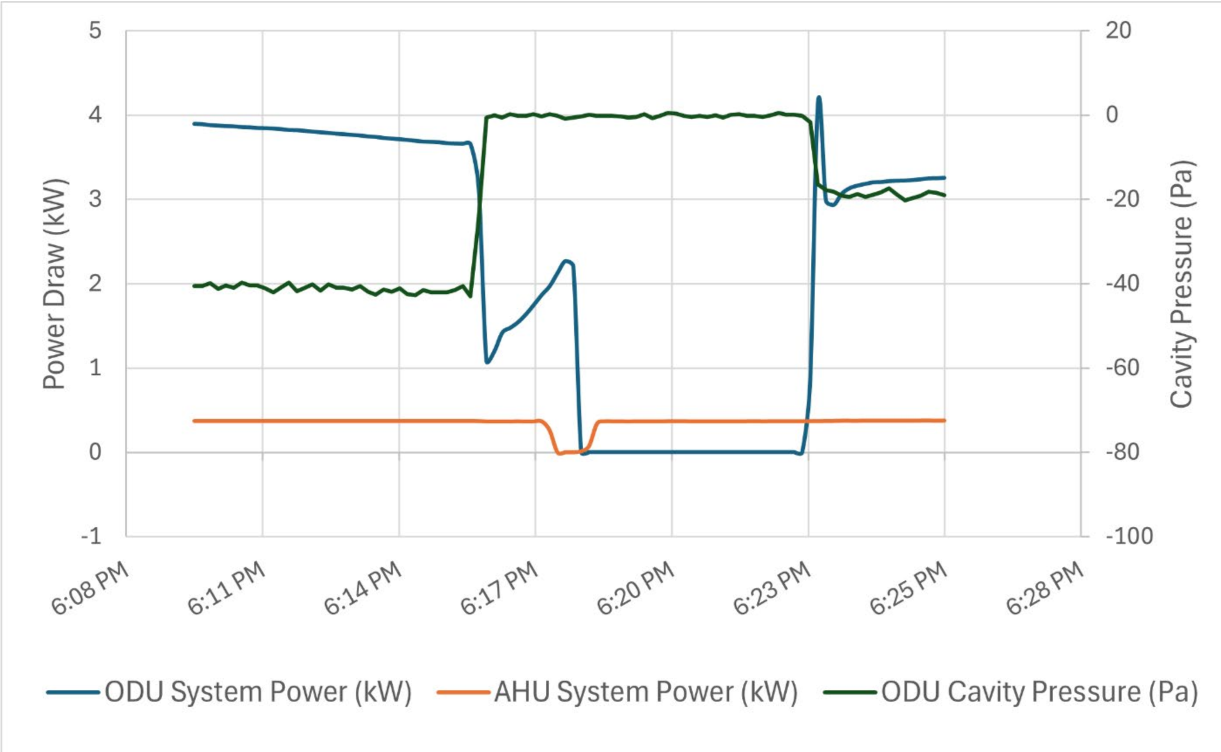

Figure 18. Defrost cycle time-series for WH mode.

Figure 18 shows the DEF mode triggered during WH mode where the ODU system power dropped from 3.9 kW to 1 kW. The DEF mode can be identified when the ODU power is non-zero but the ODU cavity pressure is zero, indicating the ODU fan is turned off. The DEF mode lasted for approximately 1.5 minutes and the ODU power rose steadily with a maximum of just over 2.2 kW as the WT temperature dropped and outdoor coil temperature rose, leading to an increase in compressor lift.

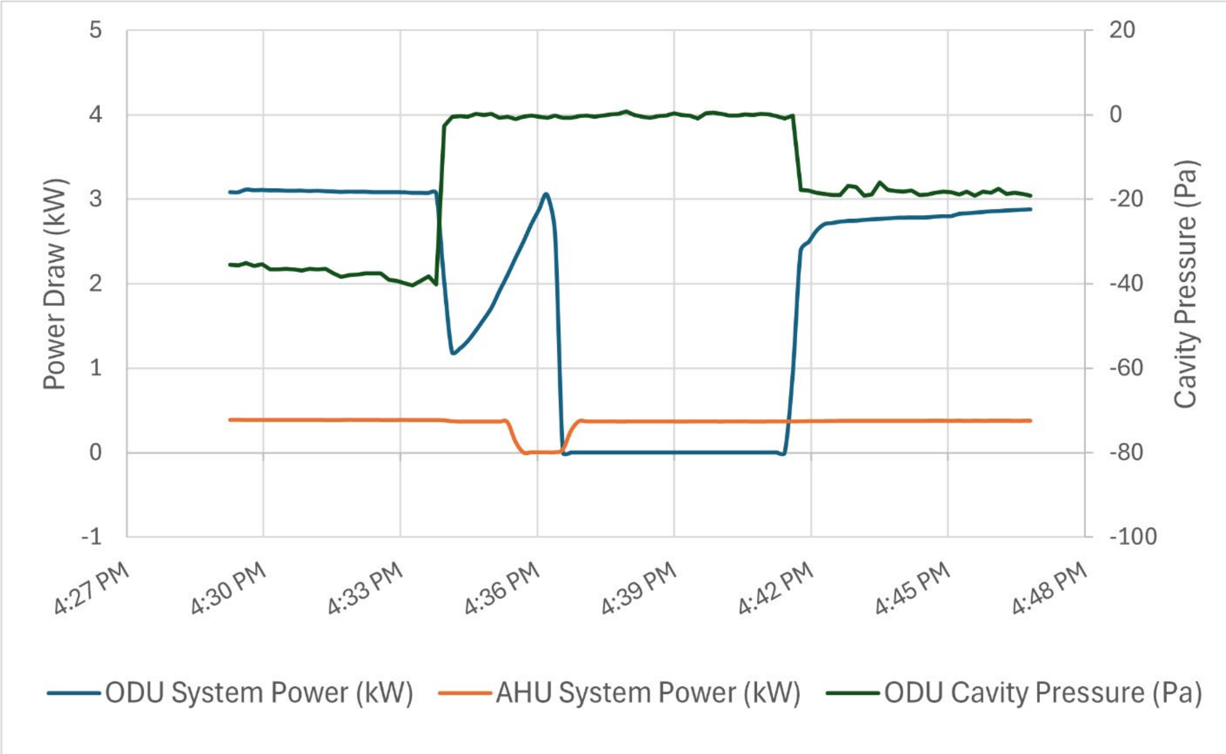

Figure 19. Defrost cycle time-series for SH mode

Similarly, Figure 19 shows the DEF mode triggered during SH mode where the ODU system power dropped from 3 kW to 1.1 kW. The cavity pressure goes to zero, indicating the ODU fan has turned off. The DEF mode lasted for approximately 2 minutes and the ODU power steadily rises as the compressor lift increases with a maximum of just over 3kW. The DEF mode duration is highly dependent on a number of variables when the DEF mode was triggered: the WT temperature, the amount of frost formation, outdoor conditions, and the preceding MFHP mode of operation. The team tested a variety of DEF operations, and the time durations varied from 1 minute to a maximum of 3 minutes, with most cycles lasting for less than 2 minutes. These times were shorter than the 7 to 10 minutes of DEF operation usually observed in single-speed HPs. The overall DEF mode power drawn by the MFHP varies between 1 to 3 kW in comparison to a traditional single-speed HP drawing 5 to 9 kW with electric resistance heaters in the AHU. This means that the total power consumption for DEF mode is significantly lower than for typical split-system single-speed heat pumps and would result in energy savings.

Following the DEF mode for both WH and SH, the team saw that the MFHP went into a lock-out period, usually seen to prevent compressor cycling. This was unusual and usually is not desirable as the manufacturers want to limit the interruption to the service from DEF operation. The team has notified the manufacturer of this single-speed MFHP about this error in DEF logic and they are working to remove this lock-out period in the next software update of their control boards. Once this minor control programming bug is resolved, the shorter DEF cycles for the MFHP will allow it to get back to SH or WH more quickly than typical single-speed split-system heat pumps, with the potential to improve occupant comfort during cold weather.

Water Heating First-Hour Rating Test

For a WT setpoint temperature of 115°F, the team calculated a FHR of 82.0 gallons for the MFHP. Analysis of the data involved isolating the individual water draws performed across the test; mapping the timing of the water draws against the one-hour time frame of the FHR test; calculating the minimum, maximum, and average outlet water temperatures for each draw event; and calculating the total volume expelled from the WT during each draw, which was measured using a flow meter. Figure 20 below shows the cumulative volume of water drawn from the tank over the course of the FHR test.

Upon analyzing the data, the team noticed a delay between when draws were meant to end (based on an observed temperature drop at the water outlet) and when the draws were actually terminated. Due to slight delays in valve actuation and rough estimations made to determine the appropriate draw stopping points during testing, a transitional period was reflected in the data, during which time the test parameters for the MFHP FHR test deviated slightly from the conditions outlined in the test standard. Regardless, the full water draws for which the outlet flow rate fell within the bounds required for the test (3.0 ± 0.25 GPM) was counted in calculating the FHR for the MFHP, as the total output volume is the most important metric in appropriately representing the DHW delivery potential of the MFHP water heater.

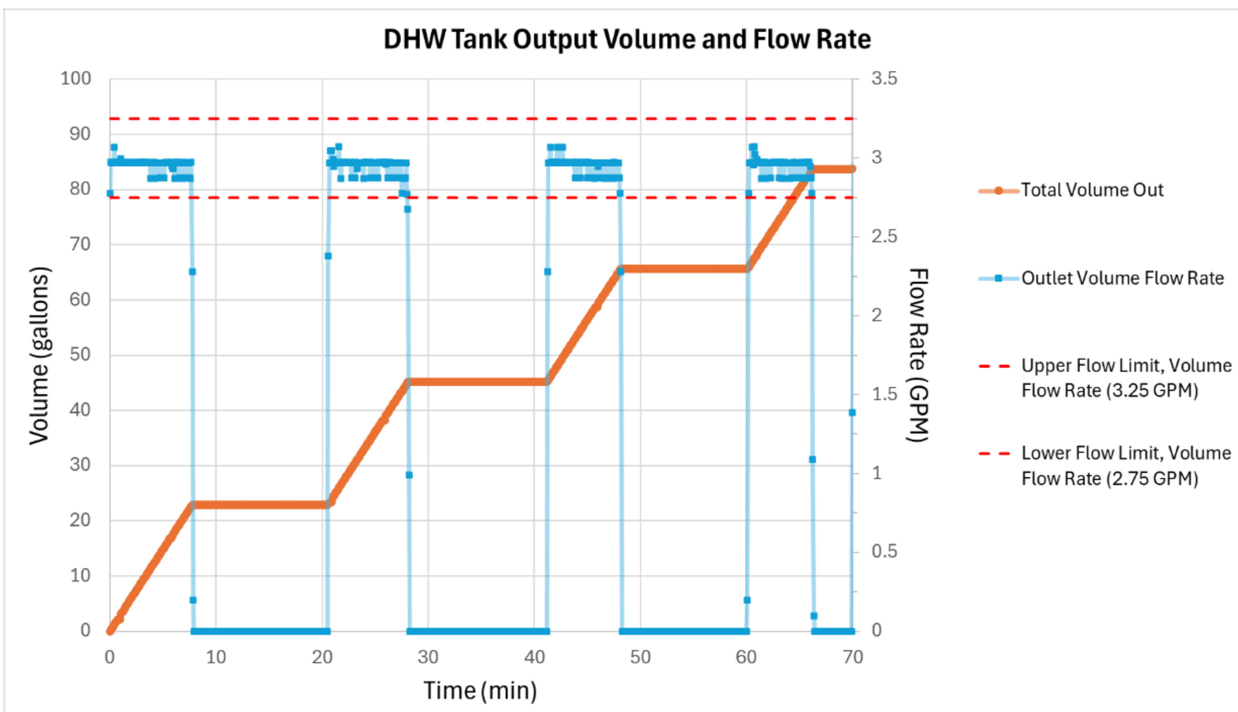

Figure 20. FHR test time-series showing water flow rate and cumulative output water volume.

