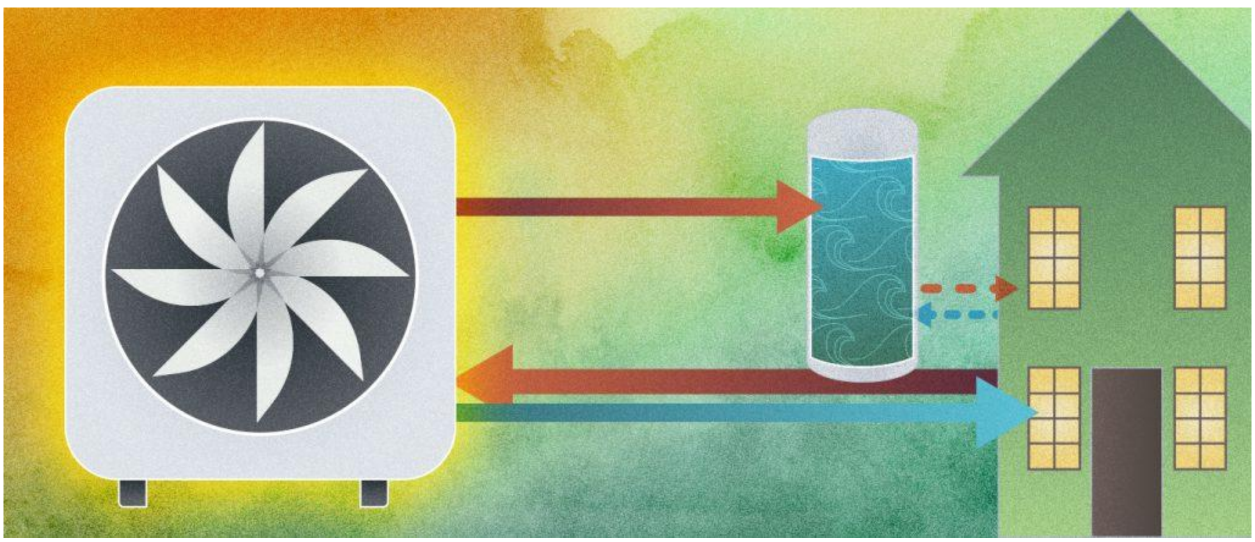
Lab Test of a Variable Speed Air-to-Air Multi-Function Heat Pump
Final Report

ET23SWE0066



Source: University of California, Davis, Western Cooling Efficiency Center

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Disclaimer

Executive Summary

Background

Residential heat pump space-conditioning and water-heating products are much 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, the potential need for electrical service upgrades can add cost and delay installation (Outcault, DePew, 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 required relative 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 more likely to avoid expensive upgrades.

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

Objectives

This project laboratory tested the residential variable-speed MFHP equipment in the University of California, Davis Western Cooling Efficiency Center environmental chambers. The tests measured its capacity and energy consumption, across a range of indoor and outdoor conditions, for: space cooling and heating; water heating; heat recovery using simultaneous space cooling and water heating; bi-heating, which simultaneously provided space heating and water heating; and defrost. The team used results to develop performance curves that future projects will use in whole-building simulations to estimate energy savings in residential buildings across climate zones and building types.

The project included tests 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 covered the California climate zones and provided 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 (US DOE 2024).

Findings

The maximum power consumption over all tests was 3.97kW.

Space Cooling: In space cooling mode, at outdoor temperature 95°F db and indoor temperature db/wb 80°F /67°F, the variable-speed MFHP space cooling mode with full compressor speed of 53Hz provided a total cooling capacity of 7.28 kW (24.84 k Btu/h, 2.07 ton) at a COP of 3.42 with sensible heat ratio of 0.699. This performance is slightly better than the European ratings test data of 7.1kW cooling capacity at a COP of 3.40. At these same outdoor and indoor conditions, minimum compressor speed of 15 Hz provided a cooling capacity of 1.91 kW (6.53 k Btu/h, 0.54 ton) at a COP of 3.61, and at 28 Hz provided a cooling capacity of 4.60 kW (15.68 k Btu/h, 1.31 ton) at a COP of 4.32.

Space Heating: In space heating mode, the variable-speed MFHP performed reliably, even at low outdoor temperatures. At outdoor temperature db/wb 47°F/43°F and indoor temperature of db 70°F, the variable-speed MFHP SH mode with full compressor speed of 49Hz provided a heating capacity of 7.28 kW (24.8 k Btu/h, 2.07 ton) at a COP of 3.96. This performance is slightly better than the EU ratings test data of 7.1kW heating capacity at a COP of 3.90. At these same outdoor and indoor conditions, the variable-speed MFHP with minimum compressor speed of 15 Hz provided a heating capacity of 2.39 kW (8.16 k Btu/h, 0.68 ton) at a COP of 4.69, and at 25 Hz provided a heating capacity of 3.83 kW (13.2 k Btu/h, 1.07 ton) at a COP of 4.90.

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. In water heating mode, at 47°F db outdoor temperature, when heating water from 105 to 115°F water tank temperature, provided a capacity of 8.3kW (28.3 k Btu/h) at a COP of 2.46. The variable-speed MFHP achieved an estimated first-hour rating of 86.4 gallons.

Heat Recovery: Heat Recovery mode with simultaneous space cooling and water heating achieved higher COP than the separate space cooling and water heating modes. For indoor temperature db/wb 80/67°F, the simultaneous mode saved an average of 33 percent of electrical energy compared to performing space cooling and water heating separately heating the water tank from 110 to 120°F at outdoor temperature db/wb 95/75°F. Heat recovery mode provided space cooling capacity similar to dedicated space cooling mode and water heating capacity lower than dedicated water heating mode for dedicated modes at outdoor temperature db/wb 95/75°F.

Bi-Heating: In Bi-Heating mode, with simultaneous space heating and water heating, the variable-speed MFHP manufacturer controls select compressor speeds higher than for either the separate SH or WH modes at the same conditions. At OA db/wb 47°F /43°F and IA db 70°F , heating the water tank from an average temperature of 80°F up to 120°F , the variable-speed MFHP Bi-Heating mode with compressor speed of 78Hz provided an SH capacity of 7.0 kW (23.9 k Btu/h, 2.0 ton) and average WH capacity of 4.5 kW for a combined COP of 3.36. There was not a significant difference between power consumption or COP for Bi-Heating compared to the estimates for separate modes delivering the same SH and WH capacity

Defrost: Defrost operation for the variable-speed 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 just under ten minutes with an average power of 1.8 kW and peak power of 3.18 kilowatts. Compared to typical single-speed split-system heat pumps, the variable-speed MFHP completed defrost at much lower system power since it does not use resistance heaters. This makes the variable-speed MFHP more likely to fit on existing electrical panels without needing an upgrade.

Table 1 Summary of key results

Mode and Conditions (Temperatures °F db/wb)	Key Results	
Space Cooling OA 95, IA 80/67	Compressor 53Hz Capacity 7.28 kW COP 3.42	Compressor 15Hz Capacity 1.91 kW COP 3.61
Space Heating OA 47/43, IA 70	Compressor 49Hz Capacity 7.28 kW COP 3.96	Compressor 15Hz Capacity 2.39 kW COP 4.69
Water Heating OA 47/43, WT from 65 to 127	Compressor 65Hz Capacity 7.25 kW COP 3.24	WT heating time 48min (WT from 65 to 120°F) FHR 86.4 gal
Heat Recovery OA 95, IA 80/67, WT from 65 to 125	Compressor 35Hz Capacity SC 5.67 kW Capacity WH 6.16 kW Combined COP 7.92	33% Energy savings compared to separate SC and WH cycles (WT from 110 to 120°F)
Bi-Heating OA 47/43, IA 70 WT from 80 to 120	Compressor 78Hz Capacity SH 7.0 kW Capacity WH 4.5 kW Combined COP 3.36	
Defrost OA 37/35	590 seconds Max electric power 3.18 kW	WT 128 drop to 119

Discussion and Recommendations

Compared to the typical single speed split system heat pumps and standalone heat pump water heaters, this variable-speed air-to-air MFHP has the potential to increase energy savings and help avoid panel upgrades to reduce retrofit electrification costs. This equipment has the potential for significant energy savings compared to the single speed air-to-air MFHP previously tested ET23SWE0047. We recommend that MFHP technologies move forward to lab and field test additional commercially available products and to develop new efficiency measures for utility energy efficiency and electrification programs.

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. We also reached out to the California Technical Forum and to San Diego Gas and Electric, the California statewide lead for heating, ventilation, and air conditioning efficiency programs, to start the measure development process.

Abbreviations and Acronyms

Acronym	Meaning
A	Amps
AC	Air conditioner
ACEEE	American Council for an Energy-Efficient Economy
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
BH	Bi-heating – simultaneous space heating and water heating
BPHX	Brazed plate heat exchangers
CalFlexHub	California Load Flexibility Hub
CalTF	California Technical Forum
CBECC-Res	California Building Energy Code Compliance Software – Residential
CEC	California Energy Commission

Acronym	Meaning
COP	Coefficient of performance
DAC	Disadvantaged communities
db	Dry bulb temperature
DEER	Database of Energy Efficiency Resources
DHW	Domestic hot water
DX	Direct expansion
EE	Energy efficiency
EIR	Energy input ratio
EU	European Union
FHR	First-hour rating
GHG	Greenhouse gas
GPM	Gallons per minute
GWP	Global warming potential
HP	Heat pump
HPWH	Heat pump water heater

Acronym	Meaning
HR	Heat recovery: Simultaneous space cooling and water heating mode
HTR	Hard-to-reach
HVAC	Heating, ventilation, and air conditioning
IA	Indoor air
IOU	Investor-owned utility
kWh	Kilowatt-hour
MFHP	Residential multi-function heat pump
NEEA	Northwest Energy Efficiency Alliance
OA	Outdoor air
ODU	Outdoor unit
PA	Program administrator
PG&E	Pacific Gas & Electric
R&D	Research and development
RT	Refrigeration ton

Acronym	Meaning
SC	Space cooling
SCE	Southern California Edison
SDG&E	San Diego Gas & Electric
SEER	Seasonal Energy Efficiency Ratio
SHR	Sensible heat ratio
TPM	Technology Priority Map
UC	University of California
V	Volts
wb	Wet bulb temperature
WCEC	Western Cooling Efficiency Center
WH	Water heating

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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, DePew, et al., Residential Electrification in Sacramento and Its Impact on Residential Appliance Sales 2021). Around 30 to 65 percent of all homes in California are expected to need electrical-service-panel upgrades to fully electrify (Efficiency First California 2020, Merski 2021, Murphy 2022, Zhao 2021, Lindsey 2023, Fournier, et al. 2024). The most recent estimate reviewed California residential building stock as-built electric service capacity by building square footage and construction date and then modified it based on the probability of previous panel upgrade based on building permits (Fournier, et al. 2024). This study found that to fully electrify, 32 percent of single-family and 59 percent of multifamily homes need panel upgrades (Fournier, et al. 2024). This paper suggests that by planning ahead to select lower peak power appliances and using smart load management equipment, the fraction of homes needing panel upgrades to fully electrify could be reduced to 3% of single-family and 10% of multi-family homes needing upgrades (Fournier, et al. 2024). This study also shows that residential buildings in DAC are two to four times more likely to need electric panel upgrades or load management strategies than buildings in non-DAC areas. Load management strategies can include choosing low peak power appliances like 120V heat pump water heaters (HPWHs) or 120V HP clothes dryers if they provide sufficient energy services; using smart panels, smart circuit breakers or circuit sharing devices; or choosing multi-purpose appliances like a multi-function heat pump (MFHP). The cost of residential electric panel upgrades in California in single-family homes is typically around $5,000 but can range from $2,000 to $30,000 (Pena, et al. 2022), potentially a prohibitive additional cost.

Residential MFHPs use one efficient compressor and outdoor heat exchanger coil to provide space cooling, space heating, and domestic hot water heating, Figure 1. 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 equipment, so they are less likely to trigger the need for a service-breaker-panel or service-wire upgrade (Outcault, DePew, 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. Under sizing 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. For variable-speed HPs, under sizing can lead to lower efficiency operation at the highest compressor speeds during the hottest weather (Wilcox, Gartland and Conant

2019). For variable-speed HPs, right sizing and slight oversizing can lead to higher efficiency operation at low to moderate compressor speeds and significant oversizing can lead to lower efficiency cycling on and off at low loads (Wilcox, Gartland and Conant 2019).

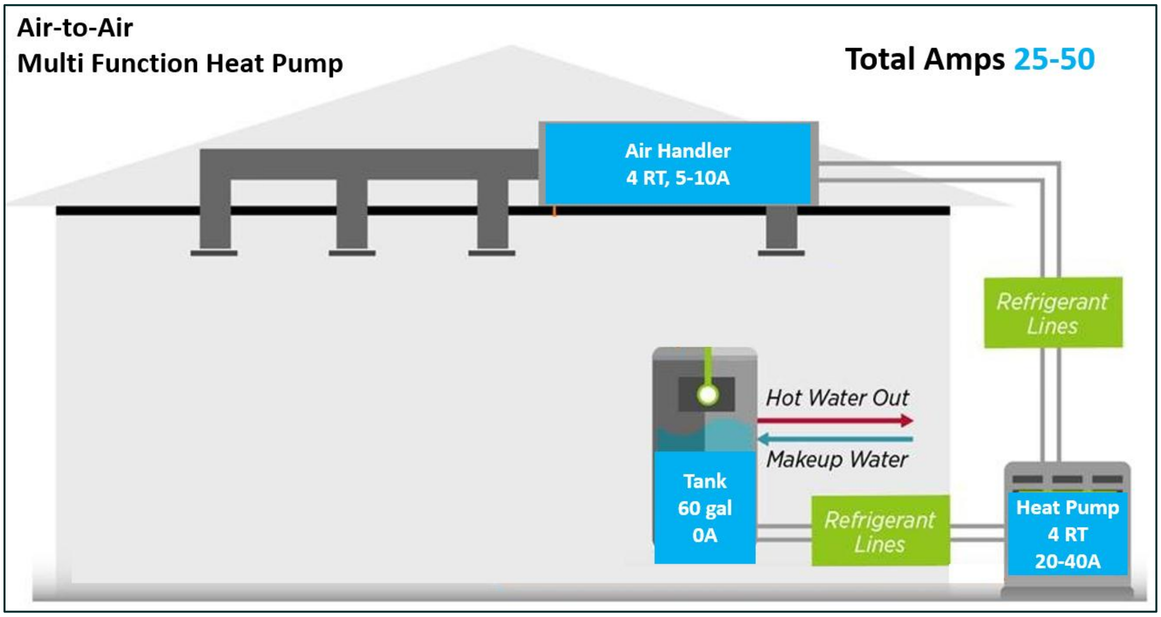

Figure 1: Air-to-air multi-function heat pump system diagram showing the outdoor unit and refrigerant lines serving both the air handler and the indoor hot water tank.

Source: Adapted from original image provided by a manufacturer.

Multi-Function Heat Pump Products

The University of California, Davis Western Cooling Efficiency Center (UC Davis 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 and globally (Vernon, Residential Multi-Function Heat Pumps: Product Search 2022). There was only one air-to-air MFHP product commercially available in California that the project team was able to find as of November 2023, but the manufacturer has stopped sales of this single-speed product as of March 2025 to focus on developing variable-speed HP products. The variable-speed air-to-air MFHP product tested in this project is produced by a large international manufacturer that sells it in Europe, but there is no announced date for offering the product in the US 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, two other large international manufacturers announced plans to offer a residential air-to-air MFHP product in the United States in 2023. A year later, at the 2024 AHR expo, the companies showcased their air-to-air MFHP products at their booths. Although both air-to-air MFHP products are now commercially available in the United States as of early 2025, neither of the companies is building and selling the water heating accessory and storage tank necessary for domestic hot water (DHW) production in the United States. These two manufacturers still planto offer these products in the United States with some delays due to the transition to lower global warming potential (GWP) refrigerants. At the 2025 ASHRAE AHR Expo, a fourth international manufacturer announced intentions to sell a variable-speed air-to-air MFHP in the United States in 2025.

Background

Residential MFHPs use one efficient compressor and outdoor heat exchanger coil to provide space conditioning and DHW. The variable-speed air-to-air MFHP tested uses one outdoor unit with outdoor coil, variable-speed compressor, and variable-speed outdoor fan to supply refrigerant to both an indoor air handler unit and an indoor hot water tank unit, as shown in Figure 2 and Figure 3. The indoor air handler unit has a direct expansion refrigerant to air heat exchanger coil and variable-speed fan. The hot water tank unit has a brazed plate refrigerant-to-water heat exchanger with an internal closed loop (primary loop) of water that circulates through the heat exchanger and then through a closed coil pipe water-to-water heat exchanger inside the hot water tank to heat the potable hot water that is delivered to the home. The MFHP tested has a two-ton cooling capacity rating and uses R32 refrigerant.

Figure 2: Outdoor unit (top) and indoor air handler unit (bottom).

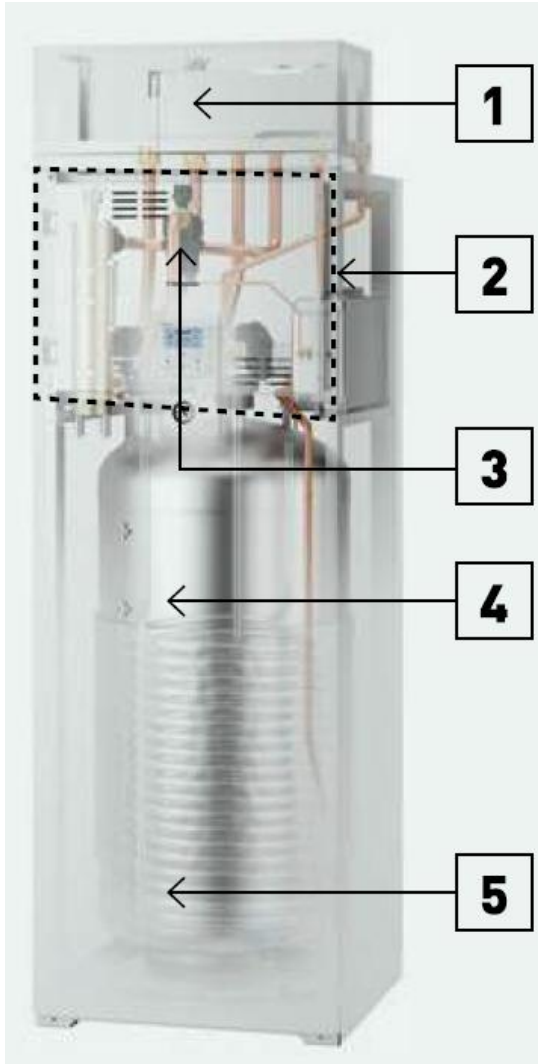

Figure 3: Indoor hot water tank unit with parts labeled.