Equipment Performance Curves for EnergyPlus

The lab test data was used to generate regression curves to estimate the single-speed MFHP performance in EnergyPlus or CBECC-Res. Figure 21 shows an example performance curve estimating SC capacity for any input conditions.

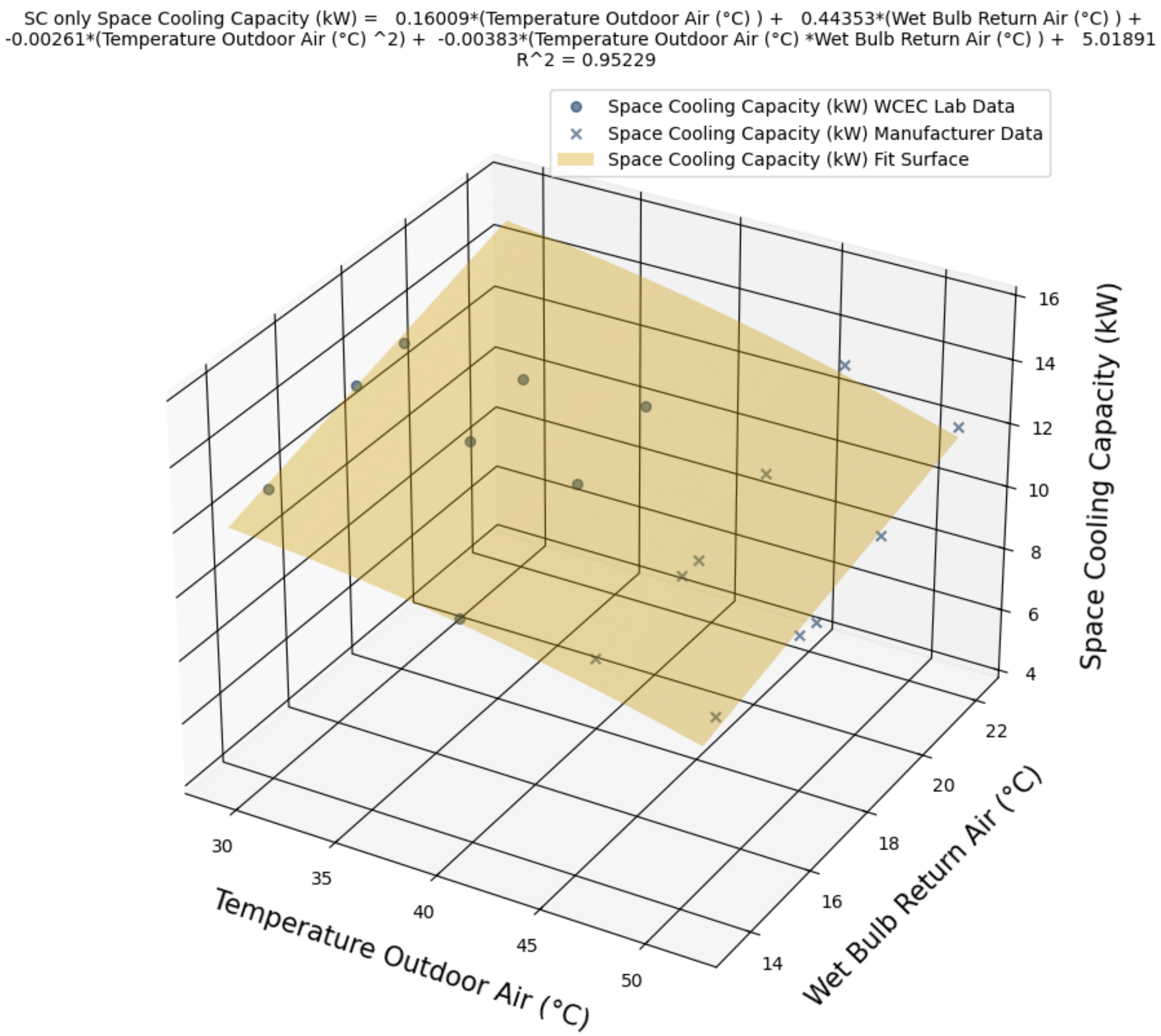

Figure 21. SC mode least squares biquadratic surface fit for temperature trend modeling in EnergyPlus.

These curves are generated by first fitting Lab data results to the test range of input variables using least squares. They are fit using a Biquadratic polynomial of the independent variables and refined to only include the terms needed to adequately characterize the trend of the Lab data being fit. In the example above Lab data points have been augmented with Manufacturer data to generate the expected performance trend for the Space Cooling, $\dot{H}_{SC}$, from the outdoor air dry bulb temperature and indoor return air wet bulb Temperature, $T_{OA,DB}$and $T_{RA,WB}$respectively.

$$\dot{H}_{SC} = \overline{\boldsymbol{C}}_{\boldsymbol{SC,T}} \cdot \begin{bmatrix} T_{OA,DB}^2 & T_{RA,WB}^2 & T_{OA,DB} \cdot T_{RA,WB} & T_{OA,DB} & T_{RA,WB} & 1 \end{bmatrix}$$

This is solved for the linear coefficients, $\overline{\boldsymbol{C}}_{\boldsymbol{SC,T}}$, using least squares, and then used to calculate the value of the curve at the rated condition, $\dot{H}_{SC,rated}$, which for this case is evaluated at $T_{OA,DB} = 95°\mathrm{F}(35°\mathrm{C})$ and $T_{RA,WB} = 67°\mathrm{F}(17.22°\mathrm{C})$. This value is needed to normalize the performance curve to meet the desired input requirements of Energy Plus.

$$\overline{\boldsymbol{C}}_{\boldsymbol{SC,H}} = \begin{bmatrix} 1.6009\mathrm{e}-01 \\ 4.4353\mathrm{e}-01 \\ -2.6125\mathrm{e}-03 \\ 0.0000\mathrm{e}+00 \\ -3.8276\mathrm{e}-03 \\ 5.0189\mathrm{e}+00 \end{bmatrix}$$

$$\dot{H}_{SC,rated} = \dot{H}_{SC} \,@\, \begin{matrix} T_{OA,DB}: 35.00°\mathrm{C} \\ T_{RA,WB}: 17.22°\mathrm{C} \end{matrix} = \mathbf{1.2752e+01}\ \boldsymbol{kW}$$

Using this to nomalize the curve with the relationships:

$$\dot{H}_{SC,norm} = \frac{\dot{H}_{SC}}{\dot{H}_{SC,rated}}$$

And:

$$\overline{\boldsymbol{C}}_{\boldsymbol{SC,T}} = \dot{H}_{SC,rated} \times \overline{\boldsymbol{C}}$$

Such that:

$$\dot{H}_{SC,norm} = \overline{\boldsymbol{C}} \cdot \begin{bmatrix} T_{OA,DB}^2 & T_{RA,WB}^2 & T_{OA,DB} \cdot T_{RA,WB} & T_{OA,DB} & T_{RA,WB} & 1 \end{bmatrix}$$

$$\overline{\boldsymbol{C}} = \overline{\boldsymbol{C}}_{\boldsymbol{SC,H}} / \dot{H}_{SC,rated} = \begin{bmatrix} \mathbf{1.2554e-02} \\ \mathbf{3.4780e-02} \\ \mathbf{-2.0486e-04} \\ \mathbf{0.0000e+00} \\ \mathbf{-3.0015e-04} \\ \mathbf{3.9357e-01} \end{bmatrix}$$

Energy Plus, with the entered values of $\dot{H}_{SC,rated}$ and $\overline{\boldsymbol{C}}$, can model Space Cooling mode operation calculating Capacity of the unit given various $T_{OA,DB}$and $T_{RA,WB}$conditions. Along with other fits outlined in **Appendix A**, for EIR and SHR, Energy Plus can model Electrical Power use Supply air Temperature and Humidity.

Stakeholder Engagement

The team engaged relevant stakeholders as part of the original project idea development, in the CalNEXT proposal scoring process, and through ongoing collaboration and input solicitation.

Equipment Manufacturer

The most direct stakeholder is the MFHP equipment manufacturer, who provided the equipment and worked collaboratively with the WCEC to support successful testing. The MFHP manufacturer has advocated for CEC CBECC-Res to include accurate modeling of MFHPs. This project generated performance curves for this equipment, which can be used by EnergyPlus and CBECC-Res to simulate energy savings.

Other HVAC manufacturers are also stakeholders of this project, including the manufacturer of the residential split-system HP product that Villara modifies to make the AquaThermAire, and manufacturers planning to offer MFHP products including LG, Samsung, and Panasonic. The WCEC team previously completed a field test of a production ready prototype of the AquaThermAire and is currently field-testing advanced load management controls of this equipment at three field sites. The results of these field tests have been presented at the IEA Heat Pump Conference, May 2023 (Chally, Haile and Chakraborty, et al. 2023) and load management results have been presented at California Load Flexibility Hub Symposium (Chakraborty, Integrated Heat Pump with Storage for DHW and Space Conditioning 2022). The field test results have been presented in a presentation and poster at a private WCEC-organized event for HVAC manufacturers and utility efficiency program providers, held in parallel with the 2024 ASHRAE Winter Conference and AHR Expo in Chicago. Attendees included HVAC manufacturers Daikin, Delta Controls, Geary Pacific Corp, MicroMetl, Panasonic, Rheem, Seeley, and Trane, as well as EE program implementers Leidos and Amren, the National Energy Management Institute (NEMI), the Sheet Metal and Air Conditioning Contractors' National Association (SMACNA), and engineering consulting firms TRC and Emanant Systems. WCEC recently participated in a local news television feature about residential building electrification where we discussed the benefits of MFHP products (Chakraborty 2024).

CalNEXT Team

Key stakeholders for this project also include the teams that will use EnergyPlus, CBECC-Res, and other energy modeling tools to estimate energy savings of the MFHP equipment and, in doing so, develop a new measure package for deemed efficiency programs. These teams include WCEC and, potentially, other CalNEXT partners such as Energy Solutions, TRC, and Alternative Energy Systems Consulting, Inc. (AESC). Energy Solutions, TRC, and AESC have provided feedback on this project, beginning in the idea stage and throughout development of the project plan. The UC Davis team has engaged with CalNEXT team members to prepare the team to do the next step measure development. WCEC will contribute to – or lead –the measure development efforts, to ensure that the performance curves accurately represent the real equipment efficiency in the required modelling tools to predict energy savings.

IOUs

IOUs are additional key stakeholders for development and adoption of new MFHP efficiency measures for EE programs. In preparation for the next step measure development project, the team has reached out to and is setting up meetings with the California HVAC program administrator San Diego Gas & Electric (SDG&E) to determine what standards and requirements the equipment needs to meet to be included in the program as a new efficiency measure. WCEC has engaged – and will continue to engage – with IOU representatives to share information about MFHP equipment and to learn how this new class of products can enable efficiency-program designs to overcome residential

HP adoption barriers. The UC Davis team shared the field test results with utility stakeholders through multiple presentations – including to Southern California Edison (SCE), PG&E, SDG&E – at the UC Davis Energy and Efficiency Institute June 13th, 2024 board of advisors meeting and to SCE efficiency programs staff November 6th, 2023. Multiple IOU representatives expressed interest in the MFHP technology for their service areas. The UC Davis team has contacted SDG&E to map out next steps towards measure development.

CalTF

The California Technical Forum (CalTF) is a key stakeholder for the next steps measure development process. The team has reached out to and is scheduling a time to present the MFHP lab and field test results to the CalTF Deemed Initiative Subcommittee to begin this process.

The UC Davis team will continue to engage with the stakeholders identified above, including HVAC manufacturers, energy modelling tool developers, CalTF, and IOU staff. The UC Davis team will continue to recruit and prepare the teams to perform next step measure development for the technology, including other CalNEXT team members, to maximize the impact of this project. Table 7 lists the stakeholders the team has engaged with during this project, specifically and more broadly, around MFHP products.

Table 7. Stakeholder Engagement

Organization Type	Organization
Energy Modelling Tool Developer	CEC CBECC-RES
Efficiency Measure Evaluator	CalTF
IOU	SDG&E
IOU	SCE
IOU	PG&E
CalNEXT Partners	TRC
CalNEXT Partners	AESC
CalNEXT Partners	Energy Solutions
CalNEXT Partners	The Ortiz Group

Organization Type	Organization
CalNEXT Partners	VEIC
Manufacturer	Villara
Manufacturer	Carrier

Stakeholder Feedback

In a March 15, 2024, discussion with Vice President of Customer Programs and Services at SCE, mentioned that SCE is very interested in efficiency measures that also support electrification. This work is targeting EE and cost reductions for electrification of space conditioning and WH.

A Senior Staff Engineer with Energy Solutions, raised the concern that customers who have a new HVAC system or new water heater may not want to replace both systems at the same time with a MFHP.

A Professional HVAC installer from Villara mentioned that multiple customers having separate space conditioning and water heating heat pumps installed have asked him why they have to buy two separate HPs for space conditioning and WH and said that they would like to buy a single HP that would do both.

Recommendations and Next Steps

The project team's next steps include:

- Continue to engage with the California HVAC Program Administrator SDG&E to determine what standards and requirements the equipment needs to meet to be included in the efficiency program as a new efficiency measure. Follow up with the MFHP manufacturer to support meeting those requirements.
- Continue to engage with the CalTF to present the to present the MFHP lab and field test results to the CalTF Deemed Initiative Subcommittee to begin the measure development process
- Continue to recruit and prepare the team to do the next step measure development. WCEC will contribute to, or lead, the measure development efforts to ensure that the performance curves can represent equipment efficiency in the required modelling tools to predict energy savings.
- Continue to engage with EnergyPlus and CEC CBECC-Res software developers to promote the use of the performance curves for cost-benefit analysis and for code compliance.

Future laboratory tests of one or more competing air-to-air MFHP products will evaluate variable-speed MFHP equipment performance. Future field demonstration(s) are being planned with the potential to leverage the three existing field sites or establish new sites to evaluate typical real-world energy performance, installation costs, and any unforeseen barriers to adoption. A research home investigation is planned to compare energy savings between air-to-air and air-to-water versions of MFHP in a real building with repeatable setpoints, internal loads, and hot water draws. Each of these steps will include products that are a good fit for older residential buildings that are often in DACs with HTR residents.

Existing programs including deemed and custom efficiency programs as well as TECH Clean California have multiple measures for retrofitting space conditioning HPs and stand-alone HPWHs. Efficiency measures using MFHPs can build from and combine the existing measures while using the performance curves generated by this project to update the energy savings. MFHPs have the potential to achieve more cost-effective energy savings than separate space conditioning and WH

HPs because MFHPs enable reductions in retrofit costs along with high efficiency operation and utilization of waste heat for hot water heating. MFHP measures can utilize many of the existing HP program design elements.

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Appendix A

Performance curve development and details for modeling.

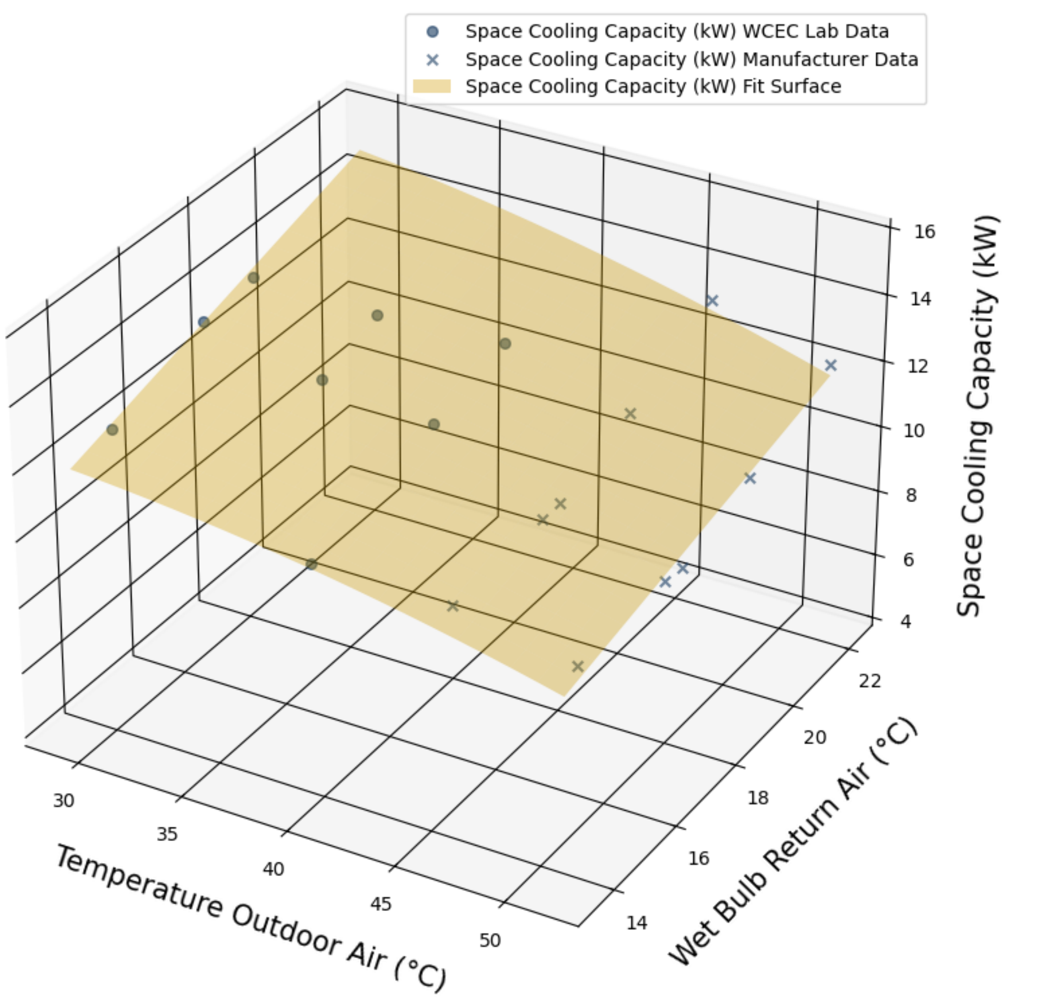
Results and Documentation from Performance Curve Generation

Performance curves developed from the laboratory test data and formatted for entry into Energy Plus. The **Variables in Bold** are the intended entry values.

Space Cooling Modeling Fits

Only for the space cooling mode, regression curve fits using lab test data alone had a lower/flatter slope than manufacturer data which could overpredict COP at extremely high outdoor temperatures and lead to overestimates of Total System Benefit. For this reason, the project team included manufacturer published performance data for 115°F and 125°F outside temperatures to influence the curvature of the modeling fits and avoid over predicting COP.

SC only Space Cooling Capacity (kW) = 0.16009*(Temperature Outdoor Air (°C)) + 0.44353*(Wet Bulb Return Air (°C)) + -0.00261*(Temperature Outdoor Air (°C) ^2) + -0.00383*(Temperature Outdoor Air (°C) *Wet Bulb Return Air (°C)) + 5.01891 : R^2 = 0.95229



Capacity:

$$\dot{H}_{SC} = \overline{\boldsymbol{C}}_{\boldsymbol{SC,H}} \cdot \begin{bmatrix} T_{OA,DB} & T_{RA,WB} & T_{OA,DB}^2 & T_{RA,WB}^2 & T_{OA,DB} \cdot T_{RA,WB} & 1 \end{bmatrix}$$

Linear Least Squares Coefficients and coefficient of determination:

$$\overline{\boldsymbol{C}}_{\boldsymbol{SC,H}} = \begin{bmatrix} 1.6009\mathrm{e}-01 \\ 4.4353\mathrm{e}-01 \\ -2.6125\mathrm{e}-03 \\ 0.0000\mathrm{e}+00 \\ -3.8276\mathrm{e}-03 \\ 5.0189\mathrm{e}+00 \end{bmatrix}$$