1. Refrigerant-to-water brazed plate heat exchanger
2. All hydronic components accessible for maintenance
3. Accessible water filter for maintenance
4. closed coil pipe water-to-water heat exchanger inside the hot water tank
5. Vacuum insulation to reduce heat loss.

(Source: publicly available manufacturer brochure)

The variable-speed air-to-air MFHP tested has several operating modes:

1. Space Cooling
2. Space Heating

3. Water Heating
4. Simultaneous Space Cooling and Water Heating (Heat Recovery)
5. Simultaneous Space Heating and Water Heating (Bi-Heating)
6. Defrost

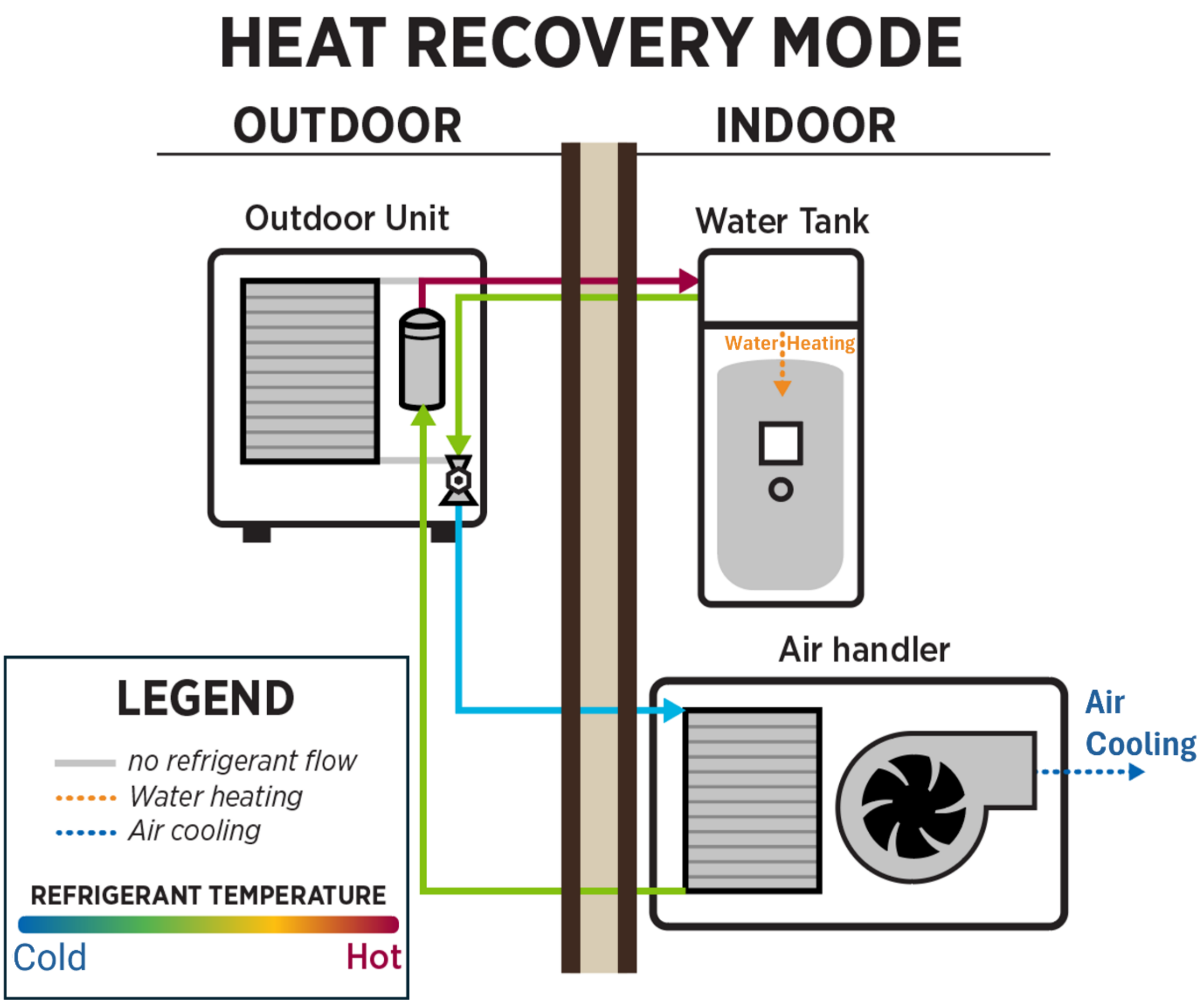

Figure 4: Heat Recovery mode with simultaneous space cooling and domestic hot water heating for an air-to-air multi-function heat pump system, showing the outdoor unit and refrigerant flowing through both the air handler and the indoor hot water tank and no refrigerant flowing through the outdoor coil.

Source: University of California, Davis, Western Cooling Efficiency Center.

Refrigerant valves and expansion valves control which components act as the refrigerant evaporator and condenser for each of the operating modes, as detailed in Table 2.

Table 2: MFHP operational modes and respective evaporator and condenser combinations.

Mode	Evaporator	Condenser
Space Cooling	Indoor coil	Outdoor coil
Space Heating	Outdoor coil	Indoor air handler coil
Water Heating	Outdoor coil	DHW tank refrigerant-to-water brazed plate heat exchanger
Heat Recovery (simultaneous space cooling and water heating)	Indoor coil	DHW tank refrigerant-to-water brazed plate heat exchanger
Bi-Heating (simultaneous space heating and water heating)	Outdoor coil	Indoor air handler coil + DHW tank refrigerant-to-water brazed plate heat exchanger
Defrost	DHW tank refrigerant-to-water brazed plate heat exchanger	Outdoor coil
Nonstop Defrost	DHW tank refrigerant-to-water brazed plate heat exchanger	Outdoor coil + indoor air handler coil

WCEC's study of HP market adoption 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 can 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; most choose lower-cost and faster-turnaround AC replacement. DAC and HTR customers are more likely to need electrical service upgrades and, therefore, have a larger barrier to electrification than non-DACs. MFHPs have the potential to significantly reduce electrification costs and installation times, particularly for DAC and HTR customers.

A review of publicly available equipment specifications shows that a typical three-ton capacity, single-speed 15.2 SEER2 outdoor central AC or heat pump requires a 20 to 35-amp breaker for the outdoor unit and a 10- to 15-amp breaker for the air handler for a total of 30 to 50 amps of breakers. A typical three-ton capacity, single speed 15.2 SEER2 central heat pump has similar requirements to the AC and adds a requirement for the backup electric resistance heaters (strip heat) of 25 to 60 amps, for a total of 55 to 110 amps. A typical standalone hybrid heat pump water heater with a 50-gallon tank has a 4.5kW resistance heater and requires a 30-amp breaker. The typical combination of separate heat pumps for space conditioning and for hot water heating would require 85 to 150 amps of breakers. MFHPs do not require backup electric resistance heaters so they would require the same electrical service capacity as a comparable AC and do not need a separate electrical circuit for hot water heating, so that both space conditioning and water heating would require a total of 25-35 amp breaker. This is a reduction of 60 to 115 amps of breakers compared to the typical equipment.

Some past work in combination HP heating and hot water heating used equipment that was not capable of providing space cooling, such as CO_2-refrigerant HPs (Eklund 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 large barriers to adoption from installers and customers.

There is one provider of custom-engineered residential air-to-water MFHPs based on hydronic thermal energy distribution in Northern California: Harvest Thermal. Harvest Thermal uses a separate AC for cooling, so it is expected to be significantly more expensive. These systems use air-to-water HPs that send heated or chilled water to air handlers with hydronic coils and one or more hot water tanks. 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. These systems are expected to have premium product pricing and to have higher installation costs, making them a less-than-ideal fit for DACs.

Other previous work in air-to-air MFHPs that provide space heating, space cooling, and hot water heating used a prototype that connected a Mitsubishi ductless HP outdoor unit to an indoor air handler unit and an adapted standalone HPWH pressurized tank and integrated refrigerant-to-water heat exchanger (Energy 350 2015). This prototype used backup electric resistance heaters in the hot water tank so that it would still have moderately high peak power consumption and may require similar electrical service upgrades as typical separate HP systems.

WCEC MFHP Studies

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 has developed control strategies to enable MFHPs to deploy simultaneous space cooling and hot water heating more often. As part of the CEC-funded CalFlexHub project focused on the single speed air-to-air MFHP, WCEC completed monitoring in March 2025 for the original field site, as well as two additional field test sites in two tenant units in a Central Valley multifamily building, all of which have shown good performance.

MFHP Field Test

WCEC and Frontier Energy completed a field test of a prototype, single-speed air-to-air MFHP in 2022 (Chally and Haile 2024). The completed field test of the prototype showed slightly lower coefficients of performance (COPs) than the Air Conditioning, Heating, and Refrigeration Institute (AHRI) ratings for the original split system HP before it was modified with additional refrigerant valves to become the MFHP. The MFHP simultaneous space cooling and hot water heating moved the thermal energy from the space into the hot water tank to heat hot water and provided both services 36 percent more efficiently than using a separate space cooling and hot water heating cycle, each exchanging heat with the outdoors. The results of the field test were presented at the IEA Heat Pump Conference in May 2023 (Chakraborty, Chally and Levering 2023).

Single-Speed MFHP Lab Test

WCEC completed laboratory testing of the then commercially available production model four-ton-capacity single-speed air-to air MFHP in late 2024 (Vernon and Chakraborty, Residential Multi-Function Heat Pump Laboratory Testing 2025). The laboratory test results show that the single-speed MFHP unit capacity and COP for both space heating and space cooling modes match the third-party lab test results for AHRI-rating manufacturer-published data for the mass-produced split system HP before it was modified with additional refrigerant valves to become the MFHP. This alignment suggests that discrepancies observed in previous field tests 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 in the field test prototype.

The four-ton single-speed MFHP in laboratory tests at 95°F outdoor air temperature and 80°F indoor air temperature had a measured space cooling capacity of 45.3 k Btu/h at a COP of 3.57. The MFHP space heating mode performed reliably, even at low outdoor temperatures, and at 47°F outdoor air temperature and 70°F indoor air temperature, showed a capacity of 49.9 k Btu/h at a COP of 3.48.

As expected for all HPWH equipment, the laboratory test-measured water heating capacity and COP at a given outdoor air temperature decreased as water tank temperatures increased. At 47°F outdoor air temperature and 115°F water tank temperature, the dedicated water heating mode demonstrated a capacity of 36.1 k Btu/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 air temperatures below 17°F. The MFHP achieved an estimated first-hour rating (FHR) 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 FHR.

In simultaneous space cooling and water heating mode, the COP generally outperformed the separate space cooling and water heating modes. The Heat Recovery mode saved an average of 36 percent of electrical energy compared to performing space cooling and water heating separately, very similar to the field test results. Even under extreme conditions with water tank setpoint temperatures above 135°F and outdoor air temperatures below 75°F, the COP was higher for simultaneous mode than two separate modes.

This project is the next step to understanding the potential for MFHP products by investigating variable-speed air-to-air MFHP equipment to evaluate the potential for even larger energy savings than single speed MFHP equipment while retaining the potential to reduce barriers to electrification.

Objectives

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

- Develop a test plan to measure equipment performance of the variable-speed air-to-air MFHP across a range of outdoor air conditions to match California climate zones for: space heating, space cooling, water heating, simultaneous space heating and water heating, and simultaneous space cooling with heat recovery water heating informed by ANSI/ASHRAE Standard 206-2024 with operation across a range of equipment speeds.
- Test the efficiency and capacity performance of the variable-speed air-to-air MFHP for: space heating, space cooling, water heating, simultaneous space cooling with heat recovery water heating, simultaneous space heating and water heating, and defrost.
- Use lab test results to develop equipment-performance curves for use by future projects in EnergyPlus and CBECC-Res to estimate energy savings in residential buildings.
- Compare the single-speed MFHP lab test results and those from this variable-speed MFHP lab test project.
- 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; 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, investor-owned utilities (IOU) staff, and relevant engineering consulting firms; and other stakeholders including heating, ventilation, and air conditioning (HVAC) manufacturers (Villara,

Panasonic, Samsung, LG, Mitsubishi), and emerging technology groups and researchers from other states.

This project will not make energy savings baseline comparisons. A future measure package development project will use the EnergyPlus performance curves created in this project with the California Technical Forum (CalTF) Database of Energy Efficiency Resources (DEER) Building Prototypes to simulate energy savings for new construction and retrofit scenarios.

Methodology and Approach

The core expertise of the UC Davis 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 water heating equipment.

The ASHRAE Standard 206 test standard defines test requirements for MFHP equipment (ASHRAE 2024). The US DOE may adopt this test standard and define ratings minimum criteria. The AHRI 210/240 standard for heat pump space conditioning has been adopted by the US DOE and minimum performance ratings have been established (AHRI 2023). The EU has a separate but similar test standard EN 14511. These test standards are designed to calculate high level system performance metrics based on minimal lab testing. The tests defined in these standards are not sufficient to create performance curves for building energy simulations and accurate energy savings estimations.

Test Plan

WCEC engineers developed the space cooling and space heating test plan (Tables 3–9) selecting outdoor air (OA) conditions and indoor air (IA) conditions from the ASHRAE Standard 206 for MFHPs shown in Table 4, the AHRI 210/240 standard. Additional test points were added to fill in gaps in the test standards so that accurate performance curves could be created. The air conditions below are listed with the dry-bulb (db) temperature and wet-bulb (wb) temperature in degrees Fahrenheit. The ASHRAE MFHP standard calls for cooling test points with OA temperatures of 95 db, 82 db, 67 db, and extreme weather test at 115 db. To get performance curves accurate in California's hot summer climates, a fourth OA condition of 105 db was added for the lab tests. Testing used the ASHRAE MFHP-stated cooling IA condition of 80 db/67 wb and additional IA conditions of 78 db/64 wb and 75 db/60 wb that more closely match actual residential return air conditions for typical California thermostat setpoints. At each set of OA and IA conditions, tests were performed at two or more compressor speeds. At the 95 db OA condition, tests were performed across four compressor speeds and separately across three fan speeds to map their influence on capacity and COP.

Compressor speeds were selected at the minimum operating point (min), at the speed that achieved the EU rated cooling or heating capacity at the EU rating conditions (full), and at intermediate speeds (int1, int2). The European rating standard EN 14511 defines the test conditions and test methods for that market, with conditions very close to the US ASHRAE and AHRI test standards Table 3. The variable speed MFHP compressor is capable of speeds greater than the full speed for rated capacity. For selected conditions the equipment was run at maximum compressor speed (max). Under manufacturer controls in Bi-Heating mode with low outdoor air temperatures the unit can run at maximum compressor speed.

Table 3: US and EU standard test conditions for heat pumps.

Test Standard	Outdoor Dry Bulb °F (°C)	Outdoor Wet Bulb °F (°C)	Indoor Dry Bulb °F (°C)	Indoor Wet Bulb °F (°C)
Space Cooling US ASHRAE 206	95 (35)		80 (26.7)	67 (19.5)
Space Cooling EU EN14511	95 (35)		80.6 (27)	68 (20)
Space Heating US ASHRAE 206	47 (8.3)	43 (6.1)	70 (21.1)	
Space Heating EU EN14511	44.7 (7)	42.9 (6)	68 (20)	

Space heating (SH) conditions were selected similarly. The OA conditions selected are 62 db/56.5 wb, 47 db/43 wb, and 17 db/15 wb. These OA conditions match the ASHRAE and AHRI standard. For the IA conditions, the ASHRAE heating condition of 70 db is used. Two additional IA conditions of 75 db and 65 db are also selected to provide more data for various indoor home conditions performance mapping. The IA condition 75 db was chosen as it was the only common point between heating and cooling commonly referenced in datasheets from other original equipment manufacturers. This should enable clearer visibility into performance differences between operating conditions when using the different MFHP modes. The OA conditions in heating and cooling tests cover a broad range and match the boundaries of the California climate zone conditions. These OA and IA conditions were delivered by the environmental chambers to measure equipment performance. At the 47 db/43 wb OA condition, tests were performed across four compressor speeds and separately across three fan speeds to map their influence on capacity and COP.

Water heating (WH) test conditions were based on the ASHRAE Standard 206 providing OA points 95 db/75 wb, 47 db/43 wb, and 17 db/15 wb. The WCEC test plan is expanded to cover up to the hottest OA condition 105 db/77 wb. The WH tests ran a water heating cycle starting from 65°F and heating to the heat pump cutoff temperature. Separate from the performance map, an estimated FHR test was performed with three-gallons-per-minute (GPM) water draws for the dedicated WH mode. The WH tests were performed using the manufacturer controls of compressor speed, ODU fan speed, and refrigerant-to-water heat exchanger water pump speed.

The simultaneous space cooling and water heating mode (Heat Recovery), where the MFHP recovers heat from the indoor space and puts it into the hot water tank, and the simultaneous space heating and water heating mode (Bi-Heating) were also tested. The Heat Recovery mode performance was characterized across the three different IA conditions for space cooling variation, 80 db/67 wb, 78 db/64 wb and 75 db/60 wb. There is no heat transfer with the outdoors in Heat Recovery mode, so the OA condition has little effect and was held at 95°F db. Like WH tests, the Heat Recovery mode water heating tests start the tank at 65°F average water temperature and heat until the heat pump cutoff temperature. The Heat Recovery mode tests were performed using the manufacturer controls of compressor speed and refrigerant-to-water heat exchanger water pump speed.

Table 4: Test points from MFHP ASHRAE Test Standard 206.