$$R^2 = 9.5229\mathrm{e}-01 \approx 95.2\%$$

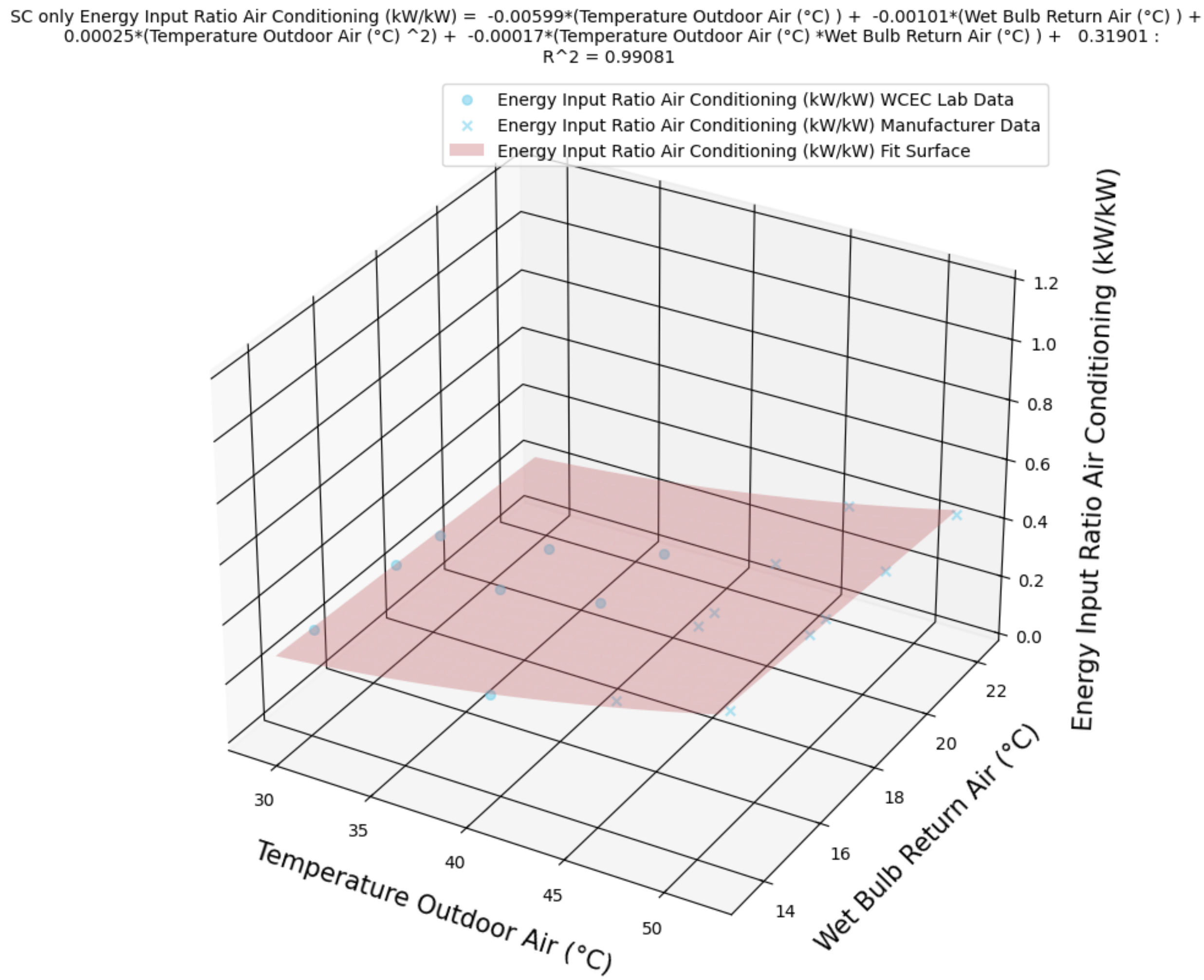
Rated and normalized Capacities:

$$\dot{H}_{SC,rated} = \dot{H}_{SC} @ \begin{matrix} T_{OA,DB}: 35.00°\mathrm{C} \\ T_{RA,WB}: 17.22°\mathrm{C} \end{matrix} = \mathbf{1.2752e+01}\ \boldsymbol{kW}$$

$$\dot{H}_{SC,norm} = \overline{\boldsymbol{C}} \cdot \begin{bmatrix} T_{OA,DB} & T_{RA,WB} & T_{OA,DB}^2 & T_{RA,WB}^2 & T_{OA,DB} \cdot T_{RA,WB} & 1 \end{bmatrix}$$

Normalized coefficients:

$$\overline{\boldsymbol{C}} = \overline{\boldsymbol{C}}_{\boldsymbol{SC,H}} / \dot{H}_{SC,rated} = \begin{bmatrix} \mathbf{1.2554e-02} \\ \mathbf{3.4780e-02} \\ \mathbf{-2.0486e-04} \\ \mathbf{0.0000e+00} \\ \mathbf{-3.0015e-04} \\ \mathbf{3.9357e-01} \end{bmatrix}$$



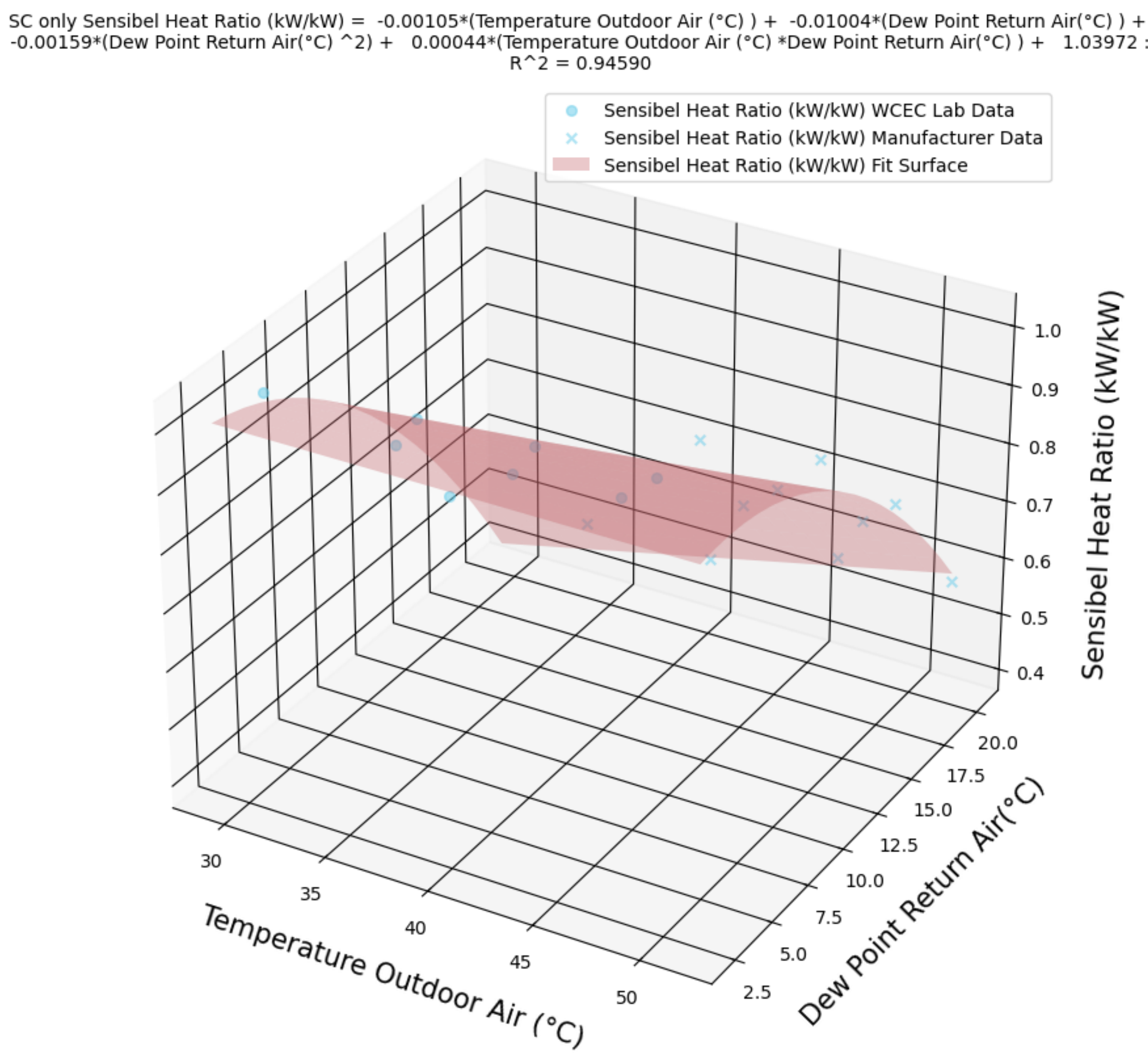
Energy Input Ratio (inverse of Coefficient of Performance):

$$EIR_{SC} = \frac{1}{COP_{SC}} = \overline{\boldsymbol{C}}_{\boldsymbol{SC,E}} \cdot \begin{bmatrix} T_{OA,DB} & T_{RA,WB} & T_{OA,DB}^2 & T_{RA,WB}^2 & T_{OA,DB} \cdot T_{RA,WB} & 1 \end{bmatrix}$$

Linear Least Squares Coefficients and coefficient of determination:

$$\overline{\boldsymbol{C}}_{\boldsymbol{SC,E}} = \begin{bmatrix} \mathbf{-5.9943e-03} \\ \mathbf{-1.0068e-03} \\ \mathbf{2.4630e-04} \\ \mathbf{0.0000e+00} \\ \mathbf{-1.6933e-04} \\ \mathbf{3.1901e-01} \end{bmatrix}$$

$$R^2 = 9.9081\text{e}-01 \approx 99.1\%$$



Sensible Heat Ratio:

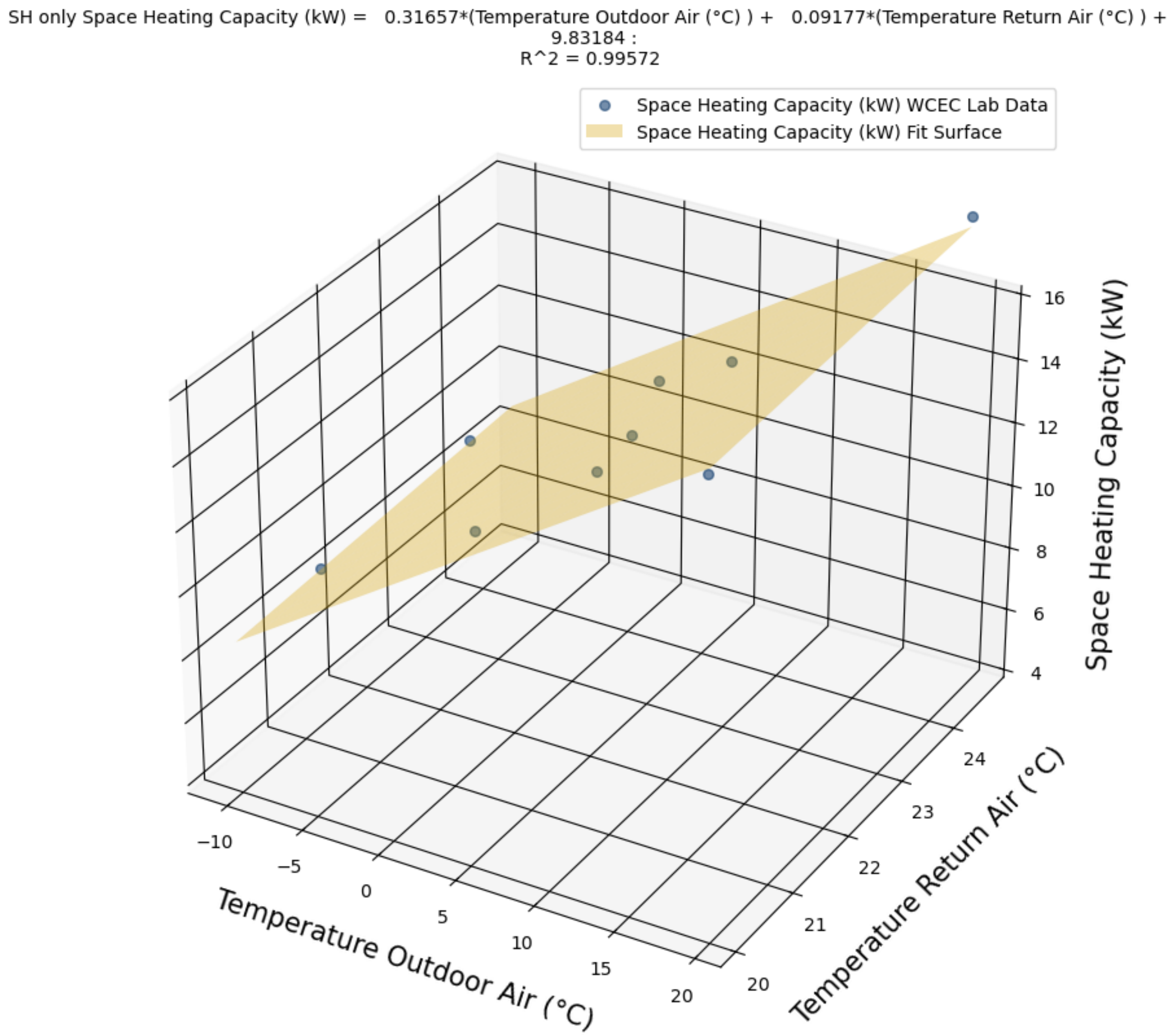
$$SHR_{SC} = \overline{\boldsymbol{C}}_{\boldsymbol{SC,R}} \cdot \begin{bmatrix} T_{OA,DB} & T_{RA,DP} & T_{OA,DB}^2 & T_{RA,DP}^2 & T_{OA,DB} \cdot T_{RA,DP} & 1 \end{bmatrix}$$

Linear Least Squares Coefficients and coefficient of determination:

$$\overline{\boldsymbol{C}}_{\boldsymbol{SC,R}} = \begin{bmatrix} \mathbf{-1.0456e-03} \\ \mathbf{-1.0044e-02} \\ \mathbf{0.0000e+00} \\ \mathbf{-1.5860e-03} \\ \mathbf{4.3855e-04} \\ \mathbf{1.0397e+00} \end{bmatrix}$$

$$R^2 = 9.4590\text{e} - 01 \approx 94.6\%$$

Space Heating Modeling Fits



Capacity:

$$\dot{H}_{SH} = \overline{\boldsymbol{C}}_{\boldsymbol{SH,H}} \cdot \begin{bmatrix} T_{OA,DB} & T_{RA,DB} & T_{OA,DB}^2 & T_{RA,DB}^2 & T_{OA,DB} \cdot T_{RA,DB} & 1 \end{bmatrix}$$

Linear Least Squares Coefficients and coefficient of determination:

$$\overline{\boldsymbol{C}}_{\boldsymbol{SH,H}} = \begin{bmatrix} 3.1657\mathrm{e}-01 \\ 9.1767\mathrm{e}-02 \\ 0.0000\mathrm{e}+00 \\ 0.0000\mathrm{e}+00 \\ 0.0000\mathrm{e}+00 \\ 9.8318\mathrm{e}+00 \end{bmatrix}$$

$$R^2 = 9.9572\text{e}-01 \approx 99.6\%$$

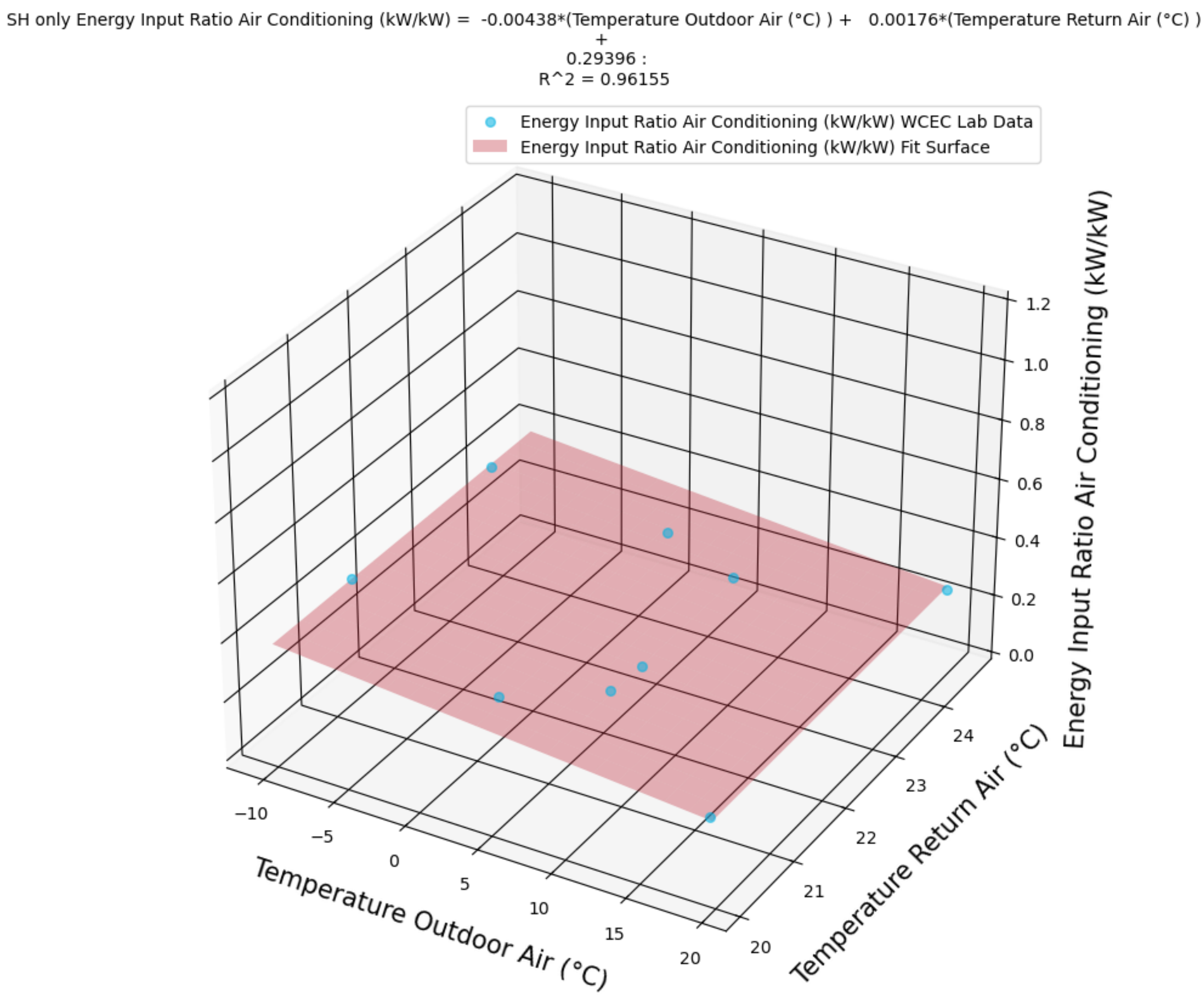
Rated and normalized Capacities:

$$\dot{H}_{SH,rated} = \dot{H}_{SH} @_{T_{RA,DB}:21.11°C}^{T_{OA,DB}:8.33°C} = \mathbf{1.4406e+01}\ \boldsymbol{kW}$$

$$\dot{H}_{SH,norm} = \overline{\boldsymbol{C}} \cdot \begin{bmatrix} T_{OA,DB} & T_{RA,DB} & T_{OA,DB}^2 & T_{RA,DB}^2 & T_{OA,DB} \cdot T_{RA,DB} & 1 \end{bmatrix}$$

Normalized coefficients:

$$\overline{\boldsymbol{C}} = \overline{\boldsymbol{C}}_{\boldsymbol{HC,H}} / \dot{H}_{SH,rated} = \begin{bmatrix} \mathbf{2.1975e-02} \\ \mathbf{6.3700e-03} \\ \mathbf{0.0000e+00} \\ \mathbf{0.0000e+00} \\ \mathbf{0.0000e+00} \\ \mathbf{6.8248e-01} \end{bmatrix}$$



Energy Input Ratio (inverse of Coefficient of Performance):

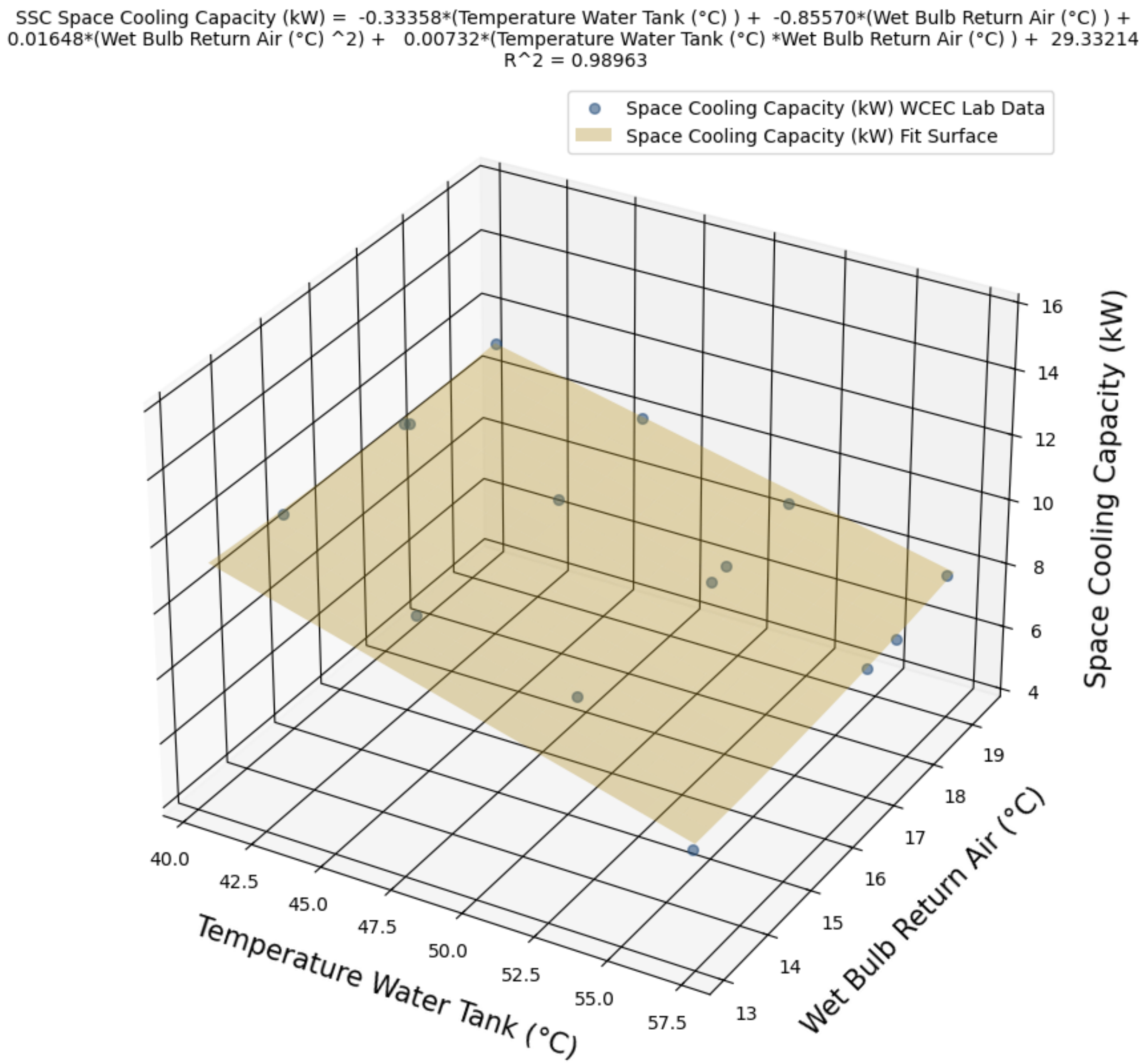
$$EIR_{SH} = \frac{1}{COP_{SH}} = \overline{\boldsymbol{C}}_{\boldsymbol{SH,E}} \cdot \begin{bmatrix} T_{OA,DB} & T_{RA,DB} & T_{OA,DB}^2 & T_{RA,DB}^2 & T_{OA,DB} \cdot T_{RA,DB} & 1 \end{bmatrix}$$