MFHP ASHRAE Standard 206 Tests	OA Conditions		IA Conditions		Compressor Speed
Temperature (°F)	db	wb	db	wb	—
Space Cooling	95	75	80	67	full
	82	65	80	67	int
	82	65	80	67	full
	82	65	80	67	min
	67	53.5	80	57	min
Space Heating	62	56.5	70	60	min
	47	43	70	60	full

MFHP ASHRAE Standard 206 Tests	OA Conditions		IA Conditions		Compressor Speed
	47	43	70	60	min
	17	15	70	60	full
Defrost	35	33	70	60	int
Water Heating	67	53.5	80	67	default
	47	43	70	60	default
Heat Recovery (space cooling and water heating)	95	75	80	67	full
	82	65	80	67	full
	82	65	80	67	min
	67	53.5	80	67	min
Bi-Heating (space heating and water heating)	62	56.5	70	60	min
	47	43	70	60	full
	47	43	70	60	min
	17	15	70	60	full

Source: ANSI/ASHRAE Standard 206-2024

Table 5: Test plan for variable-speed MFHP testing in the WCEC environmental chambers, space cooling.

Test #	OA Conditions		IA Conditions		Compressor Speed
Temperature (°F)	db	wb	db	wb	---
C.1.1.S1	105	--	80	67	min
C.2.1.S1	95	--	80	67	min
C.3.1.S1	82	--	80	67	min
C.4.1.S1	67	--	80	67	min
C.().1.S2	115	--	80	67	full
C.1.1.S2	105	--	80	67	full
C.2.1.S2	95	--	80	67	full
C.3.1.S2	82	--	80	67	full
C.4.1.S2	67	--	80	67	full
C.2.1.INT1	95	--	80	67	Int1
C.2.1.INT2	95	--	80	67	Int2
C.2.1.Max	95	--	80	67	max
C.1.2.S1	105	--	78	64	min
C.2.2.S1	95	--	78	64	min
C.3.2.S1	82	--	78	64	min
C.1.2.S2	105	--	78	64	full
C.2.2.S2	95	--	78	64	full
C.3.2.S2	82	--	78	64	full

Test #	OA Conditions		IA Conditions		Compressor Speed
C.2.3.INT1 fan high	95	--	80	53	Int1
C.2.3.INT1 fan med	95	--	80	53	Int1
C.2.3.INT1 fan low	95	--	80	53	Int1
C.1.4.S1	105	--	75	60	min
C.2.4.S1	95	--	75	60	min
C.3.4.S1	82	--	75	60	min
C.1.4.S2	105	--	75	60	full
C.2.4.S2	95	--	75	60	full
C.3.4.S2	82	--	75	60	full

Table 6: Test plan for variable-speed MFHP testing in the WCEC environmental chambers, space heating.

Test #	OA Conditions		IA Conditions		Compressor Speed
Temperature (°F)	db	wb	db	wb	---
H.1.1.S1	62	56.5	70	≤60	min
H.2.1.S1	47	43	70	≤60	min
H.3.1.S1	17	15	70	≤60	min
H.1.1.INT1	62	56.5	70	≤60	Int1
H.2.1.INT1	47	43	70	≤60	Int1
H.3.1.INT1	17	15	70	≤60	Int1
H.2.1.INT2	47	43	70	≤60	Int2

Test #	OA Conditions		IA Conditions		Compressor Speed
H.2.1.max	47	43	70	≤60	max
H.2.1.S2 fan high	47	43	70	≤60	full
H.2.1.S2 fan med	47	43	70	≤60	full
H.2.1.S2 fan low	47	43	70	≤60	full
H.1.2.S1	62	56.5	65	≤55	min
H.2.2.S1	47	43	65	≤55	min
H.4.2.S1	17	15	65	≤55	min
H.1.2.S2	62	56.5	65	≤55	full
H.2.2.S2	47	43	65	≤55	full
H.4.2.S2	17	15	65	≤55	full
H.1.3.S1	62	56.5	75	≤65	min
H.2.3.S1	47	43	75	≤65	min
H.4.3.S1	17	15	75	≤65	min
H.1.3.S2	62	56.5	75	≤65	full
H.2.3.S2	47	43	75	≤65	full
H.4.3.S2	17	15	75	≤65	full

Table 7: Test plan for variable-speed MFHP testing in the WCEC environmental chambers, water heating.

Test #	OA Conditions		IA Conditions		Compressor Speed
Temperature (°F)	db	wb	db	wb	—
WH.1.mnf control	105	77	NA	NA	default
WH.2.mnf control	95	75	NA	NA	default
WH.3.mnf control	47	43	NA	NA	default
WH.4.mnf control	17	15	NA	NA	default

Table 8: Test plan for variable-speed MFHP testing in the WCEC environmental chambers, heat recovery simultaneous space cooling and water heating.

Test #	OA Conditions		IA Conditions		Compressor Speed
Temperature (°F)	db	wb	db	wb	—
HR.C.2.mnf control	95		80	67	default
HR.C.2.mnf control	95		78	64	default
HR.C.2.mnf control	95		75	60	default

Table 9: Test plan for variable-speed MFHP testing in the WCEC environmental chambers, bi-heating simultaneous space heating and water heating.

Test #	OA Conditions		IA Conditions		Compressor Speed
Temperature (°F)	db	wb	db	wb	—
BH.H.1.mnf control	62	56.5	70	≤60	default
BH.H.2.mnf control	47	43	70	≤60	default

Test #	OA Conditions		IA Conditions		Compressor Speed
BH.H.4.mnf control	17	15	70	≤60	default

Table 10: Test plan for variable-speed MFHP testing in the WCEC environmental chambers, defrost.

Test #	OA Conditions		Compressor Speed
Temperature (°F)	db	wb	–
WH.3.mnf control	37	34	default

Instrumentation

Data Collection

Each test condition was run for multiple 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 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 arrays of RTD probes, Vaisala temperature and humidity probes, and OptiSonde chilled mirror hygrometers. For the AHU these sensor arrays were positioned at the SA and RA ducts close to the connection with the air handler unit. The chilled mirrors used four-point humidity sampling arrays, and four-sensor RTD probe arrays measured db temperature to accurately measure conditions even if the air stream happened to be stratified. The outdoor air temperature was monitored using 12 RTDs and a Vaisala temperature and humidity sensors positioned around the ODU condenser coil inlet. The averaged value of these outdoor air sensor arrays was taken as the effective outdoor db used to control the chamber temperature.

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

The AHU return air volumetric flow rate was measured using a calibrated nozzle box. Mass flow rates were calculated from the volumetric flow rates with densities calculated from the psychrometric properties of the given air flow.

Power

Power was measured at the power connection to the MFHP ODU, AHU, and water tank unit separately, using three 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 water tank (WT) using an Omega pulse flow meter. The inlet and outlet water temperatures were measured using well insulated Omega surface-mount RTDs on the inlet and outlet pipes to the water tank. WT temperatures were measured by an array of seven hermetically sealed Omega RTDs positioned near the center of the tank at evenly spaced vertical interval heights.

Refrigerant Flow Rate, Temperatures, and Pressures

The MFHP unit features two refrigerant line sets, connecting the ODU to the AHU and to the WT. Well insulated 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

External static pressure (ESP) was measured using a four point array in the AHU connector ducts for both the SA and RA using two of the channels on a DG8 differential pressure sensor each referenced to the indoor chamber pressure. Further static pressure measurements were recorded along the ODU interior cavity between the coil and the fan. All pressures were measured with an Energy Conservatory differential pressure gauge. The pressure of the outdoor chamber was also measured using an additional atmospheric absolute pressure gauge.

The sensor types, model numbers, and accuracy are listed in Table 11.

Table 11: Table of sensors and their model numbers.

Air Pressure Sensors	Channel	Model (accuracy)
Outdoor chamber air pressure	P_A_OC	Energy Conservatory APT (±1% of reading)
ODU inlet air pressure	P_A_ODU_IN	
ODU internal cavity air pressure	P_A_ODU_CAV	
ODU exhaust air pressure	P_A_ODU_EXH	
External static pressure*	P_A_AHU_ESP	
AHU inlet air pressure	P_A_AHU_IN	
AHU outlet air pressure	P_A_AHU_OUT	
Indoor chamber air pressure	P_A_IC	
Outdoor Unit Air Temperature RTDs	**Channel**	**Model (accuracy)**
ODU air inlet temp side 1 upper-left	T_A_ODU_RTU 1	Omega 100Ω class A platinum RTD-805 ±(0.27+0.0036*T)°F
ODU air inlet temp side 1 middle-left	T_A_ODU_RTU 2	
ODU air inlet temp side 1 bottom-left	T_A_ODU_RTU 3	
ODU air inlet temp side 1 upper-middle	T_A_ODU_RTU 4	
ODU air inlet temp side 1 middle-middle	T_A_ODU_RTU 5	
ODU air inlet temp side 1 bottom-middle	T_A_ODU_RTU 6	
ODU air inlet temp side 1 upper-right	T_A_ODU_RTU 7	

Air Pressure Sensors	Channel	Model (accuracy)
ODU air inlet temp side 1 middle-right	T_A_ODU_RTU 8	
ODU air inlet temp side 1 bottom-right	T_A_ODU_RTU 9	
ODU air inlet temp side2 top	T_A_ODU_RTU 10	
ODU air inlet temp side2 middle	T_A_ODU_RTU 11	
ODU air inlet temp side2bottom	T_A_ODU_RTU 12	
Water Temperatures	**Channel**	**Model (accuracy)**
DHW tank water temp in	T_W_WT_In	Omega SA1-RTD-4W (DIN Class A: ±0.15+0.002t, where t=temp [°C])
DHW tank water temp out	T_W_WT_Out	Omega SA1-RTD-4W
DHW BPHX water temp in	T_W_HX_In	
DHW BPHX water temp out	T_W_HX_Out	
DHW internal tank temps 1–7 (bottom to top)	T_W_WT_1-7	Omega HSRTD-3-100-A
Air Dew Point and Temperature	**Channel**	**Model (accuracy)**
Outdoor chamber / ODU inlet	DP_A_ODU_IN T_A_ODU_IN	GE Optisonde Chilled Mirror Hygrometer (±0.27°F dry-bulb, ±0.36°F dew point)
ODU exhaust	DP_A_ODU_EXH T_A_ODU_EXH	GE Optisonde Chilled Mirror Hygrometer (±0.27°F dry-bulb, ±0.36°F dew point)

Air Pressure Sensors	Channel	Model (accuracy)
Indoor chamber / AHU return (inlet)	DP_A_AHU_IN T_A_AHU_IN	Temperature 4-point array – GE Optisonde Chilled Mirror Hygrometer (±0.27°F dry-bulb, ±0.36°F dew point) Vaisala HMP110 (±0.36 to ±0.72°F) 2x Omega RTD PR-10L-3-100-1-4-6 (IEC Class A: ±0.15+0.002t, where t=temp [°C]))
AHU supply air (outlet)	DP_A_AHU_OUT T_A_AHU_OUT	Temperature 4-point array – GE Optisonde Chilled Mirror Hygrometer (±0.27°F dry-bulb, ±0.36°F dew point) Vaisala HMP110 (±0.36 to ±0.72°F) 2x Omega RTD PR-10L-3-100-1-4-6
Electrical Power	**Channel / Input**	**Model (accuracy)**
ODU+AHU PowerScout	PS_ODU / RS-485 Converter port d	Dent PowerScout PS3037 (±0.7%)
AHU PowerScout	PS_AHU / RS-485 Converter port d	
Water tank PowerScout	PS_WT / RS-485 Converter port d	

Air Pressure Sensors	Channel	Model (accuracy)
Water tank backup electric resistance heater PowerScout	PS_BH / RS-485 Converter port d	
Water Tank	**Channel**	**Model (accuracy)**
Water flow rate	FV_W_WT	Omega FTB4607 (0.66-13.0 GPM±1.5% of reading, below 0.66 GPM ±2% of reading)
Refrigerant Line Pressures	**Channel**	**Model (accuracy)**
AHU liquid line	P_R_AHU_LL	ClimaCheck Pressure Transducer 22S, 50 Bar, Teflon seal, 1-5 V (<1% Full Scale)
Refrigerant Pressure – AHU vapor line	P_R_AHU_VL	
Refrigerant Pressure – Water tank liquid line	P_R_WH_LL	
Refrigerant Pressure – Water tank vapor line	P_R_WH_VL	
Refrigerant Line Temperatures	**Channel**	**Model (accuracy)**
Refrigerant Temperature – AHU liquid line	T_R_AHU_LL	Omega SA1-RTD-4W (DIN Class A: ±0.15+0.002t, where t=temp [°C])
Refrigerant Temperature – AHU vapor line	T_R_AHU_VL	
Refrigerant Temperature – Water tank liquid line	T_R_WH_LL	
Refrigerant Temperature – Water tank vapor line	T_R_WH_VL	

*External static pressure is the difference between AHU outlet air pressure and inlet air pressure ($P_{out} - P_{in}$).

Laboratory Testing Methods

Setup

The WCEC laboratory environmental chambers, an outdoor chamber and an indoor chamber, have independent air ducts and controls to hold each chamber at different air conditions for HVAC testing. The Variable-speed MFHP equipment was set up and commissioned in the environmental chambers including design, fabrication and installation of ducts to connect the air handler to the chambers. The sensors were selected, purchased, and installed on the equipment. The manufacturer sent two representatives to support equipment setup and commissioning through the refrigerant charging and preliminary testing.

The ODU was installed in the outdoor chamber to mimic the environmental effects of being in an unconditioned space, while the water tank and AHU were installed in the indoor chamber to mimic the environmental effects of being placed in a conditioned space. The AHU was placed in the indoor chamber because it is designed to be in conditioned space common for apartments and European homes, so it does not have insulation on the return air section. Return air (RA) and supply air (SA) from the AHU were ducted to the indoor chamber. Insulated copper refrigerant lines, city entering and domestic hot water leaving water hoses, and 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.

Figure 5 shows a schematic of the equipment in the environmental chambers with sensor types and positions labeled. Figure 6 and Figure 7 show the Variable-speed MFHP equipment in the environmental chambers with instrumentation installed.

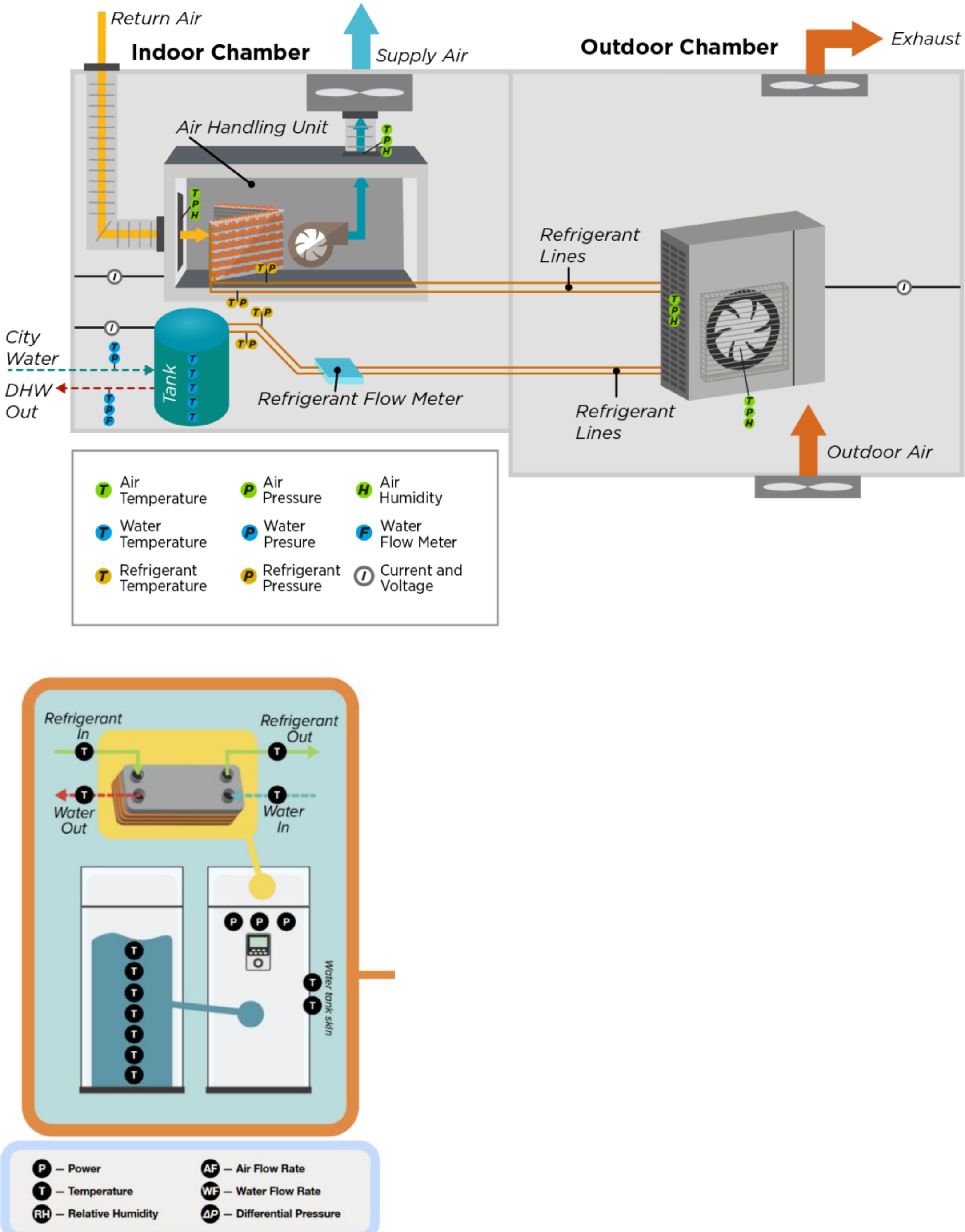

Figure 5: Schematic of the test setup showing the unit in the chambers and the position of sensors.