Linear Least Squares Coefficients and coefficient of determination:

$$\overline{\boldsymbol{C}}_{\boldsymbol{SC,E}} = \begin{bmatrix} \mathbf{-4.3799e-03} \\ \mathbf{1.7575e-03} \\ \mathbf{0.0000e+00} \\ \mathbf{0.0000e+00} \\ \mathbf{0.0000e+00} \\ \mathbf{2.9396}\boldsymbol{e}\mathbf{-01} \end{bmatrix}$$

$$R^2 = 9.6155\text{e} - 01 \approx 96.2\%$$

Simultaneous Space Cooling Modeling Fits (operates along with SWH mode)



Capacity:

$$\dot{H}_{SC} = \overline{\boldsymbol{C}}_{\boldsymbol{SSC,H}} \cdot \begin{bmatrix} T_{WTH} & T_{RA,WB} & T_{WTH}^2 & T_{RA,WB}^2 & T_{WTH} \cdot T_{RA,WB} & 1 \end{bmatrix}$$

Linear Least Squares Coefficients and coefficient of determination:

$$\overline{\boldsymbol{C}}_{\boldsymbol{SSC,H}} = \begin{bmatrix} -3.3358\text{e}-01 \\ -8.5570\text{e}-01 \\ 0.0000\text{e}+00 \\ 1.6484\text{e}-02 \\ 7.3193\text{e}-03 \\ 2.9332\text{e}+01 \end{bmatrix}$$

$$R^2 = 9.8963\text{e} - 01 \approx 99.0\%$$

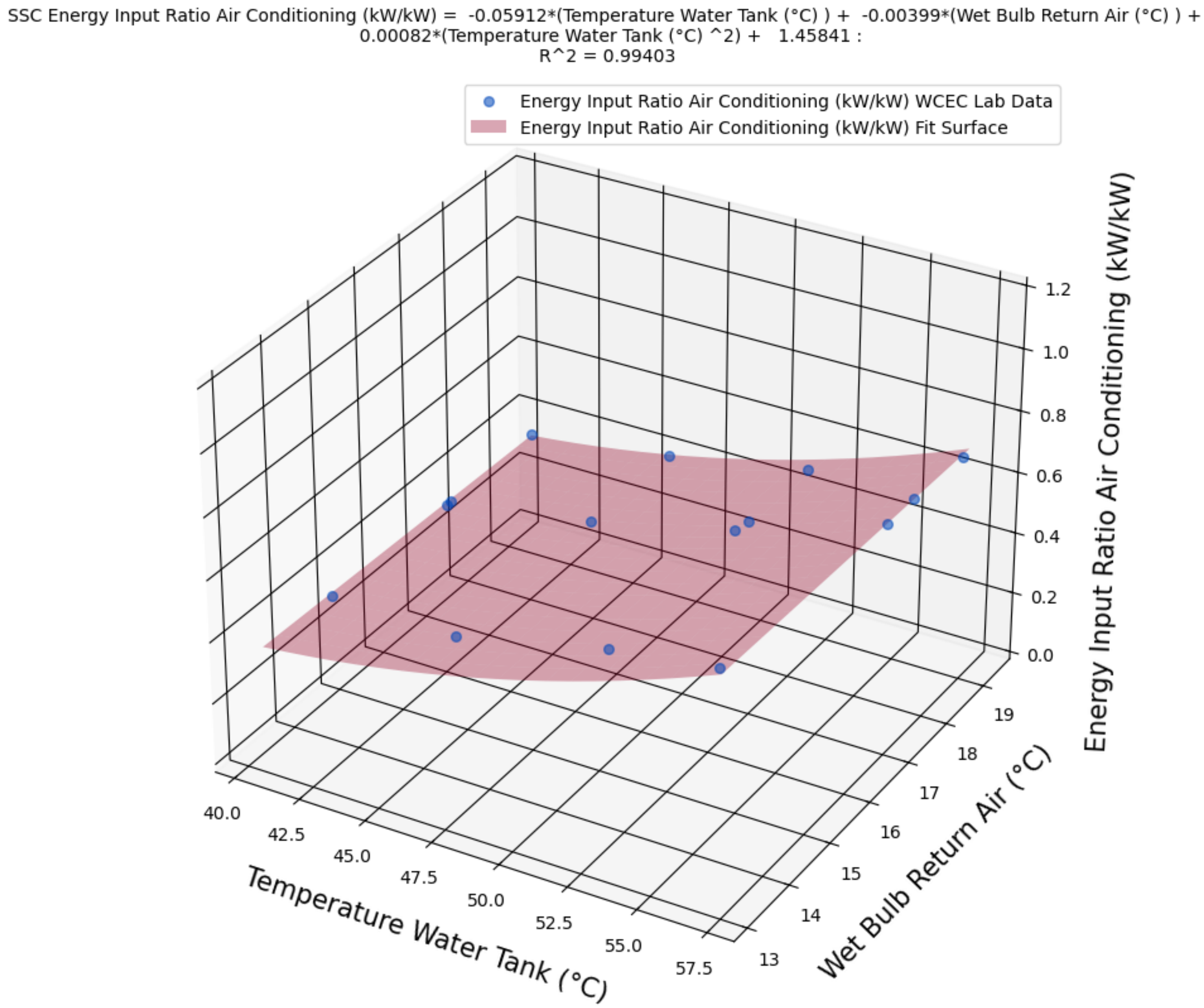
Rated and normalized Capacities:

$$\dot{H}_{SSC,rated} = \dot{H}_{SSC} @_{T_{RA,WB}: 17.22°\text{C}}^{T_{WTH}: 51.67°\text{C}} = \mathbf{8.7611e + 00}\ \boldsymbol{kW}$$

$$\dot{H}_{SSC,norm} = \overline{\boldsymbol{C}} \cdot \begin{bmatrix} T_{WTH} & T_{RA,WB} & T_{WTH}^2 & T_{RA,WB}^2 & T_{WTH} \cdot T_{RA,WB} & 1 \end{bmatrix}$$

Normalized coefficients:

$$\overline{\boldsymbol{C}} = \overline{\boldsymbol{C}}_{\boldsymbol{SSC,H}} / \dot{H}_{SSC,rated} = \begin{bmatrix} \mathbf{1 - 3.8076e - 02} \\ \mathbf{-9.7670e - 02} \\ \mathbf{0.0000e + 00} \\ \mathbf{1.8815e - 03} \\ \mathbf{8.3544e - 04} \\ \mathbf{3.3480e + 00} \end{bmatrix}$$



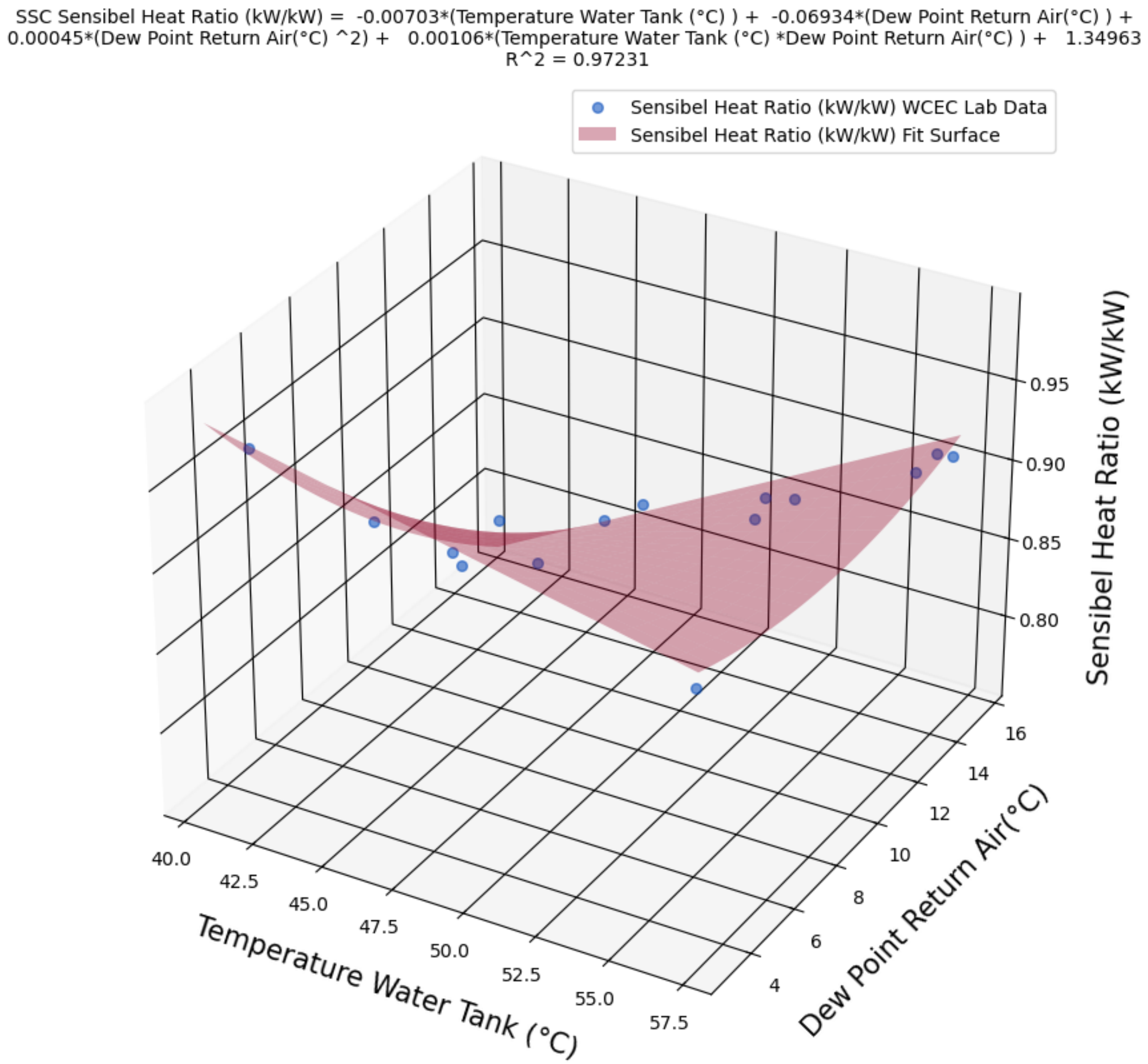
Energy Input Ratio (inverse of Coefficient of Performance):

$$EIR_{SSC} = \frac{1}{COP_{SSC}} = \overline{\boldsymbol{C}}_{\boldsymbol{SSC,E}} \cdot \begin{bmatrix} T_{WTH} & T_{RA,WB} & T_{WTH}^2 & T_{RA,WB}^2 & T_{WTH} \cdot T_{RA,WB} & 1 \end{bmatrix}$$