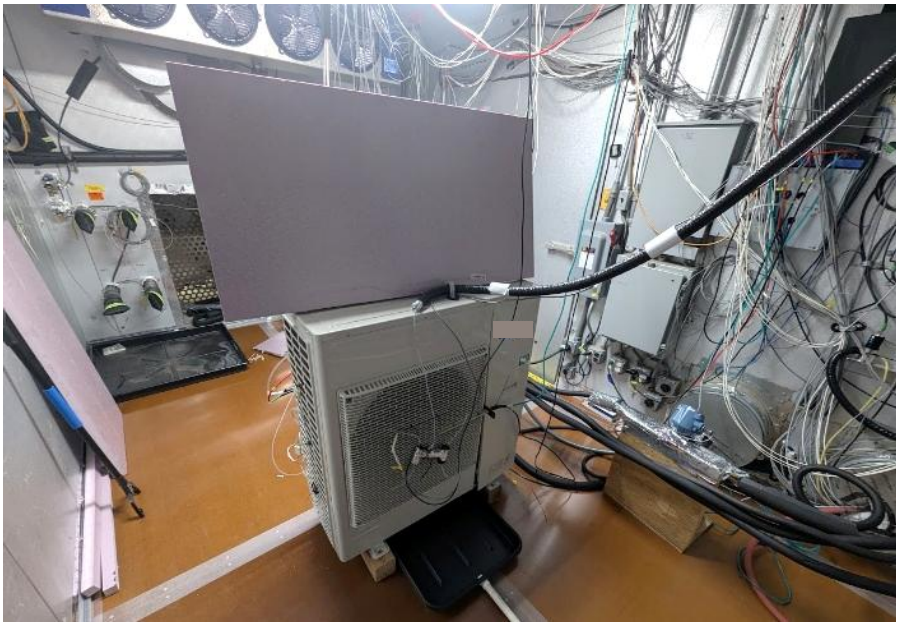

Figure 6: Variable-speed MFHP outdoor unit installed and instrumented in the outdoor chamber, ODU exhaust side visible.

Figure 7: Variable-speed MFHP air handler unit (back horizontal grey section) and domestic hot water tank (front right with front cover off and white insulation visible) installed and instrumented in the indoor chamber.

The outdoor unit had more than the manufacturer required minimum clearance on all sides. The outdoor unit was oriented so that the outdoor chamber air flow traveled from the inlet side towards the outlet side and a section of foam board insulation was positioned on top of the ODU to avoid air from recirculating from the outlet side of the ODU back to the inlet side. The 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. This cavity pressure was monitored to identify any restriction from frost accumulation in heating tests.

Pressures across the AHU were monitored and used to establish appropriate indoor chamber conditioning loop fan settings to maintain target operating external static pressure (ESP) according to testing standards based on supply air flow rate, using the fan affinity law square exponential relationship (AHRI 2023).

Testing

For each space conditioning test, the team ran the MFHP until we observed a 30-minute duration of steady-state operation. For the WH tests, the team held the environmental chambers at steady ambient conditions for the duration of a full water heating cycle. 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 evaporator 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 direct evaporative humidifier with supplemental humidity added with a steam humidifier when required.

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) as well as two additional outdoor conditions with one indoor condition. For each of these combinations the tests were repeated with compressor speed held constant at three or more different speeds.

Preliminary test runs identified that under manufacturer controls, the Variable-speed MFHP unit maintains a relatively high air flow rate even at low-speed compressor operation. This matches results from an ongoing project in the UC Davis EEI Smart Home that tested the same equipment in a real building. The manufacturer has confirmed that this is a normal operation and recommended that testing be performed at high fan speed for cooling tests. Test runs with intermediate compressor speed, across three different AHU fan speeds, with the standard indoor air dry-bulb temperature and low humidity for dry coil cooling showed a relatively small sensitivity to fan speed for capacity and COP with linear behavior.

Space Heating (SH)

For SH mode, 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. For each of these combinations the tests were repeated with compressor speed held constant at three or more different speeds. The plan also included a set of tests at intermediate compressor speed, standard outdoor and indoor conditions, across three different AHU fan speeds, with the standard indoor air dry-bulb temperature for heating showed a relatively small sensitivity to fan speed for capacity and COP with linear behavior.

Water Heating (WH)

For this particular model of variable-speed MFHP, as the manufacturer external water tank sensor temperature approaches the lower of either the setpoint or approximately 115°F, the manufacturer controls reduce the compressor speed. When the external water tank sensor gets to 129°F, the heat pump WH ends and if the setpoint is higher the equipment uses the supplemental electric resistance heater to reach the final setpoint temperature. This suggests that for this particular model of variable-speed MFHP, using water temperature setpoints above 129°F would decrease efficiency and increase energy consumption.

For WH mode testing, the team ran tests at the OA conditions selected in the test plan. Inlet water was run through the WT until the average tank temperature across all seven vertically spaced sensors was 65°F, WH was initiated, and the tank was heated until the heat pump cutoff temperature. For analysis of whole cycles, the WH cycle was defined to end at compressor cutoff . In this report, we define tank temperature as the average of two sensors that provide the interior tank temperature at the height of the manufacturer temperature sensor that the equipment uses to control water heating. During WH, the manufacturer controls managed compressor speed and internal water tank unit water pump flow rate through the brazed plate refrigerant to water heat exchanger and the immersed coil water-to-water heat exchanger. The controls settings were set to standard compressor speed and the default standard operation controls. Tests were run with the 3-kW supplemental electric resistance heater disabled.

There are other WH controls options, including variable capacity water heating and smart controls mode, that slow down and delay heat pump water heating respectively. The team decided not to test these modes because they could reduce the amount of hot water that can be delivered and are less likely to satisfy US end users. The team also decided not to test an installer-selectable fast boiling rapid water heating mode, because it is not the default and the standard controls mode can supply enough hot water for typical US residential use cases.

Heat Recovery – Simultaneous Space Cooling and Water Heating Mode

For Heat Recovery mode testing, the team ran tests at the environmental conditions selected in the test plan. Heat Recovery mode tests controlled the IA conditions at SC levels. In Heat Recovery mode there is no intentional heat exchange with the outdoors, so OA conditions were held at an arbitrary 95°F db with a range of IA conditions of db/wb 80/67, 78/64, and 75/60°F. The team measured MFHP performance over a full WH cycle from 65°F average tank temperature to compressor cutoff, while simultaneously cooling return air at each of the IA conditions under the manufacturer compressor speed controls. The space conditioning thermostat setpoint was set below but close to the IA return air temperature, as would be expected in real installations when the HP equipment is correctly sized. The manufacturer controller selects compressor speed near the minimum speed when the thermostat setpoint is close to the return air temperature. An additional Heat Recovery test was run at OA 105°F db with IA db/wb 80/67, and thermostat setpoint far below the IA return air temperature where the manufacturer controls select a high compressor speed for comparison.

Bi-Heating – Simultaneous Space Heating and Water Heating

For Bi-Heating mode testing, the team ran tests at the environmental conditions selected in the test plan. Bi-Heating mode tests controlled the OA conditions at the same three levels as for SH, with IA conditions at 70°F dry bulb. The team measured MFHP performance over WH cycles from an average water temperature of 80 up to 120°F while simultaneously heating return air at each of the OA conditions. The manufacturer-controls-selected compressor speed and internal water pump speed.

Defrost (DEF)

The variable-speed MFHP has demand defrost control to initiate a DEF cycle when it is needed. DEF cycles occurred during SH and WH tests. Before these events, decreasing ODU cavity pressure values indicated an accumulation of frost occluding the condenser coils. When defrost was initiated, the AHU fan stopped, and AHU power dropped to zero.

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:

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

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.

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

Where:
$h_{SA@RA\_DP}$ is the supply air enthalpy at the same dew point as the return air
h_{RA} is the return air enthalpy
$\dot{m}_{SA}$ is the supply air mass flowrate

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 Heating First-Hour Rating Testing

After completing environmental chamber tests to measure the performance of each of the MFHP's operational modes, the team performed an 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 (US DOE 2024). Table 12 below summarizes the required test conditions.

Table 12: Test conditions for FHR test from the 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)
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: US 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 125°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 water tank setpoint was set to 125°F setpoint with a 14.4°F (8°C) deadband. Once the WT was fully preheated and the tank unit controls 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 then calculated from the array of temperature sensors installed at interval heights within the WT. The team assessed the maximum outlet water temperature of the draw from live plots, and terminated the water draw once the outlet water temperature dropped 15°F below the maximum observed, starting 15 seconds after the start of the draw. Subsequent water draws were initiated after the WT had reheated and the controls stopped calling for WH. These draws were terminated when the outlet water temperature dropped 15°F below the maximum outlet temperature observed starting 15 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.

Water Tank Thermal Storage and Capacity

The water tank storage capacity (WTH) was determined using volume, density, and specific heat properties of water and tank components according to measured volumes and equipment specifications. The WT temperatures were measured using a vertical array of seven sensors to capture temperature stratification. 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 stainless steel of the tank walls and water-to-water heat exchanger tubes.

Equation 5: Water tank heat storage capacity.

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

Where for each control volume i

$$H_{WH,i} = (m_{WT,i} \cdot C_{p,water} + m_{steel,i} \cdot C_{p,steel}) \cdot T_{WT,i}$$

$m_{WT,i}$ is the mass of water for control volume i
$m_{steel,i}$ is the mass of tank wall and heat exchanger tube stainless steel for control volume i
$C_{p,water}$ is the specific heat of water
$C_{p,steel}$ is the specific heat of stainless steel
$T_{WT,i}$ 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 added up these rates of change to quantify heat capacity gain of the entire control volume.

Equation 6: Water tank heating capacity.

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

Simultaneous Water Heating and Space Cooling

For Heat Recovery mode for the MFHP, both the SC and WH 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.

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

Where

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

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

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; 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 Heat Recovery mode was calculated as follows:

Equation 8: Refrigerant water heating capacity.

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

Water Tank Skin Heat Loss

Another element 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 in watts per unit area to ambient temperature ($W/m^2 \cdot K$), 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 9: Newton's law of cooling heat loss.

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

Equation 10: Conservation of energy.

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

Equation 11: 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 delivered over electrical energy consumed—or, inversely, the electric input ratio (EIR). Both are commonly used: COP intuitively increases as energy efficiency increases and EIR increases with power draw, which can be more intuitive when analyzing energy use.

Equation 12: Coefficient of performance and electric input ratio.

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

and

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

The COP and EIR were calculated for each of the MFHP modes using their respective capacities (calculated previously) and power consumption. Thus, for the Heat Recovery 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 13: 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 Curves

Future projects developing a measure package for MFHP products will use EnergyPlus whole-building energy simulation software to predict energy savings. This project will not perform EnergyPlus modelling but will develop performance curve inputs for future projects.

There is an existing EnergyPlus object called the "Air Source Integrated Heat Pump" with coil system object made for an air-source integrated heat pump (ASIHP) model developed by Shen et al. (Shen et al 2017). This object can model MFHP modes with a pumped water heating condenser similar to the Variable-speed MFHP system being tested that has a refrigerant-to-water brazed plate heat exchanger (BPHX) with a pumped internal closed loop of water that circulates through the heat exchanger and then through a closed coil inside the hot 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. Future projects will either work with the EnergyPlus developers to tweak the ASIHP coil system object in EnergyPlus to be more expandable to incorporate additional operational modes or develop shareable workarounds using EnergyPlus energy management system capabilities. This would enable both our team and a wide range of other users to accurately simulate different kinds of MFHPs with various mode architectures.

EnergyPlus models the performance of the equipment using performance curves for the heat exchangers and connected systems, referenced to as coils, for each of the equipment's operating modes. These performance curves are indexed based on the rating compressor speed and conditions, i.e., the evaporator and condenser environmental conditions. The performance curves result in a factor that is multiplied by performance at the rated condition. This allows the capacity or COP to be adjusted for different outdoor conditions, indoor conditions. This also allows the capacity and COP to be adjusted for different equipment operation including compressor speeds, fan speeds, and hot water tank water pump speeds.

For variable-speed space cooling, at each compressor speed the capacity and Energy Input Ratio (EIR = 1/COP) are modified using performance curves.

Equation 14: SC capacity performance curve as a function of outdoor db and indoor wb

$$TotCapTempModFac = a + b \cdot WB_i + c \cdot {WB_i}^2 + d \cdot DB_o + e \cdot {DB_o}^2 + f \cdot WB_i \cdot DB_o$$

Equation 15: SC COP performance curve as a function of outdoor db and indoor wb

$$EIRTempModFac = a + b \cdot WB_i + c \cdot {WB_i}^2 + d \cdot DB_o + e \cdot {DB_o}^2 + f \cdot WB_i \cdot DB_o$$

Where:

WB_i is the wet-bulb temperature of the air entering the indoor cooling coil (°C)

DB_o is the dry-bulb temperature of the air entering the condenser coil (°C)

$a \dots f$ are the regression curve-fit coefficients, different for each equation.

Equation 16: SC capacity performance curve as a function of AHU air flow rate

$$TotCapAirFlowModFac = a + b \cdot ff_a + c \cdot {ff_a}^2 + d \cdot {ff_a}^3$$

Equation 17: SC EIR performance curve as a function of AHU air flow rate

$$EIRAirFlowModFac = a + b \cdot ff_a + c \cdot {ff_a}^2 + d \cdot {ff_a}^3$$

Where:

ff_a is the actual air mass flow rate/design air mass flow rate, at the rated condition fan speed level

$a \dots d$ are the regression curve fit coefficients, different for each equation. If no data for correction is available, the user can simply use a = 1.0 and set the other coefficients to 0.0.

For variable-speed space heating, at each compressor speed the capacity and EIR are modified using performance curves.

Equation 18: SH capacity performance curve as a function of outdoor db and indoor db

$$TotCapTempModFrac = a + b \cdot DB_i + c \cdot {DB_i}^2 + d \cdot {DB_o}^2 + f \cdot DB_i \cdot DB_o$$

Equation 19: SH EIR performance curve as a function of outdoor db and indoor db

$$EIRTempModFunc = a + b \cdot DB_i + c \cdot {DB_i}^2 + d \cdot DB_o + e \cdot {DB_o}^2 + f \cdot DB_i \cdot DB_o$$

Where:

DB_i is the dry-bulb temperature of the air entering the indoor cooling coil (°C)

DB_o is the dry-bulb temperature of the air entering the outdoor coil (°C)

$a \dots f$ are the regression curve-fit coefficients, different for each equation.

Equation 20: SH capacity performance curve as a function of AHU air flow rate

$$TotCapAirFlowModFrac = a + b \cdot ff_a + c \cdot ff_a^{\ 2} + d \cdot ff_a^{\ 3}$$

Equation 21: SH EIR performance curve as a function of AHU air flow rate

$$EIRAirFlowModFunc = a + b \cdot ff_a + c \cdot ff_a^{\ 2} + d \cdot ff_a^{\ 3}$$

Where:

ff_a is the actual air mass flow rate/design air mass flow rate, at one speed level

$a \ldots d$ are the regression curve fit coefficients, different for each equation. If no data is available, the user can simply use a = 1.0 and set the other coefficients to 0.0.

For variable-speed water heating, the capacity and COP are modified using performance curves.

Equation 22: WH capacity performance curve as a function of water temperature entering the BPHX and outdoor db

$$WHCapacityFuncTemp = a + b \cdot T_e + c \cdot T_e^{\ 2} + d \cdot EWT + e \cdot EWT^2 + f \cdot T_e \cdot EWT$$

Equation 23: WH COP performance curve as a function of water temperature entering the BPHX and outdoor db

$$WHCOPFuncTemp = a + b \cdot T_e + c \cdot T_e^{\ 2} + d \cdot EWT + e \cdot EWT^2 + f \cdot T_e \cdot EWT$$

Where:

T_e is either the evaporator wet-bulb or dry-bulb temperature

EWT is the entering water temperature of the condenser

Additional performance curves can adjust performance based on the water pump flow rate through the BPHX and outdoor air flow rate through the outdoor coil. The manufacturer controls actively manage both of these flow rates without options to control them externally. These additional performance curves will set the coefficient a = 1.0 and set the other coefficients to 0.0, and the EnergyPlus options will be selected to include these effects in the capacity and COP as a function of temperature curves above.

For variable-speed Heat Recovery and Bi-Heating modes, at each compressor speed the capacity and COP performance curves have similar types of equations, for more details, see Appendix A.

Results

The maximum power consumption over all tests was 3.97kW. This is less than the typical 4.5 to 5kW electric resistance heater in typical separate heat pump water heaters and single speed heat pumps. Key results are summarized in Table 13.

Table 13 Summary of key results

Mode and Conditions (Temperatures °F db/wb)	Key Results	
Space Cooling **OA 95, IA 80/67**	Compressor 53Hz Capacity 7.28 kW COP 3.42	Compressor 15Hz Capacity 1.91 kW COP 3.61
Space Heating **OA 47/43, IA 70**	Compressor 49Hz Capacity 7.28 kW COP 3.96	Compressor 15Hz Capacity 2.39 kW COP 4.69
Water Heating **OA 47/43, WT from 65 to 127**	Compressor 65Hz Capacity 7.25 kW COP 3.24	WT heating time 48min (WT from 65 to 120°F) FHR 86.4 gal
Heat Recovery **OA 95, IA 80/67,** **WT from 65 to 125**	Compressor 35Hz Capacity SC 5.67 kW Capacity WH 6.16 kW Combined COP 7.92	33% Energy savings compared to separate SC and WH cycles (WT from 110 to 120°F)
Bi-Heating **OA 47/43, IA 70** **WT from 80 to 120**	Compressor 78Hz Capacity SH 7.0 kW Capacity WH 4.5 kW Combined COP 3.36	
Defrost **OA 37/35**	590 seconds Max electric power 3.18 kW	WT 128 drop to 119

Space Cooling (SC)

Figure 8 and Figure 9 show the measured SC capacity and COP for the two-ton EU-rated variable-speed MFHP in SC mode with high AHU supply air fan speed. At OA db 95°F and IA db/wb 80°F /67°F , the variable-speed MFHP SC mode with full compressor speed of 53Hz provided a total cooling capacity of 7.28 kW (24.84 k Btu/h, 2.07 ton) at a COP of 3.42 with sensible heat ratio of 0.699. This performance is slightly better than the EU ratings test data of 7.1kW cooling capacity at a COP of 3.40. At these same OA and IA conditions, the variable-speed MFHP with minimum compressor speed of 15 Hz provided a cooling capacity of 1.91 kW (6.53 k Btu/h, 0.54 ton) at a COP of 3.61, and at 28 Hz provided a cooling capacity of 4.60 kW (15.68 k Btu/h, 1.31 ton) at a COP of 4.32.

Test runs across a range of compressor speeds show that from minimum to full speed the cooling capacity varies from 0.73 to 2.2 tons, and the COP varies from 4.86 to a high of 5.16 and then down to 3.68 at 95db outdoor air temperature and 80db 67wb indoor temperature.