Linear Least Squares Coefficients and coefficient of determination:

$$\overline{\boldsymbol{C}}_{\boldsymbol{SSC,E}} = \begin{bmatrix} \mathbf{-5.9119e-02} \\ \mathbf{-3.9948e-03} \\ \mathbf{8.2459e-04} \\ \mathbf{0.0000e+00} \\ \mathbf{0.0000e+00} \\ \mathbf{1.4584e+00} \end{bmatrix}$$

$$R^2 = 9.9403\text{e}-01 \approx 99.4\%$$



Sensible Heat Ratio:

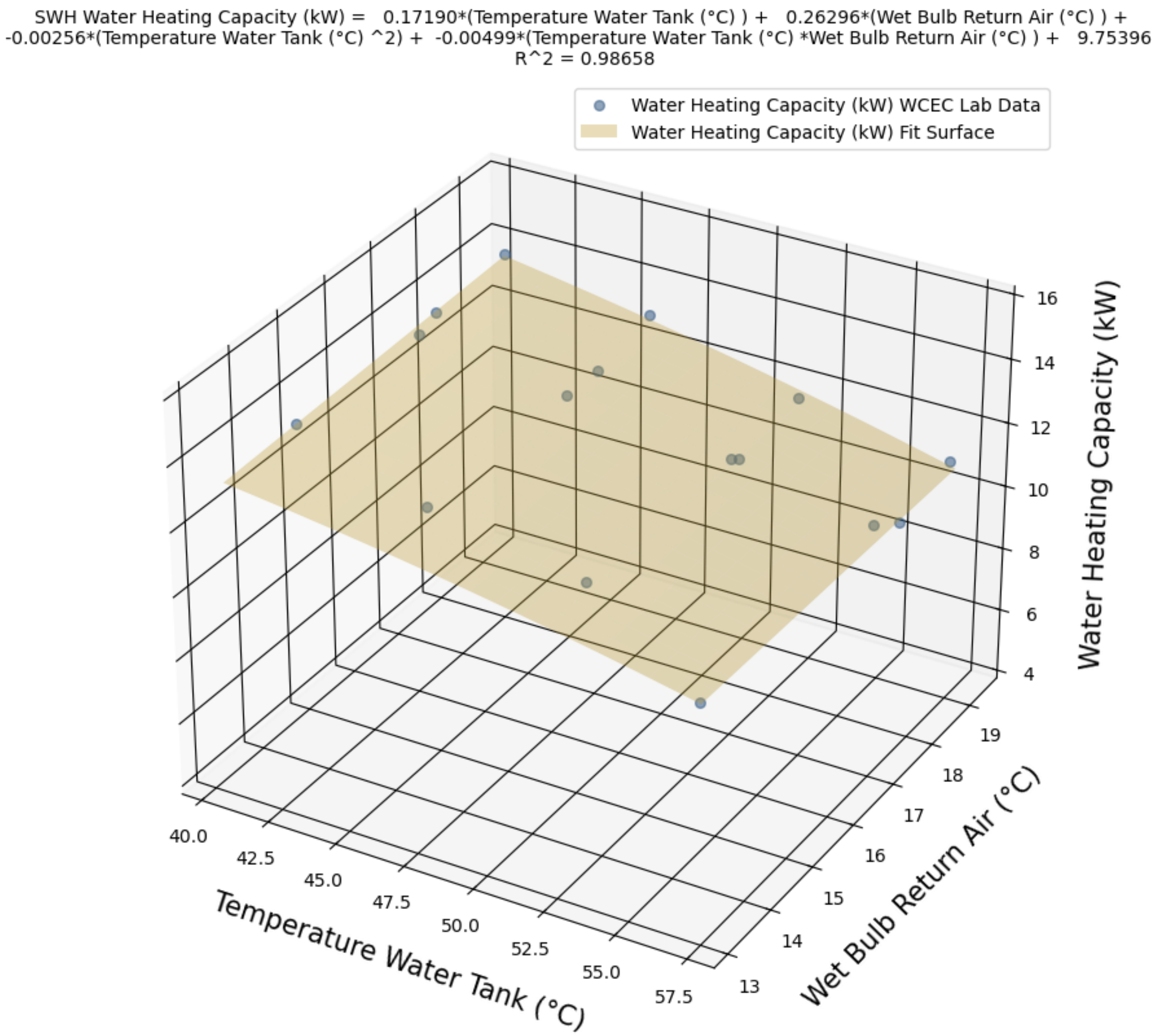
$$SHR_{SSC} = \overline{\boldsymbol{C}}_{\boldsymbol{SSC,R}} \cdot \begin{bmatrix} T_{WTH} & T_{RA,DP} & T_{WTH}^2 & T_{RA,DP}^2 & T_{WTH} \cdot T_{RA,DP} & 1 \end{bmatrix}$$

Linear Least Squares Coefficients and coefficient of determination:

$$\overline{\boldsymbol{C}}_{SSC,R} = \begin{bmatrix} \mathbf{-7.0347e-03} \\ \mathbf{-6.9341e-02} \\ \mathbf{0.0000e+00} \\ \mathbf{4.5247e-04} \\ \mathbf{1.0589e-03} \\ \mathbf{1.3496e+00} \end{bmatrix}$$

$$R^2 = 9.7231\text{e}-01 \approx 97.2\%$$

Simultaneous Water Heating Modeling Fits (operates along with SSC mode)



Capacity:

$$\dot{H}_{SWH} = \overline{\boldsymbol{C}}_{\boldsymbol{SWH,H}} \cdot \begin{bmatrix} T_{WTH} & T_{RA,WB} & T_{WTH}^2 & T_{RA,WB}^2 & T_{WTH} \cdot T_{RA,WB} & 1 \end{bmatrix}$$

Linear Least Squares Coefficients and coefficient of determination:

$$\overline{\boldsymbol{C}}_{\boldsymbol{SWH,H}} = \begin{bmatrix} 1.7190\text{e}-01 \\ 2.6296\text{e}-01 \\ -2.5573\text{e}-03 \\ 0.0000\text{e}+00 \\ -4.9938\text{e}-03 \\ 9.7540\text{e}+00 \end{bmatrix}$$

$$R^2 = 9.8658\text{e}-01 \approx 98.7\%$$

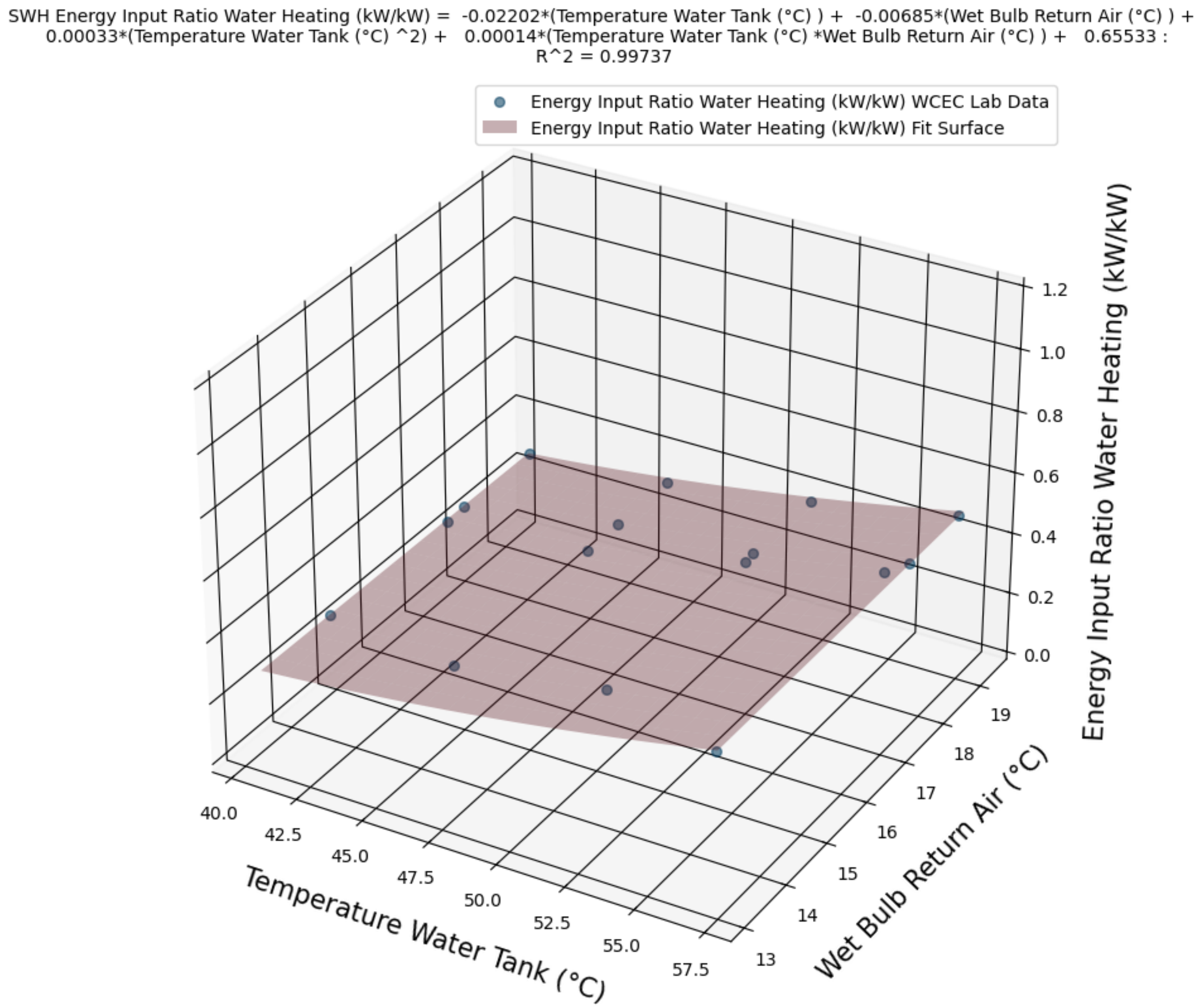
Rated and normalized Capacities:

$$\dot{H}_{SWH,rated} = \dot{H}_{SWH} \, @_{T_{RA,WB}:\,17.22°\text{C}}^{T_{WTH}:\,51.67°\text{C}} = \mathbf{1.1893e+01}\ \boldsymbol{kW}$$

$$\dot{H}_{SWH,norm} = \overline{\boldsymbol{C}} \cdot \begin{bmatrix} T_{WTH} & T_{RA,WB} & T_{WTH}^2 & T_{RA,WB}^2 & T_{WTH} \cdot T_{RA,WB} & 1 \end{bmatrix}$$

Normalized coefficients:

$$\overline{\boldsymbol{C}} = \overline{\boldsymbol{C}}_{\boldsymbol{SWH,H}} / \dot{H}_{SWH,rated} = \begin{bmatrix} \mathbf{1.4454e-02} \\ \mathbf{2.2109e-02} \\ \mathbf{-2.1502e-04} \\ \mathbf{0.0000e+00} \\ \mathbf{-4.1988e-04} \\ \mathbf{8.2011e-01} \end{bmatrix}$$



Energy Input Ratio (inverse of Coefficient of Performance):