The previously tested single-speed MFHP in SH mode, at OA db 95°F and IA db/wb of 80°F/67°F, had a capacity of 45.3 k Btu/h with a COP of 3.57. At these conditions, the variable-speed MFHP SC mode achieved higher COP than the single speed MFHP for low compressor speeds, but lower COP for full compressor speed. Comparing the COP of the variable-speed MFHP to the single speed MFHP across the range of compressor speeds gives estimated energy savings of 6 percent at minimum speed of 15Hz, 21 percent at best COP speed of 28Hz, and −4 percent at full speed of 53Hz.

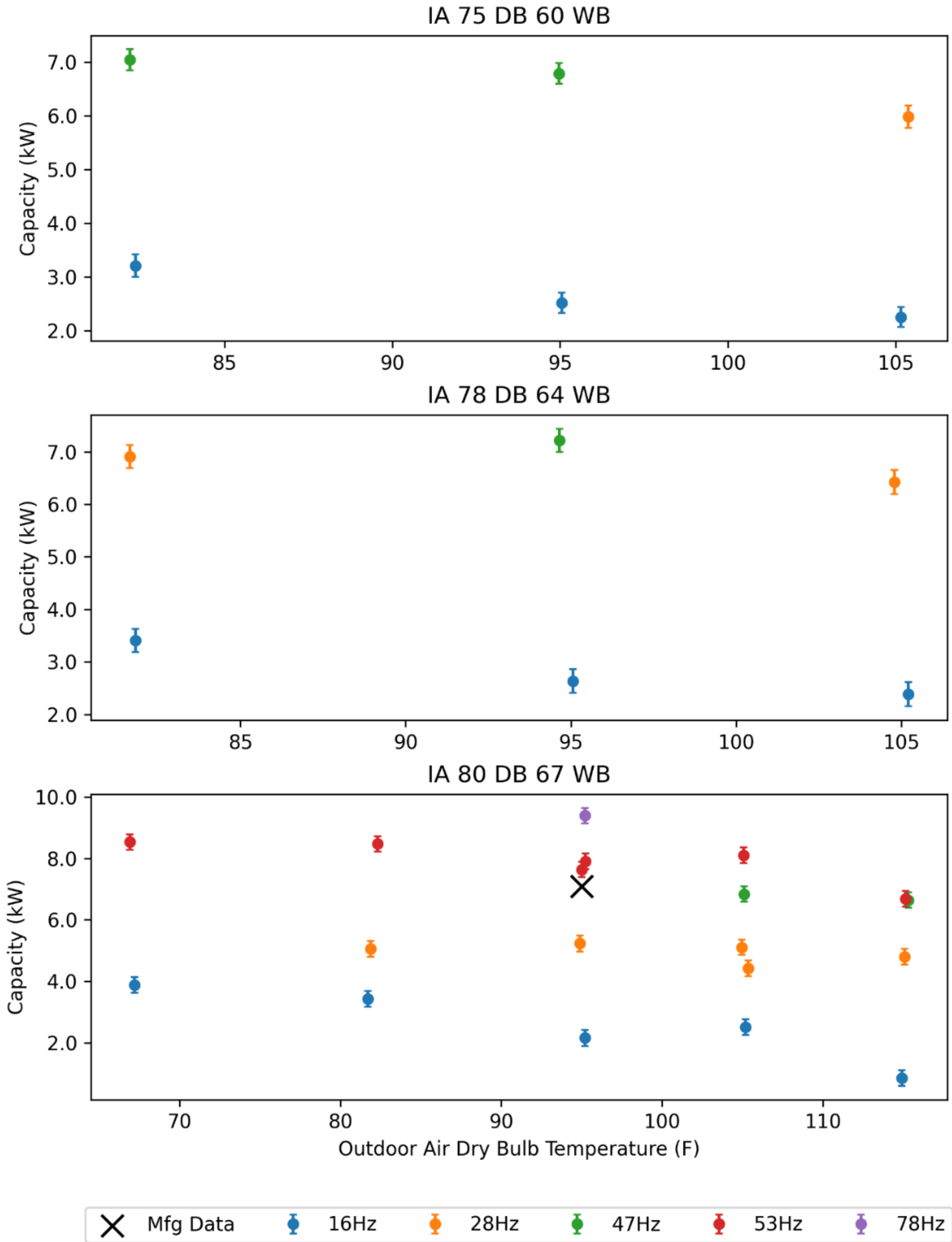

Figure 8: Measured MFHP SC capacity across compressor speeds at different outdoor and indoor conditions. *

*X shows the manufacturer-published EU rating performance data (source: public sales brochure). Bars show standard deviation uncertainty from sensor accuracy.

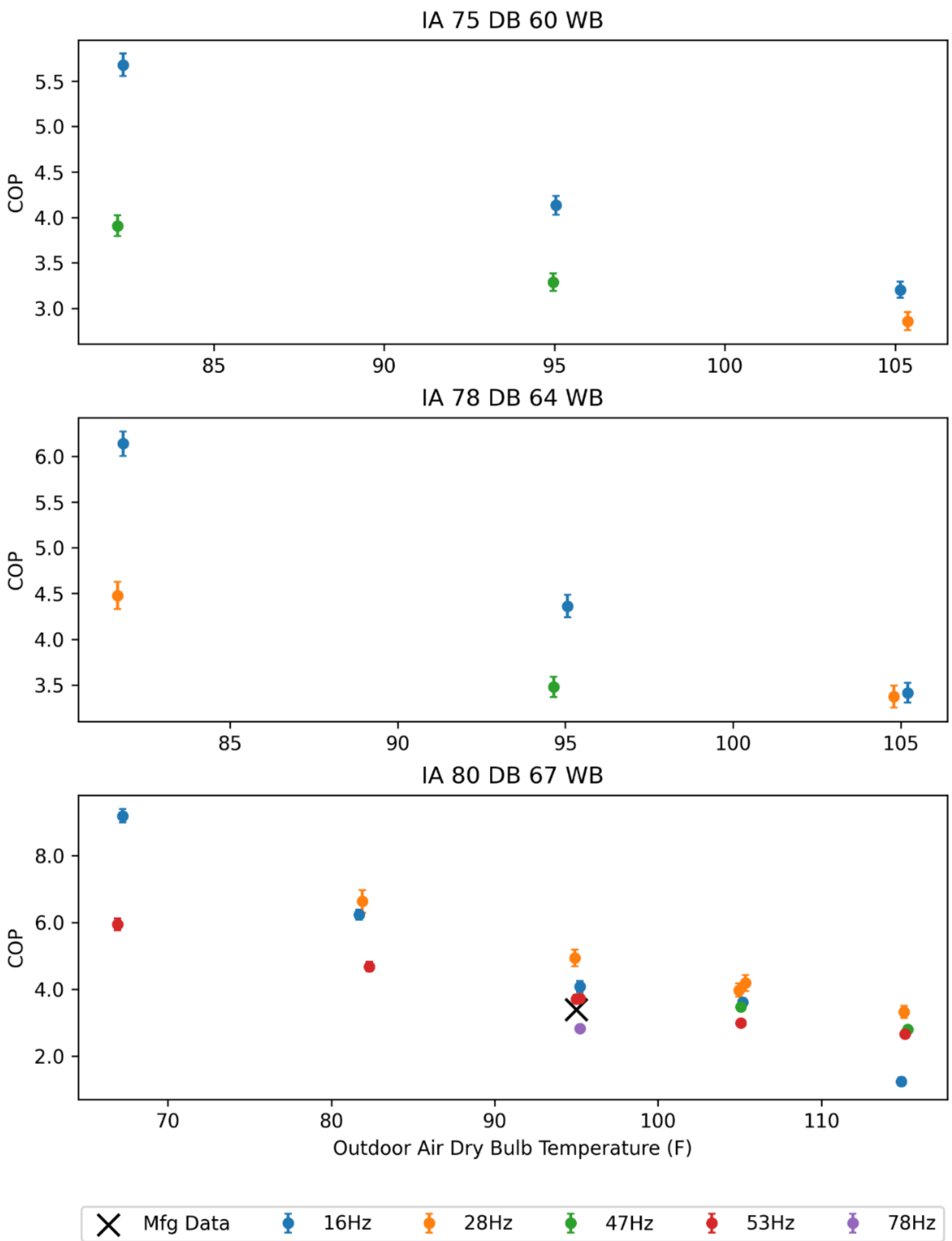

Figure 9: Measured MFHP SC COP across compressor speeds at different outdoor and indoor conditions. *

*X shows the manufacturer-published EU rating performance data (source: public sales brochure). Bars show standard deviation uncertainty from sensor accuracy.

Measurements show that changing the AHU supply air flow rate results in relatively small changes of MFHP equipment cooling capacity and COP with a linear trend, for operation at compressor speed of 30 Hz and 95°F OA db and IA db/wb of 80/67°F, as shown in Figure 10.

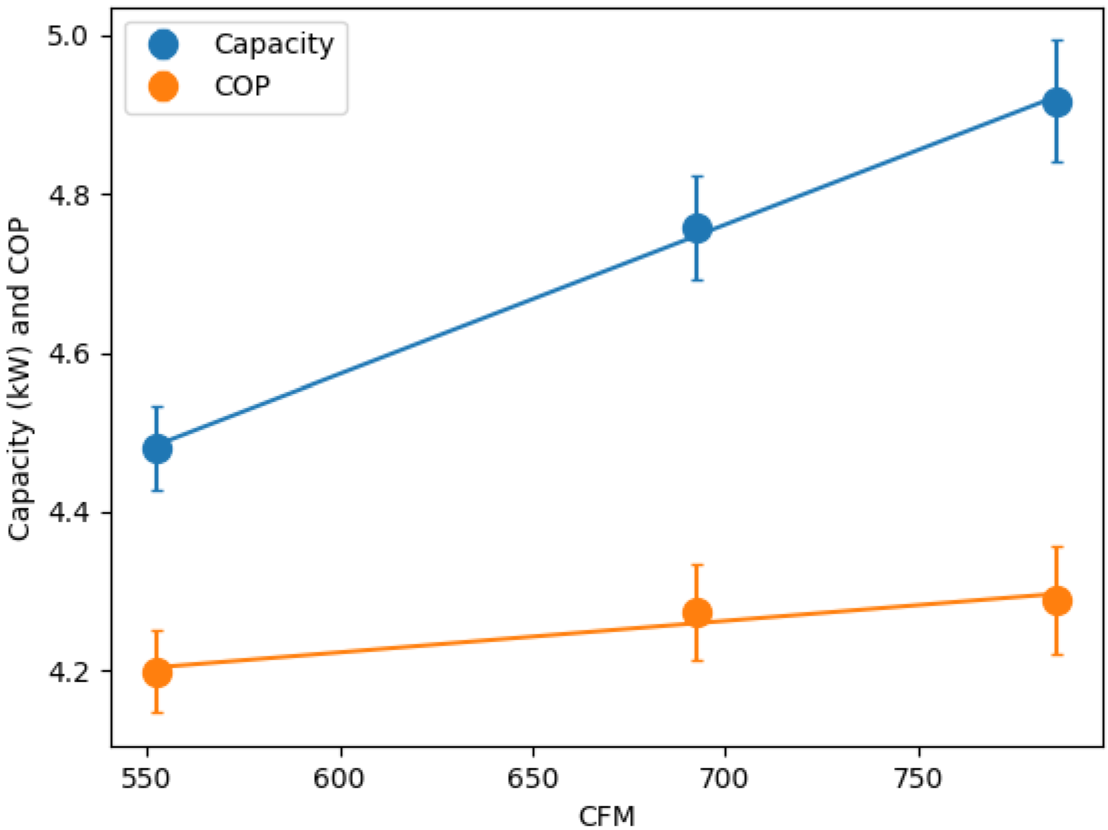

Figure 10: Air handler supply air flow rate (cfm) versus cooling capacity (kW) and COP.*

*Compressor speed 30 Hz at 95db outdoor air temperature and 80db 67wb indoor temperature. Bars show standard deviation uncertainty from sensor accuracy.

Space Heating (SH)

Figure 11 and Figure 12 show the lab-test-measured SH capacity for the two-ton, EU-rated variable-speed MFHP in SH mode with high AHU supply air fan speed. At OA db/wb 47/43°F and IA db 70°F, the variable-speed MFHP SH mode with full compressor speed of 49Hz provided a heating capacity of 7.28 kW (24.8 k Btu/h, 2.07 ton) at a COP of 3.96. This performance is slightly better than the EU ratings test data of 7.1kW heating capacity at a COP of 3.90. At these same OA and IA conditions, the variable-speed MFHP with minimum compressor speed of 15 Hz provided a heating capacity of

2.39 kW (8.16 k Btu/h, 0.68 ton) at a COP of 4.69, and at 25 Hz provided a heating capacity of 3.83 kW (13.2 k Btu/h, 1.07 ton) at a COP of 4.90.

For all tested indoor conditions, the variable-speed MFHP initiated defrost to maintain reliable performance even at low outdoor temperature conditions. The DEF operation is analyzed further in the DEF mode section.

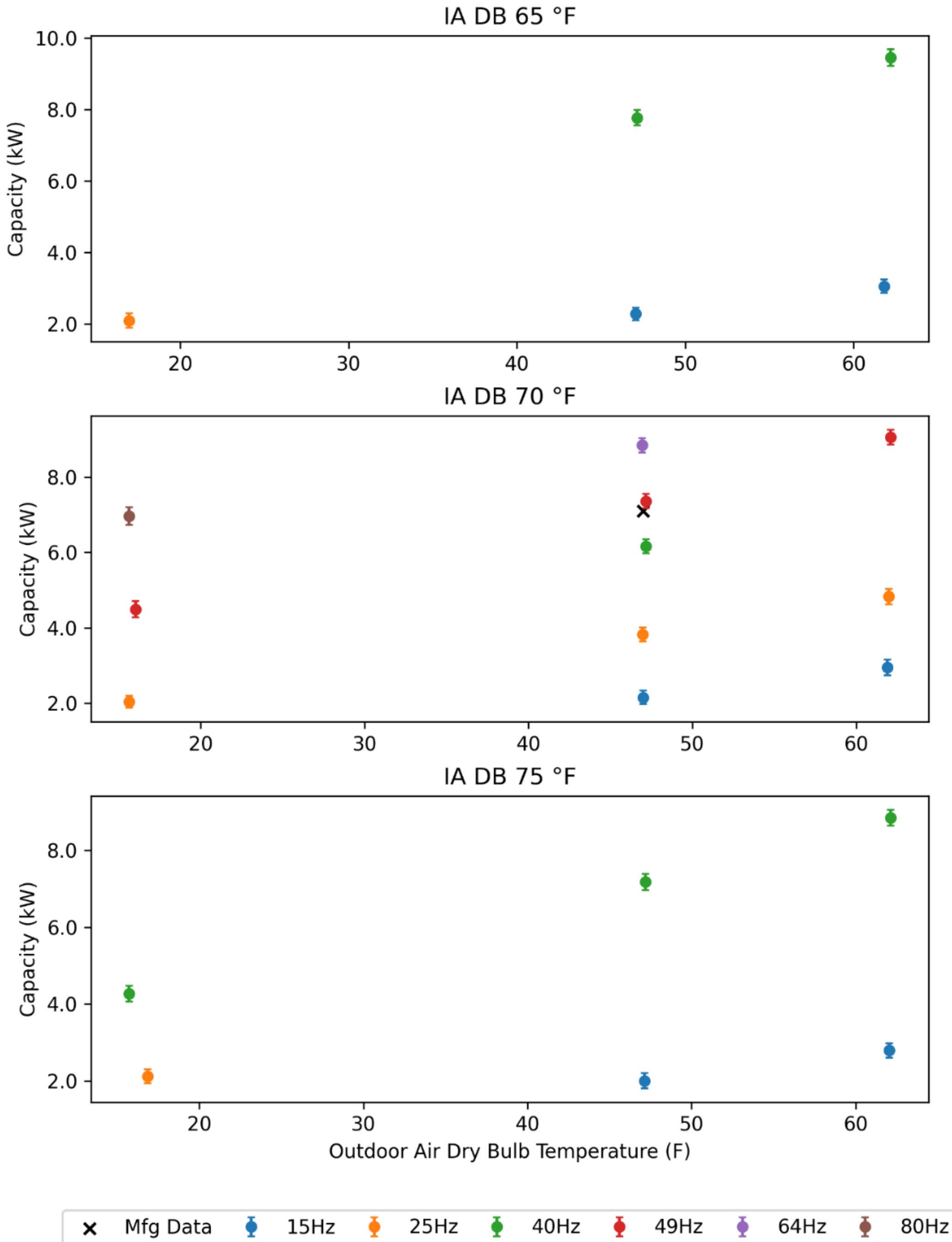

Figure 11: Measured MFHP SH capacity at three different indoor db 65, 70, 75°F across outdoor conditions for six compressor speeds. *

*X shows the manufacturer-published EU rating performance data (source: public sales brochure). Bars show standard deviation uncertainty from sensor accuracy.