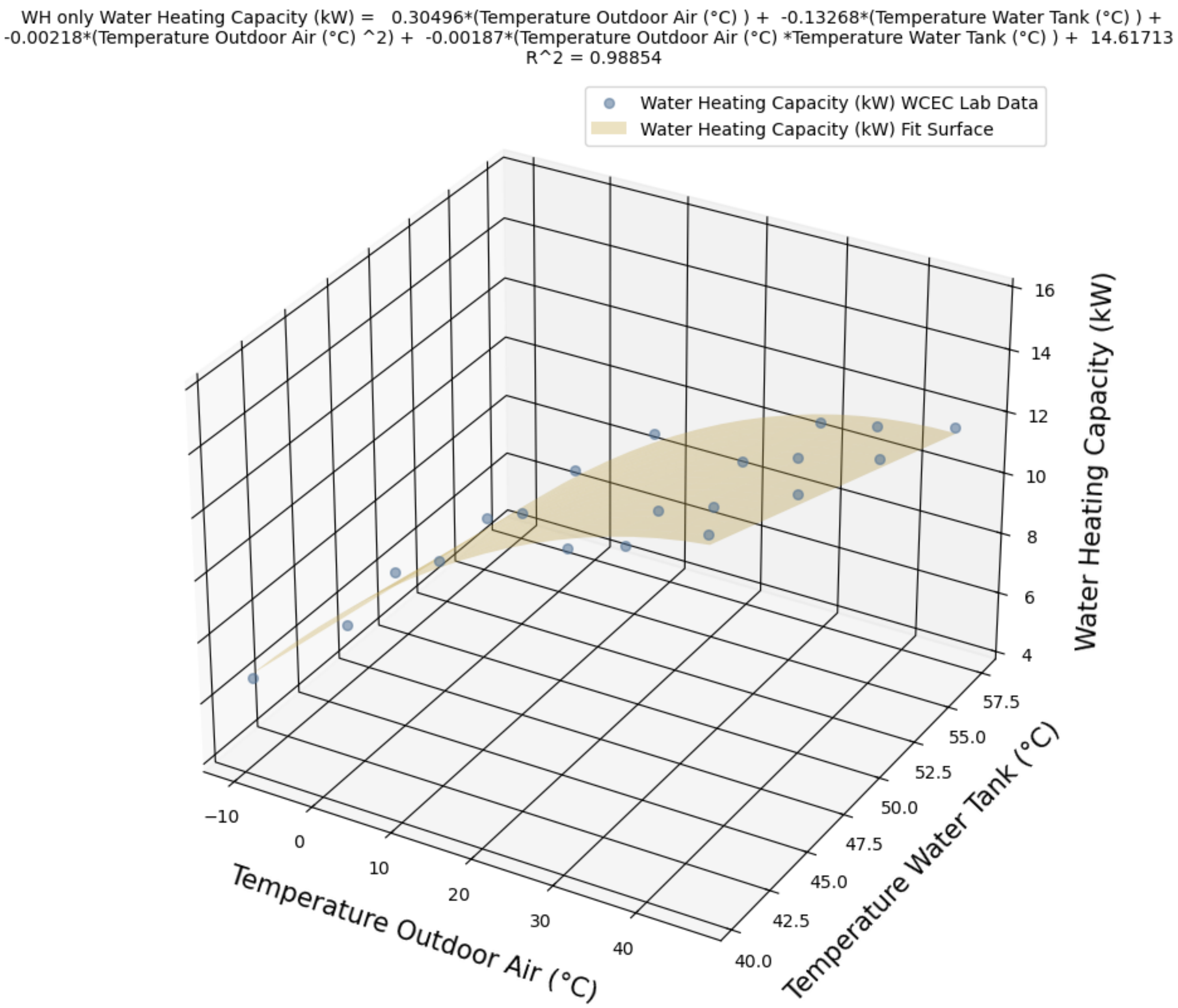
$$EIR_{SWH} = \frac{1}{COP_{SWH}} = \overline{\boldsymbol{C}}_{\boldsymbol{SWH,E}} \cdot \begin{bmatrix} T_{WTH} & T_{RA,WB} & T_{WTH}^2 & T_{RA,WB}^2 & T_{WTH} \cdot T_{RA,WB} & 1 \end{bmatrix}$$

Linear Least Squares Coefficients and coefficient of determination:

$$\overline{\boldsymbol{C}}_{\boldsymbol{SWH,E}} = \begin{bmatrix} \mathbf{-2.2019e-02} \\ \mathbf{-6.8465e-03} \\ \mathbf{3.2917e-04} \\ \mathbf{0.0000e+00} \\ \mathbf{1.3848e-04} \\ \mathbf{6.5533e-01} \end{bmatrix}$$

$$R^2 = 9.9737\mathrm{e}-01 \approx 99.7\%$$

Water Heating Modeling Fits



Capacity:

$$\dot{H}_{WH} = \overline{\boldsymbol{C}}_{\boldsymbol{WH,H}} \cdot \begin{bmatrix} T_{OA,DB} & T_{WTH} & T_{OA,DB}^2 & T_{WTH}^2 & T_{OA,DB} \cdot T_{WTH} & 1 \end{bmatrix}$$

Linear Least Squares Coefficients and coefficient of determination:

$$\overline{\boldsymbol{C}}_{\boldsymbol{WH,H}} = \begin{bmatrix} 3.0496\text{e} - 01 \\ -1.3268\text{e} - 01 \\ -2.1808\text{e} - 03 \\ 0.0000\text{e} + 00 \\ -1.8656\text{e} - 03 \\ 1.4617\text{e} + 01 \end{bmatrix}$$

$$R^2 = 9.8854\text{e} - 01 \approx 98.9\%$$

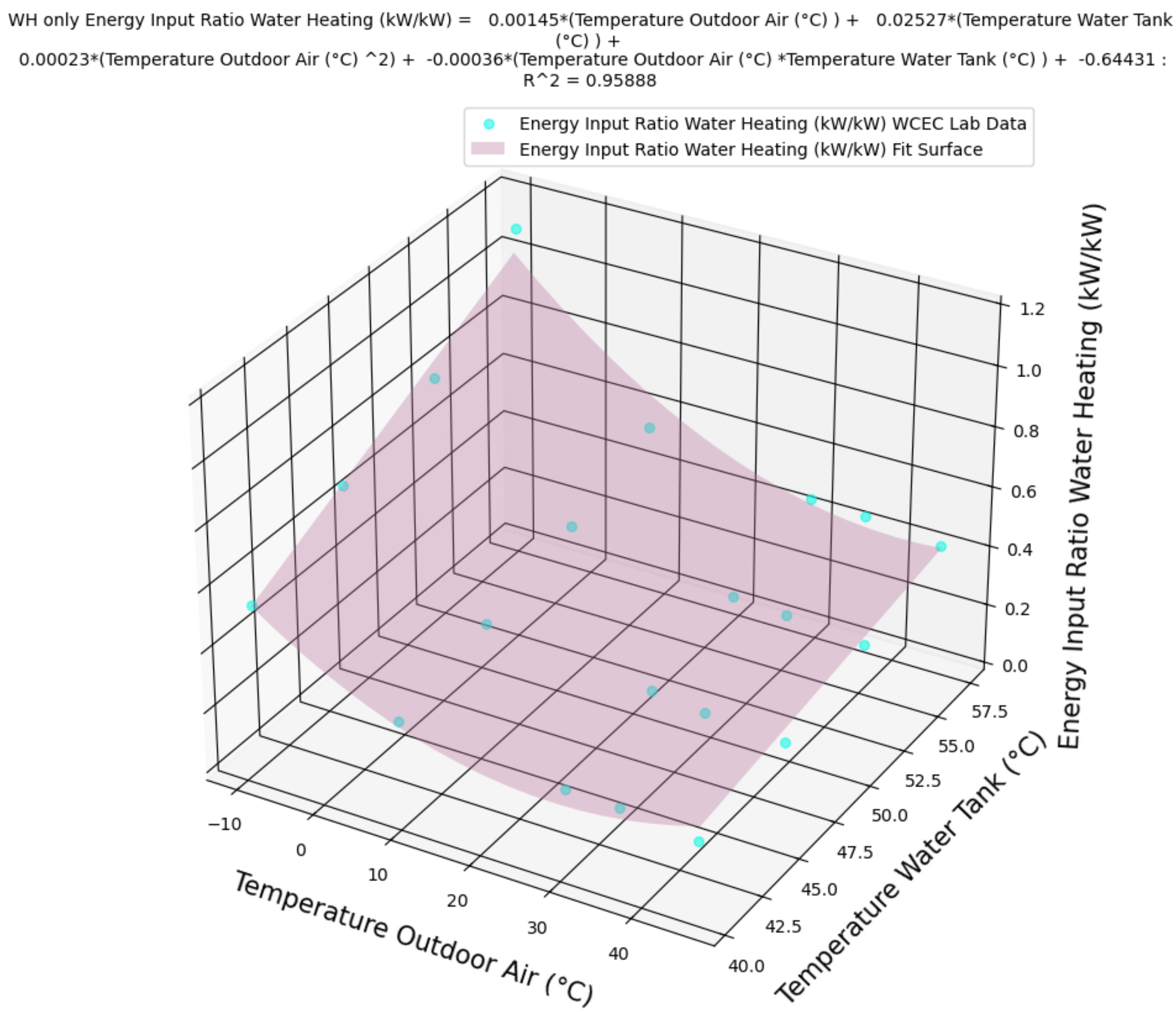
Rated and normalized Capacities:

$$\dot{H}_{WH,rated} = \dot{H}_{WH} @_{T_{OA,DB}: 19.44°C}^{T_{WTH}: 51.67°C} = \mathbf{1.0992e + 01}\ \boldsymbol{kW}$$

$$\dot{H}_{WH,norm} = \overline{\boldsymbol{C}} \cdot \begin{bmatrix} T_{OA,DB} & T_{WTH} & T_{OA,DB}^2 & T_{WTH}^2 & T_{OA,DB} \cdot T_{WTH} & 1 \end{bmatrix}$$

Normalized coefficients:

$$\overline{\boldsymbol{C}} = \overline{\boldsymbol{C}}_{\boldsymbol{WH,H}} / \dot{H}_{WH,rated} = \begin{bmatrix} \mathbf{2.7745e - 02} \\ \mathbf{-1.2071e - 02} \\ \mathbf{-1.9841e - 04} \\ \mathbf{0.0000e + 00} \\ \mathbf{-1.6973e - 04} \\ \mathbf{1.3298e + 00} \end{bmatrix}$$



Energy Input Ratio (inverse of Coefficient of Performance):

$$EIR_{WH} = \frac{1}{COP_{WH}} = \overline{\boldsymbol{C}}_{\boldsymbol{WH,E}} \cdot \begin{bmatrix} T_{OA,DB} & T_{WTH} & T_{OA,DB}^2 & T_{WTH}^2 & T_{OA,DB} \cdot T_{WTH} & 1 \end{bmatrix}$$

Linear Least Squares Coefficients and coefficient of determination:

$$\overline{\boldsymbol{C}}_{\boldsymbol{WH,E}} = \begin{bmatrix} \mathbf{1.4456e-03} \\ \mathbf{2.5267e-02} \\ \mathbf{2.3498e-04} \\ \mathbf{0.0000e+00} \\ \mathbf{-3.6140e-04} \\ \mathbf{-6.4431e-01} \end{bmatrix}$$

$$R^2 = 9.5888e-01 \approx 95.9\%$$

Defrost Operation of a Single-Speed traditional heat pump test