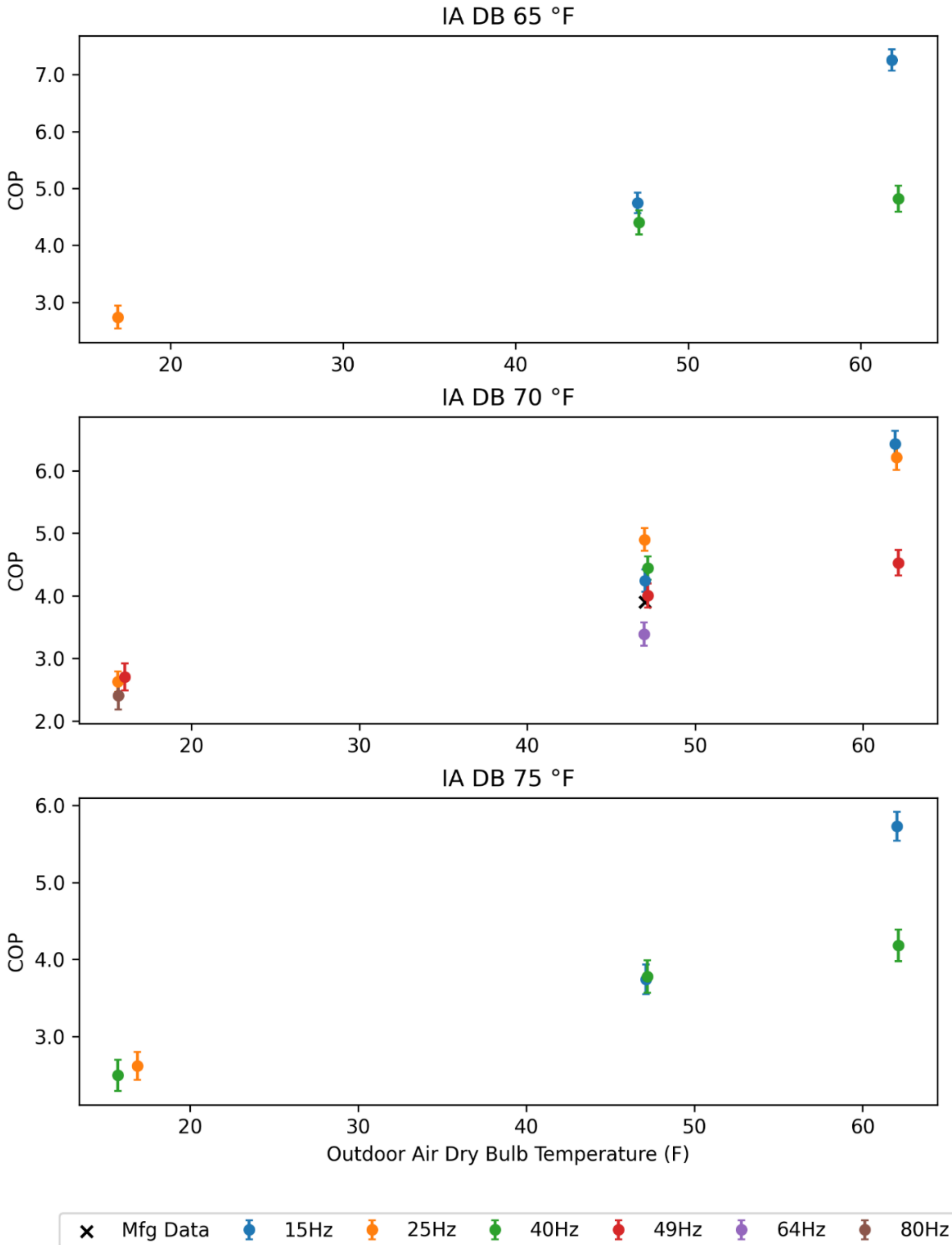

Figure 12: Measured MFHP SH COP at three different indoor db 65, 70, 75°F across outdoor conditions for six compressor speeds. *

*X shows the manufacturer-published EU rating performance data (source: public sales brochure). Bars show standard deviation uncertainty from sensor accuracy.

The previously tested single speed MFHP in SH mode, at OA db/wb 47°F /43°F and IA db of 70°F had a capacity of 49.9 k Btu/h with a COP of 3.48. At these conditions, the variable-speed MFHP SH mode achieved higher COP than the single speed MFHP. Comparing the COP of the variable-speed MFHP to the single speed MFHP across the range of compressor speeds gives estimated energy savings of 35 percent at minimum speed of 15Hz, 41 percent at best COP speed of 25Hz, and 14 percent at full speed of 49Hz.

Water Heating (WH)

Water heating capacity was calculated from the rate of change in the thermal energy stored in the water tank called the hot water tank capacity ($\dot{H}_{WTH,tot}$).

Under manufacturer controls, the variable-speed MFHP adjusted compressor speed as the WT temperatures increase, starting at low speed, increasing to a higher speed dependent on OA conditions and remaining constant for much of the WH cycle, and then reducing to lower speed as the WT temperature approached the compressor cutoff temperature. Taking the whole cycle average of WH for OA db/wb 47/43°F from tank average of 65°F to the compressor cutoff of 127°F shows a capacity of 7.50 kW (25.6 kBtu/h) with a COP of 3.24.

During the constant compressor speed segment of the WH test cycles, the water heating capacity and COP were calculated over water tank temperature bins centered at 90, 100, and 110°F, with bin width of ±5°F to cover the range of typical operation, as shown in Figure 13 and Figure 14. As expected for all heat pump water heating equipment, the water heating capacity and COP decreases as the tank temperature rises. At OA db/wb 47°F /43°F and tank temperature bin centered at 110°F the WH mode demonstrated a capacity of 8.4kW (28.7 k Btu/h) at a COP of 2.92. At OA db/wb 95/75°F and tank temperature bin centered at 110°F the WH mode demonstrated a capacity of 9.6 kW (32.8 k Btu/h) at a COP of 5.1. At very low OA 17°F db and high tank temperature bin centered at 110°F, WH mode demonstrated a capacity of 6.4 kW (21.8 k Btu/h) at a COP of 1.86. Even at these very cold OA conditions, the variable-speed MFHP still had higher capacity and better efficiency than the typical electric resistance water heater, with a capacity of 4.5kW and a COP slightly less than one.

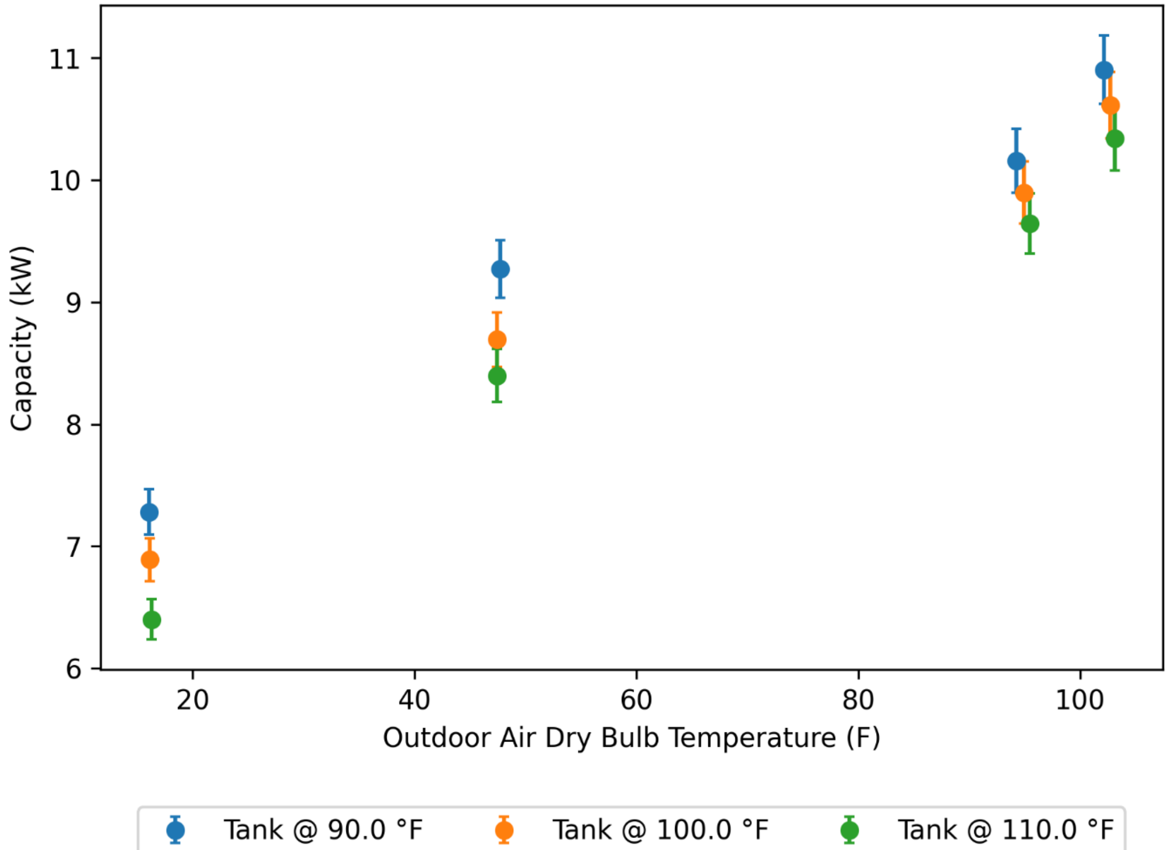

Figure 13: Measured variable-speed MFHP WH capacity at different OA conditions for WT temperature bins in the constant compressor speed segment of the WH cycles.

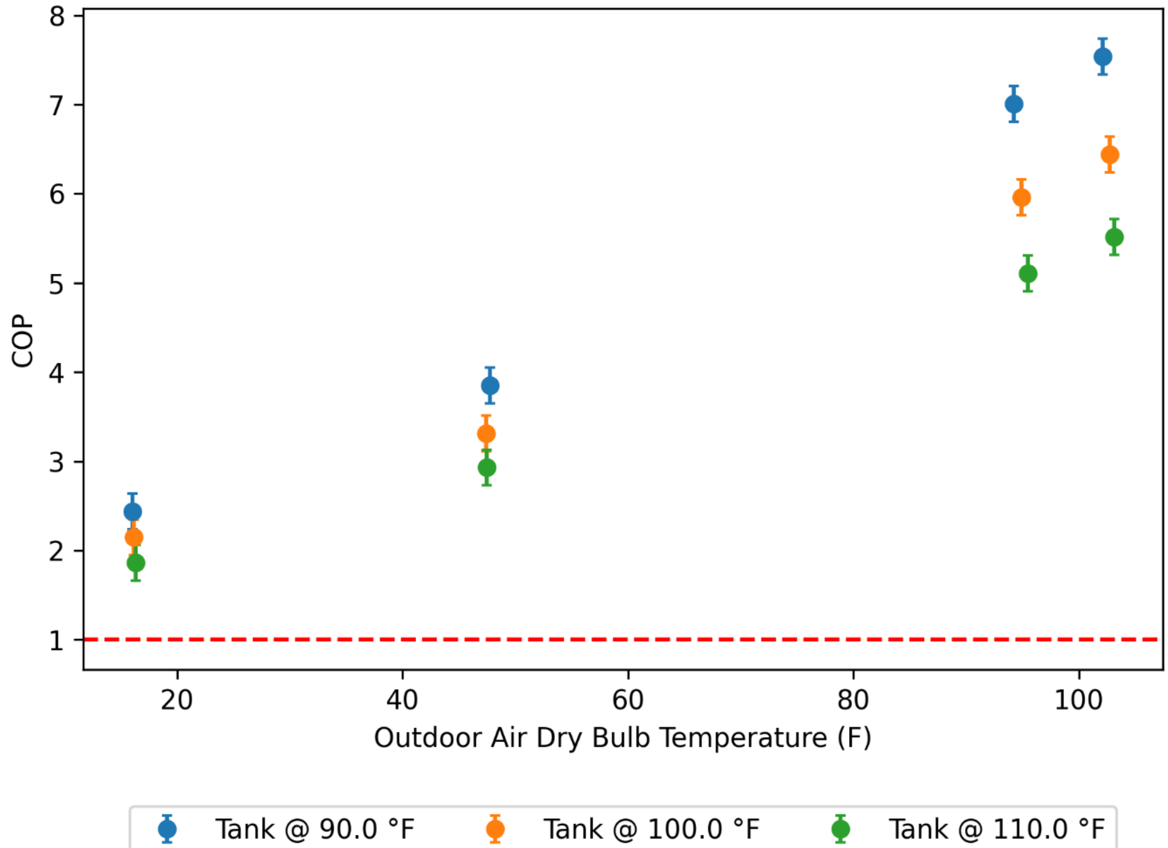

Figure 14: Measured variable-speed MFHP WH COP at different OA conditions and WT temperature bins in the constant compressor speed segment of the WH cycles. *

*For comparison, the electric resistance heater COP of 1.0 is shown as a dashed red line.

The time to heat the WT from 65°F starting to 120°F average tank temperature is shorter at higher OA temperature conditions. See Table 14 and Figure 15 below for more detail.

Table 14: WT heating times from 65°F starting average tank temperature to 120°F

OA Conditions (°F db/wb)	WT Heating Time from 65°F to 120°F (minutes)
17/15	66
47/43	48

OA Conditions (°F db/wb)	WT Heating Time from 65°F to 120°F (minutes)
95/75	45
105/77	41

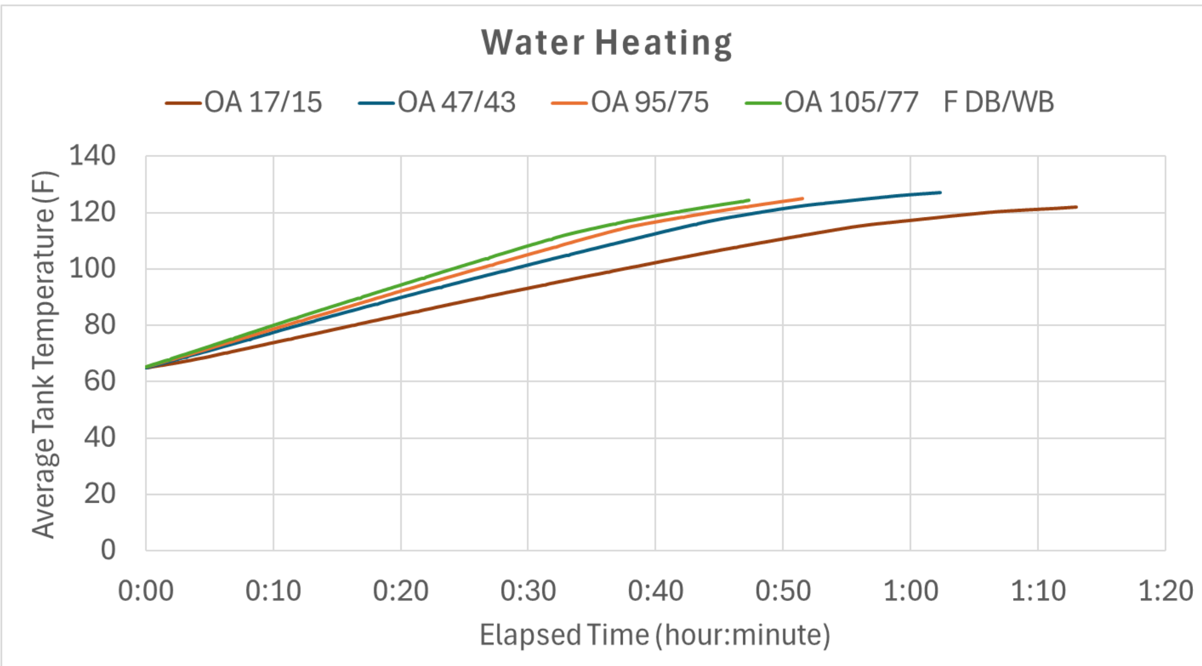

Figure 15: WH average tank temperature versus time for three different outdoor conditions, heating from 65°F.

Overnight tank heat loss tests measured the internal tank temperatures and the surrounding air temperatures. From these overnight tests, the uniform skin loss coefficient for the WT was measured as 0.81 W/m^2·K with a surface area of 2.36 m^2.

Heat Recovery—Simultaneous Space Cooling and Water Heating

Heat Recovery 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 Heat Recovery mode with high AHU supply air fan speed is shown in Figure 16 to compare with dedicated SC mode capacity. The Heat Recovery mode flows refrigerant from the AHU coil to the WT and does not flow refrigerant through the ODU coil, so it does not intentionally exchange heat with the outdoors. Heat Recovery mode was tested for OA 95°F db with three different IA conditions of db/wb 80/67, 78/64, and 75/60°F, with the space conditioning thermostat setpoint below but close to the IA return air temperature. An additional Heat Recovery test was run at OA 105°F db with IA db/wb 80/67, and thermostat setpoint far below the IA return air temperature where the manufacturer controls select a high compressor speed.

As seen in Figure 16, the Heat Recovery mode has more dependency on the WT temperature than on IA return air conditions. At OA temperature of 95°F db in Heat Recovery mode with space conditioning thermostat setpoint near the IA temperature, the manufacturer controls operate the compressor at intermediate speeds around 35 Hz, higher than the minimum speed, and deliver a larger space cooling capacity than the dedicated SC at minimum speed. The Heat Recovery mode average space cooling capacity across the whole Heat Recovery WT heating cycle from tank temperature 65 to 125°F was 5.67 kW, close to or slightly higher than the capacity predicted for dedicated SC mode at OA 95°F db, IA 80/67°F db/wb condition at the same compressor speed. The average WH capacity for these conditions was 6.16 kW, and the combined COP calculated from the sum of the SC and WH capacity divided by the electrical power consumed was 7.92.

For the high OA temperature of 105°F db in Heat Recovery mode test with the thermostat setpoint was set far below the IA temperature, the manufacturer controls operate the compressor at a speed higher than the rated capacity full speed, and deliver a larger space cooling capacity than the dedicated SC at rated full speed capacity for this OA condition.

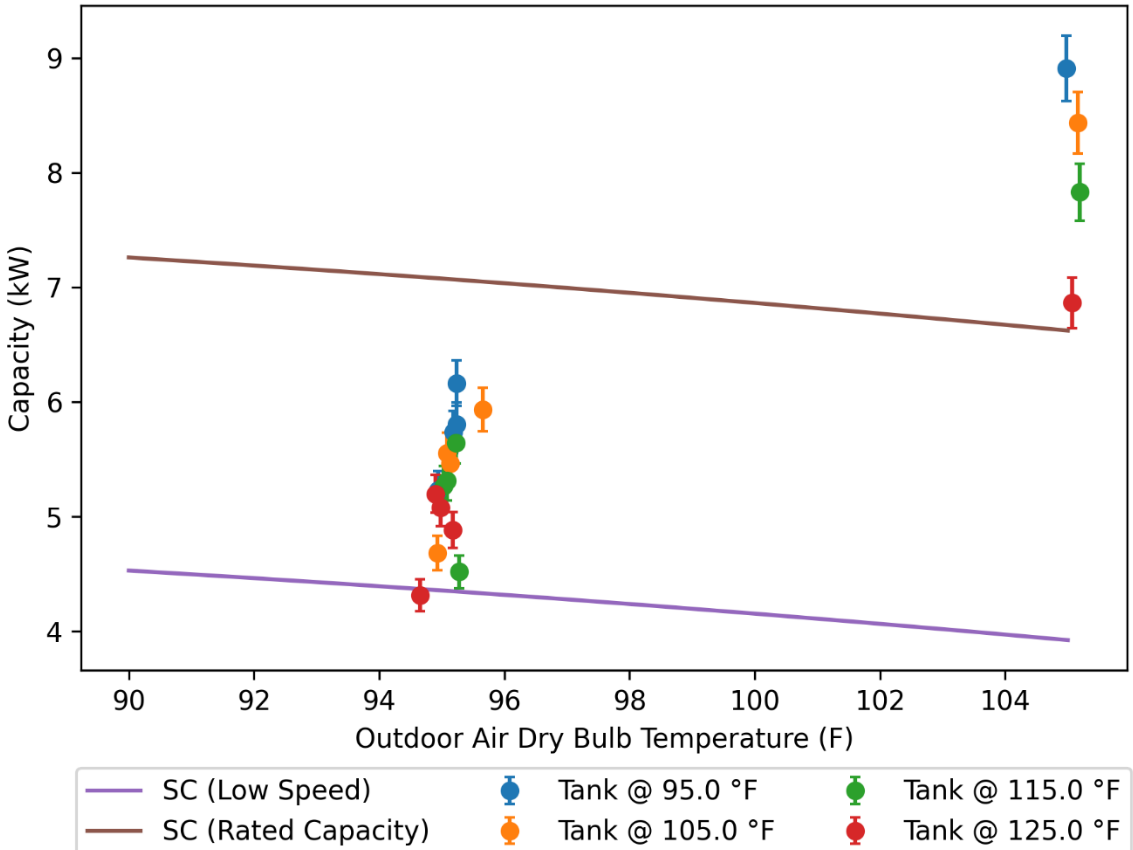

Figure 16: Measured MFHP Heat Recovery mode SC capacity (circle markers) at different OA db temperatures with regular SC mode capacities for comparison (lines).

Water heating capacities of the Heat Recovery mode, at 35Hz compressor speed OA 95°F db, are significantly lower than the dedicated WH mode at 45Hz compressor speed, Figure 17. Water heating capacities of the Heat Recovery mode, at 71Hz compressor speed OA 105°F db, are closer to but still less than dedicated WH mode at 45Hz compressor speed, as shown in Figure 17.

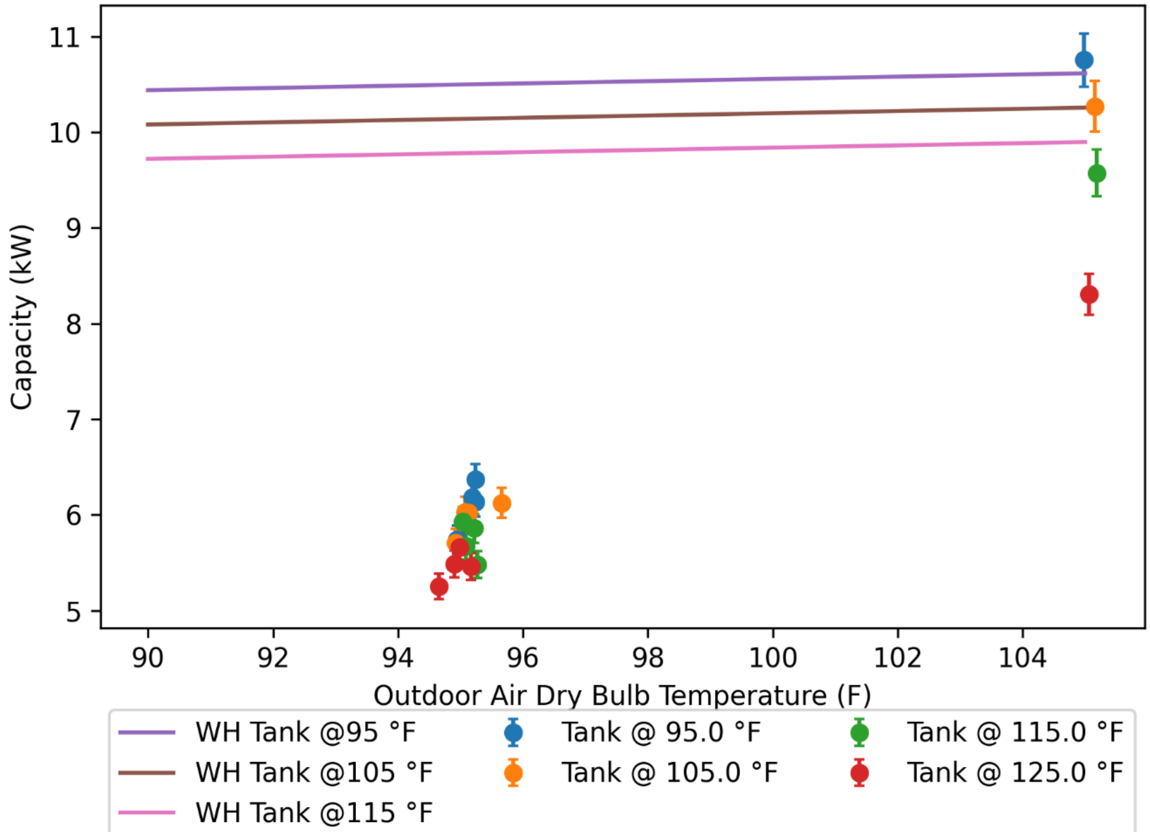

Figure 17: Measured MFHP Heat Recovery Mode water heating capacity with IA db/wb 80/67°F at 35Hz compressor speed for OA 95°F and 71Hz for OA 105°F (circle markers), compared to WH capacity at 45Hz compressor speed across a range of OA db and tank temperature bins (lines).

To fairly compare the expected power consumption of two separate dedicated SC and WH cycles, the team used the COPs of the dedicated SC mode at compressor speeds that would provide the measured SC capacity of the Heat Recovery mode, assuming the standard OA db/wb 95/75°F and IA condition of db/wb 80/67°F. This analysis is performed for Heat Recovery at 35Hz compressor speed in the four WT temperature bins and shown in four plots in Figure 18 and Figure 19. For dedicated SC mode, the test data across multiple compressor speeds enables accurate estimation. For dedicated WH mode, test data measured performance at the manufacturer-controls-selected compressor speeds of 45Hz for tank temperature bins 95 to 115°F, and 30Hz for the 125°F bin. At all conditions, Heat Recovery mode was found to use less power and have a higher COP than providing the same services with separate SC and WH cycles: See blue versus red bars in Figure 18 and Figure 19. For heating from a tank temperature 110 to 120°F, the Heat Recovery mode was found to consume 67 percent of the electrical power compared to two separate WH and SC modes. This corresponds to an average 33 percent energy savings for the Heat Recovery cycle compared to separate SC and WH mode cycles at this OA 95 and IA condition of db/wb 80/67.

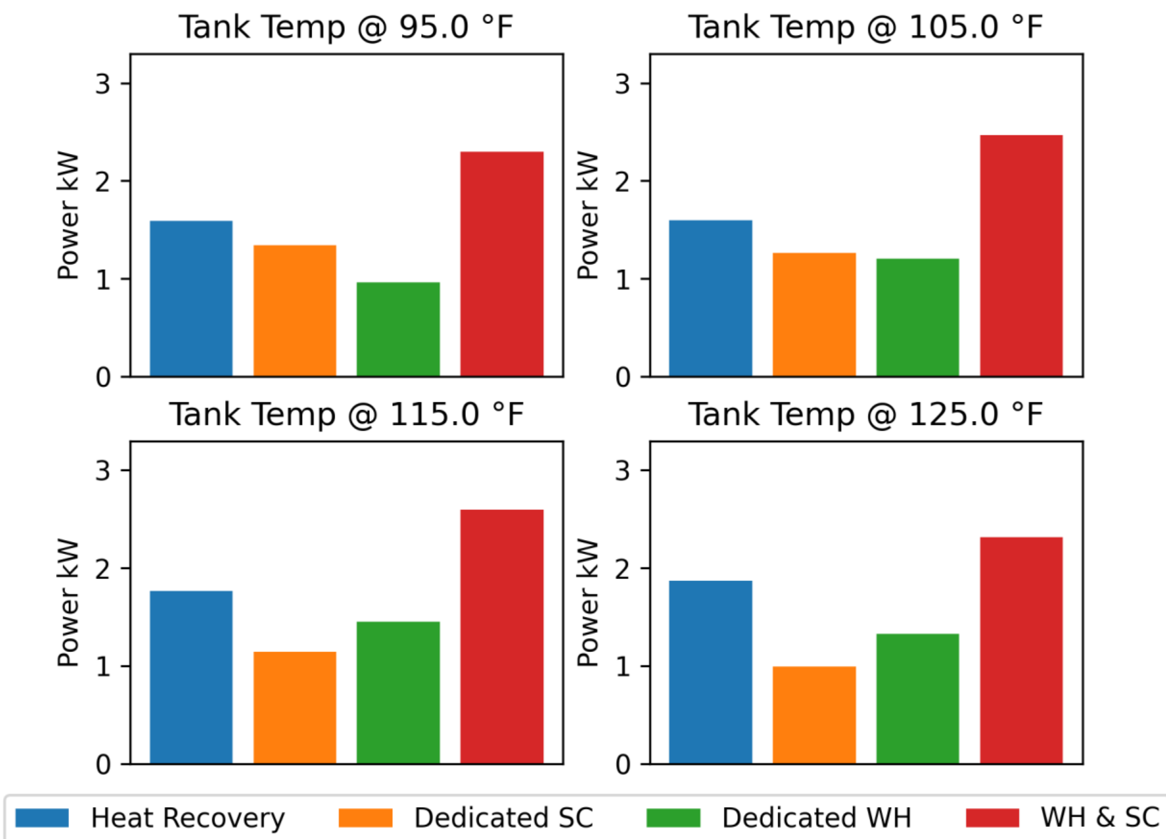

Figure 18: Power consumption of Heat Recovery mode with IA db/wb 80/67°F (blue), and equivalent power consumption estimate for separate SC and WH cycles delivering the same capacity at OA db/wb 95/75°F (orange and green), sum of power for SC and WH separate cycles (red).

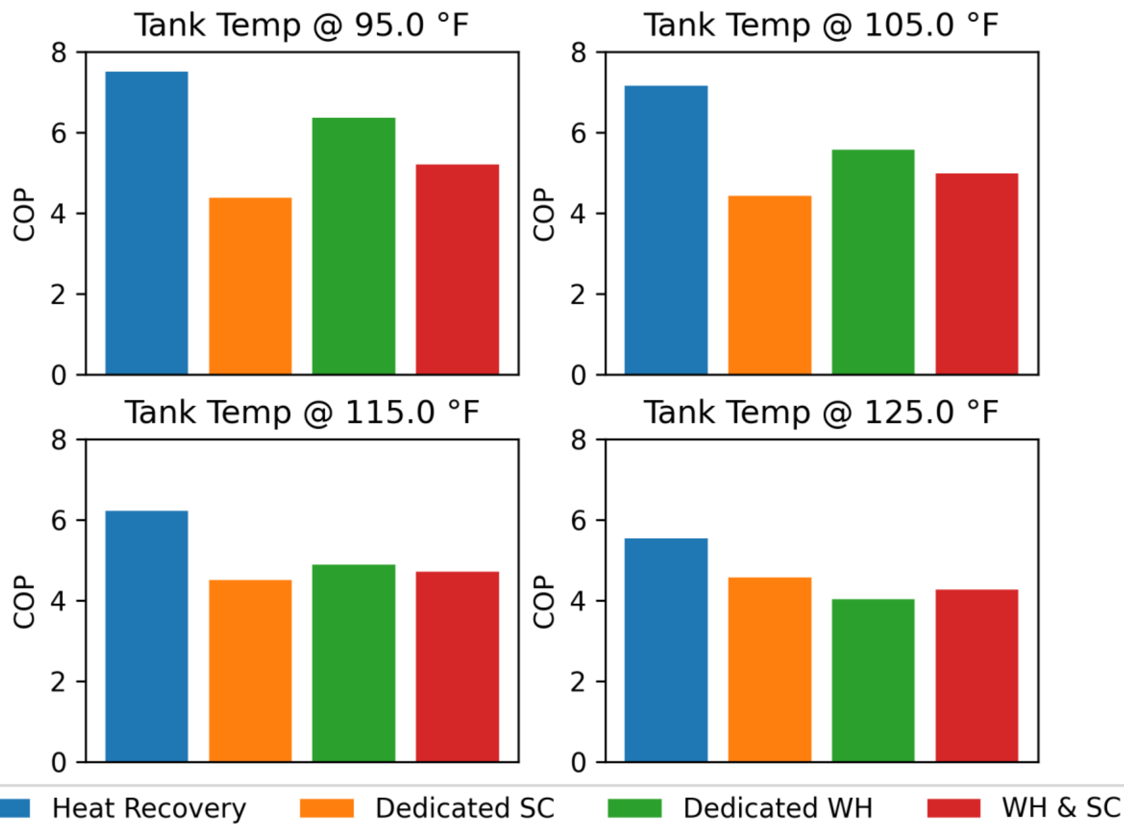

Figure 19: COP of Heat Recovery mode with IA db/wb 80/67°F (blue), and equivalent COP estimate for separate SC and WH cycles delivering the same capacity at OA db/wb 95°F/75°F (orange and green), sum of capacity divided by sum of estimated power consumption COP for separate SC and WH cycles (red).

Bi-Heating – Simultaneous Space Heating and Water Heating

In Bi-Heating mode, the variable-speed MFHP moves thermal energy from outside air to simultaneously heat both the space and water tank. The space conditioning thermostat temperatures were set close to the IA temperatures as would be expected in installations with correctly sized equipment without setbacks. In this mode the manufacturer controls select compressor speeds higher than for either the separate SH or WH modes at the same conditions. At OA db/wb 47/43°F and IA db 70°F, heating the water tank from an average temperature of 80 up to 120°F, the variable-speed MFHP Bi-Heating mode with compressor speed of 78Hz provided a SH capacity of 7.0 kW (23.9 k Btu/h, 2.0 ton) and average WH capacity of 4.5 kW for a combined COP of 3.36.

The estimates for how much power a separate WH cycle would have used to heat water at the same rate as the Bi-Heating mode were based on the WH operation COP at 65Hz for each tank temperature bin. The dedicated WH capacity at 65Hz was significantly higher than in Bi-Heating mode, so these are likely overestimates of the power consumption for dedicated water heating if that mode could have been operated at lower compressor speed. There was not a significant difference between power consumption or COP for Bi-Heating compared to the estimates for separate modes delivering the same SH and WH capacity, Figure 20 and Figure 21.

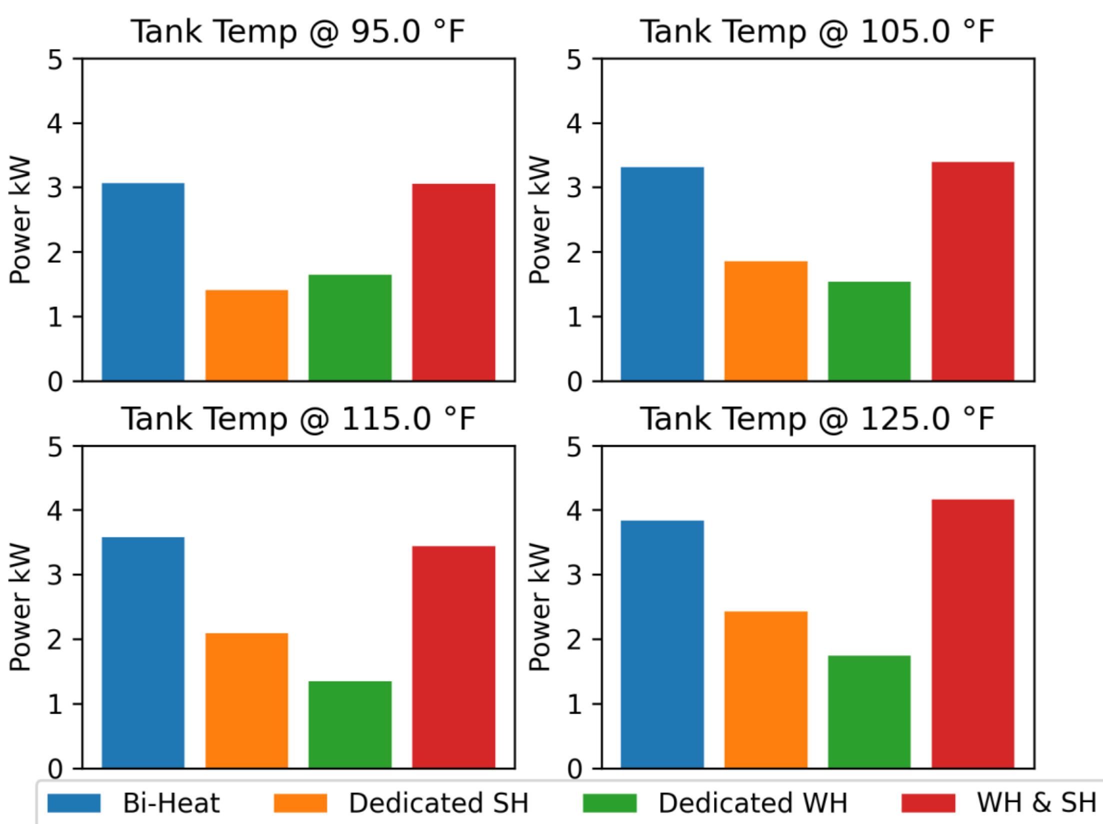

Figure 20: Power consumption of Bi-Heating mode with IA db 70°F (blue), and equivalent power consumption estimate for separate SH and WH cycles delivering the same capacity at OA db/wb 47/43°F (orange and green), sum of power for SH and WH separate cycles (red).

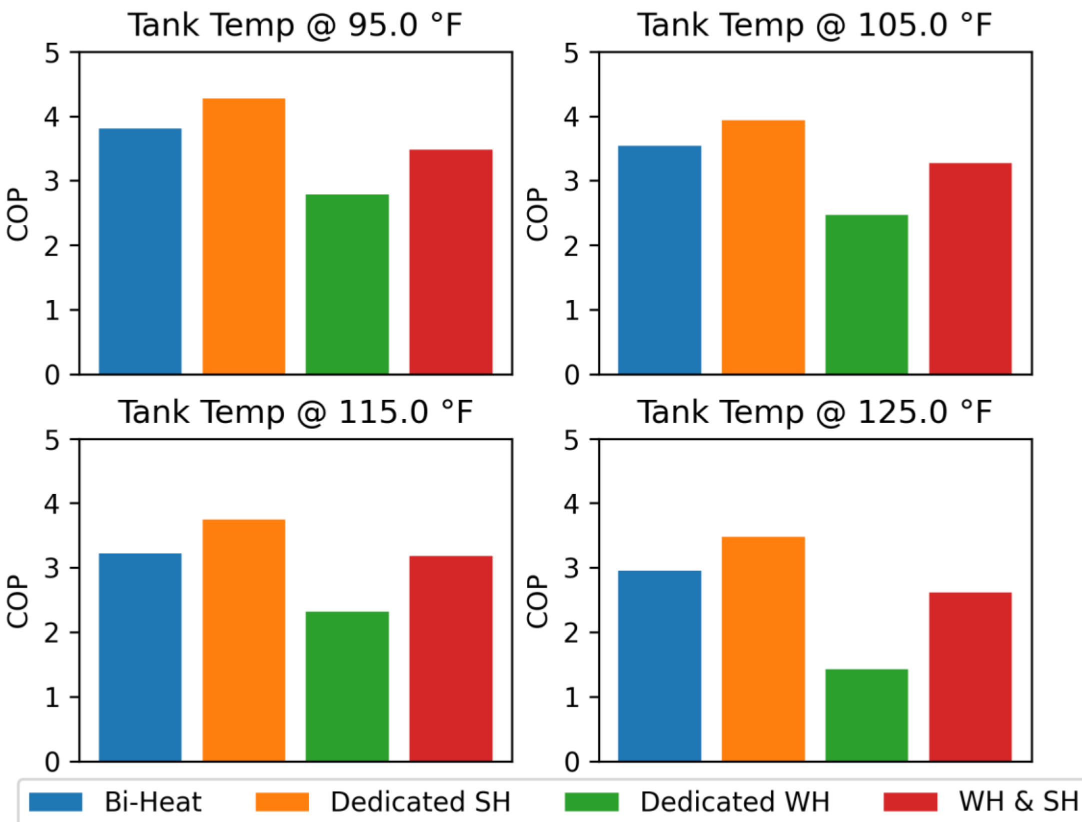

Figure 21: COP of Bi-Heating mode with IA db 70°F (blue), and equivalent COP estimate for separate SH and WH cycles delivering the same capacity at OA db/wb 47/43°F (orange and green), sum of capacity divided by sum of power consumption COP for combined SH and WH separate cycles (red).

Defrost (DEF)

To melt accumulated frost on the outdoor coil during an SH, WH, or Bi-Heating cycle, the variable-speed MFHP defrost modes reverse the refrigerant flow to transfer heat from the WT to the outdoor coil. DEF mode was initiated by the manufacturer controls.

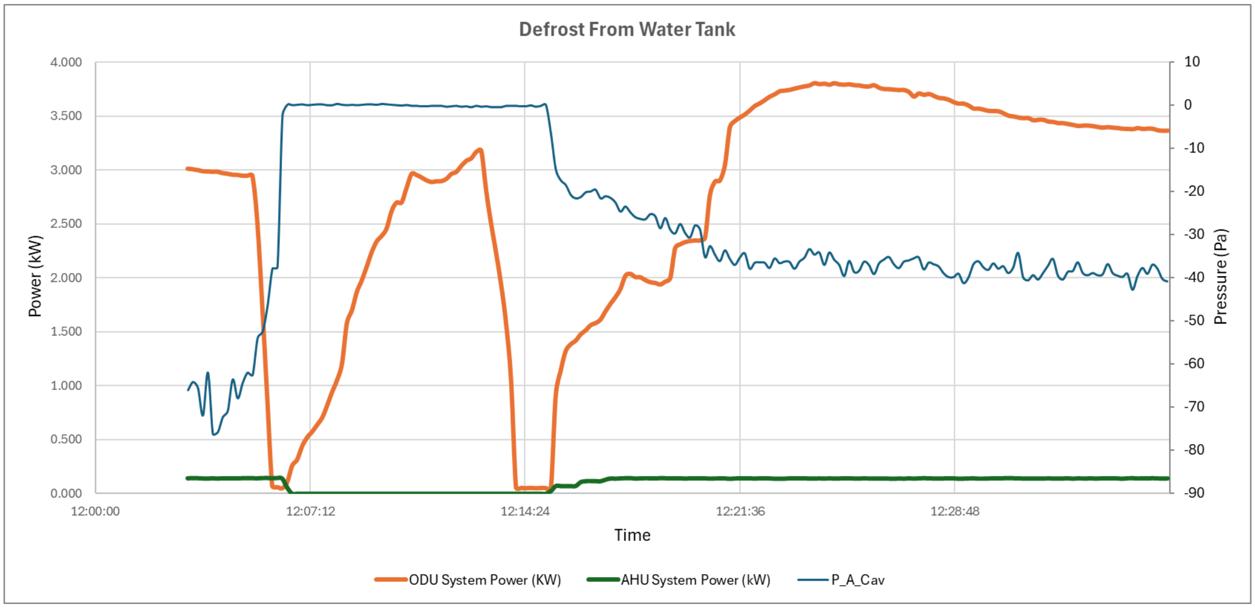

Figure 22: Defrost cycle time series with power consumption for the outdoor unit and air handler unit and the outdoor unit internal pressure.

Figure 22 shows the DEF mode triggered during SH mode when the AHU power dropped to zero and the ODU system power dropped from 2.95 kW to 0.06 kW. The DEF cycle occurs when the ODU power is non-zero, but the AHU power is zero, indicating the AHU fan is turned off. The DEF mode lasted for approximately 590 seconds with an average power of 1.85 kW, over this time the ODU power rose from 0.06 kW to a maximum of 3.18 kW as the compressor speed ramped up, the WT temperature dropped, and the outdoor coil temperature rose. During this DEF cycle the WT average temperature dropped from 116 to 106°F, a 10-degree drop, while the temperature at the top of the tank dropped by less than 0.5°F, remaining above 119°F. With the top tank temperature remaining above typical thermostatic mixing valve outlet temperatures, the domestic hot water end users would likely not experience any difference in hot water delivery.

The variable-speed MFHP DEF mode power draw varied from 0.06 kW to 3.18 kW, in a similar range as the previously tested single-speed MFHP between 1 and 3 kW. In comparison, a traditional four-ton rated capacity single-speed HP tested in the WCEC environmental chambers drew 5 to 9 kW including the electric resistance heaters in the AHU. This means that the total power consumption for DEF mode is significantly lower for variable- and single-speed MFHP than for typical split-system single-speed heat pumps and would result in energy savings. The variable- and single-speed MFHP reduced peak power draw have the potential to avoid some retrofit electrical panel upgrades.

Water Heating First-Hour Rating Test

The lab test estimated FHR was 86.4 gallons for the variable-speed 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 measuring the total volume supplied by the WT during each draw. Figure 23 below shows the cumulative volume of water drawn from the tank over the course of the FHR test.

Due to limits in valve actuation speed, there was a small 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. This resulted in a transitional period of a few seconds in the data, during which time the test parameters for the MFHP FHR test deviated slightly from the conditions outlined in the test standard. To correct for this minor deviation, only the water draw volumes within the bounds required for the test (3.0±0.25 GPM) were counted in calculating the FHR for the variable-speed MFHP.

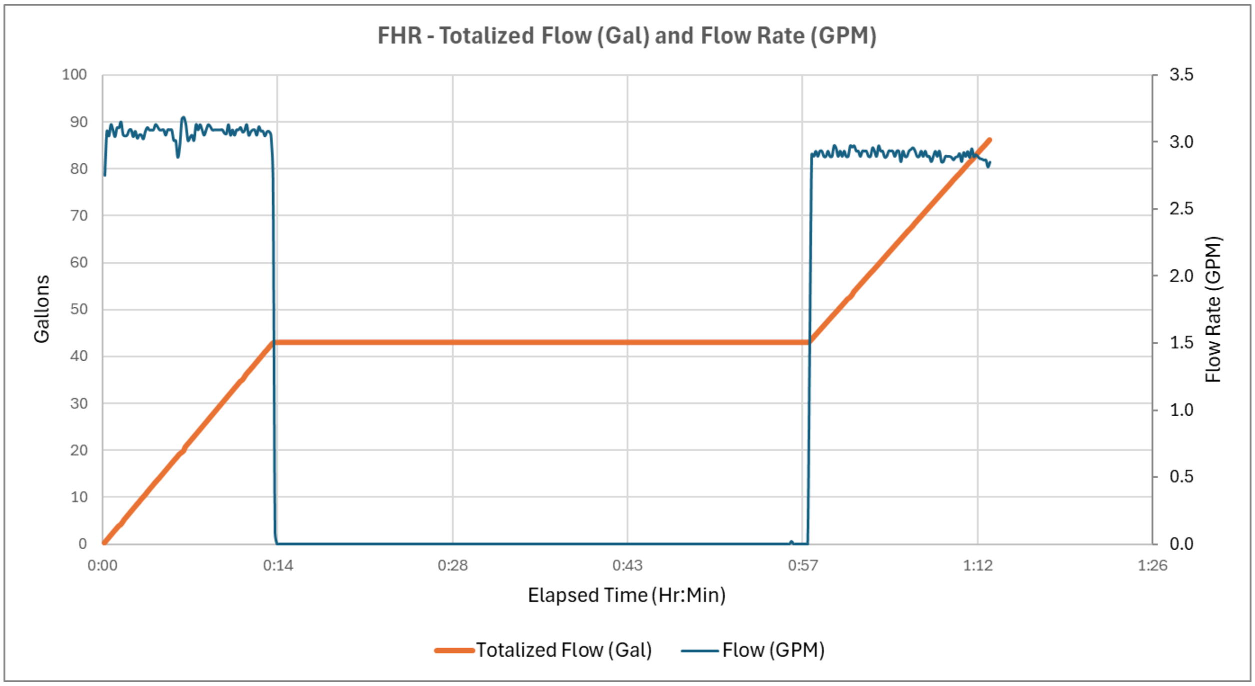

Figure 23: FHR test time series showing water flow rate and cumulative output water volume.

Equipment Performance Curves for EnergyPlus

The lab test data were used to generate regression curves to estimate the variable-speed MFHP performance in EnergyPlus or CBECC-Res. These curves are generated by first fitting lab data results to the test range of input variables using least squares regression techniques, F-statistic, and standard error. Engineering judgement was exercised to ensure the predicted value behavior is physically reasonable. Regressions quality statistics and residuals plots were inspected to verify regression accuracy and avoid runaway behavior outside the measured data ranges (Appendix B).

The data were 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 below, lab data points have been used to generate the expected performance trend for the Space Cooling, $\dot{H}_{SC}$ at compressor speed 44 Hz, 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 and meet the desired input requirements of EnergyPlus.

$$\overline{\boldsymbol{C}}_{\boldsymbol{SC,H}} = \begin{bmatrix} -6.300\mathrm{E}-04 \\ 7.477\mathrm{E}-03 \\ -2.333\mathrm{E}-03 \\ 2.366\mathrm{E}-01 \\ -5.838\mathrm{E}-01 \\ 1.079\mathrm{E}+01 \end{bmatrix}$$

$$\dot{H}_{SC,rated} = \dot{H}_{SC} \ @\begin{matrix} T_{OA,DB}: 35.00°\mathrm{C} \\ T_{RA,WB}: 17.22°\mathrm{C} \end{matrix} = \boldsymbol{7.178kW}$$

Using this to normalize 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:

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

$$\overline{\boldsymbol{C}} = \overline{\boldsymbol{C}}_{\boldsymbol{SC,H}} / \dot{H}_{SC,rated} = \begin{bmatrix} \mathbf{-8.800E-05} \\ \mathbf{1.041E-03} \\ \mathbf{-3.250E-04} \\ \mathbf{3.296E-02} \\ \mathbf{-8.132E-02} \\ \mathbf{1.503E+00} \end{bmatrix}$$

EnergyPlus, with the entered values of $\dot{H}_{SC,rated}$ and $\overline{\boldsymbol{C}}$, can model Space Cooling mode operation at a compressor speed of 44 Hz, calculating capacity of the unit given various $T_{OA,DB}$ and $T_{RA,WB}$ conditions. With this regression and the other regressions outlined in Appendix B, EnergyPlus can model electrical power consumption, supply air temperature, and supply air humidity to predict energy savings compared to a variety of baselines.

Stakeholder Feedback

The UC Davis team is engaging with stakeholders on various issues, including the variable-speed MFHP equipment manufacturer; the teams likely to perform next step field testing and measure development for the technology, including WCEC; CalNEXT team members; IOU staff; and CEC developers for CBECC-Res. The UC Davis WCEC team will continue close collaboration with the equipment manufacturer, engineering, and leadership personnel. Additionally, we will communicate with the CEC CBECC-Res developer team to make sure that the performance curves we develop are compatible with the code compliance software, as well as to request inclusion of final performance curves in the publicly available CBECC-Res. The WCEC team will solicit feedback from IOUs and from CalNEXT team members to maximize the impact of this project.

The Pacific Gas & Electric (PG&E) Codes and Standards team reached out to us for guidance on MFHPs and their future impacts on codes and efficiency programs. In an online meeting in March of 2025, we advised them on the status of different types of MFHP products, and they requested updates so that our studies can inform them of the potential for air-to-air and air-to-water MFHP products.

The UC Davis team has met with the CEC developers for CBECC-Res twice—on February 11, 2025, and August 19, 2025—to explore the requirements for CBECC-Res to model MFHP equipment for code compliance and building design efficiency credit. The CBEC-Res team is confident that they can use the results from the UC Davis work to enable MFHP simulations.

Recommendations and Technology Transfer Next Steps

The results of the lab tests for the variable-speed air-to-air MFHP show the potential for significant energy savings compared to the single speed air-to-air MFHP previously tested. This equipment has the potential to increase energy savings and can help to avoid some panel upgrades to reduce retrofit electrification costs compared to the typical single speed split system heat pumps and standalone heat pump water heaters. We recommend that MFHP technologies move forward to lab and field test additional commercially available products, and to develop new efficiency measures for utility energy efficiency and electrification programs.

The project team's next steps include:

- Continue to engage with SDG&E, the California HVAC program administrator, to determine what standards and requirements the equipment needs to meet to be included as a new measure in the efficiency program. Follow up with the MFHP manufacturer to support meeting those requirements.
- Continue to seek funding by submitting relevant proposals and engaging with stakeholders, as well as preparing 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 the CalTF throughout the measure development process
- 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.

A new CalNEXT funded research home investigation compares variable-speed air-to-air MFHP performance to the typical separate single-speed space conditioning and water heating HPs with repeatable setpoints and hot water draws (ET24SWE0058). Future laboratory tests of one or more competing variable-speed air-to-air MFHP products will be proposed to evaluate the performance of other variable-speed MFHP equipment designs.

We are also planning future follow-on field demonstration(s) to evaluate typical real-world energy performance in occupied building(s), 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 MFHPs in a real building with repeatable setpoints and hot water draws. Each of these steps will include products that are a good fit for older residential buildings, which are common for 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 water heating HPs, because MFHPs enable reductions in retrofit costs, along with high-efficiency operation and utilization of heat for hot water heating. MFHP measures can use many of the existing HP program design elements.

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Appendix A: EnergyPlus Simulation Inputs and Calculation Details

Cooling Coil

The rated conditions for obtaining the capacities, COPs, and Sensible Heat Ratios (SHRs) are indoor dry-bulb temperature at 26.67ºC (80°F), wet-bulb temperature at 19.44°C (67°F), and the source side entering air temperature at 35°C (95°F). The equipment is operated at full compressor speed with the manufacturer-selected supply air flow rate to determine the rated total cooling capacity, rated SHR, rated cooling COP, and rated air flow rate. At this same condition and air flow rate, the equipment is then tested across a range of compressor speeds

Performance Curves

1) **Total cooling capacity modifier curve (function of temperature):**

The total cooling capacity modifier as a function of temperature curve (CAP-FT) is a biquadratic curve with two independent variables: wet-bulb temperature of the air entering the cooling coil and the dry-bulb temperature of the air entering the condenser. The output of this curve is multiplied by the rated total cooling capacity at the speed, to give the total cooling capacity at the specific evaporator inlet air wet-bulb temperature and condenser inlet air dry-bulb temperature at which the direct expansion (DX) unit is operating (i.e., at temperatures different from the rating point temperatures). Note: The data used to develop the total cooling capacity modifier curve (function of temperature) should represent performance when the cooling coil is “wet” (i.e., coil providing sensible cooling and at least some dehumidification). Performance data when the cooling coil is “dry” (i.e., not providing any dehumidification) should not be included when developing this modifier curve. This model automatically detects and adjusts for “dry coil” conditions.

$$TotCapTempModFac = a + b \cdot WB_i + c \cdot {WB_i}^2 + d \cdot DB_o + e \cdot {DB_o}^2 + f \cdot WB_i \cdot DB_o$$

Where:

WB_i is the wet-bulb temperature of the air entering the indoor cooling coil (°C)

DB_o is the dry-bulb temperature of the air entering the condenser coil (°C)

$a \dots f$ are the regression curve-fit coefficients.

2) **Total cooling capacity modifier curve (function of air flow fraction):**

The total cooling capacity modifier curve (function of air flow fraction) is a cubic curve, with the independent variable being the ratio of the actual air flow rate across the cooling coil to the design air flow rate (i.e., fraction of design flow at the speed):

$$TotCapAirFlowModFac = a + b \cdot ff_a + c \cdot {ff_a}^2 + d \cdot {ff_a}^3$$

Where:

ff_a is the actual air mass flow rate/design air mass flow rate, at one speed level

$a \dots d$ are the regression curve fit coefficients. If no data for correction is available, the user can simply use a = 1.0 and set the other coefficients to 0.0.

3) **EIR modifier curve (function of temperature):**

The EIR modifier curve as a function of temperature (EIR-FT) is a biquadratic curve with two independent variables: wet-bulb temperature of the air entering the cooling coil and dry-bulb temperature of the air entering the condenser. The output of this curve is multiplied by the rated EIR (inverse of the rated COP) at the speed level, to give the EIR at the specific inlet air temperatures at which the DX unit is operating (i.e., at temperatures different from the rating point temperatures).

$$EIRTempModFac = a + b \cdot WB_i + c \cdot {WB_i}^2 + d \cdot DB_o + e \cdot {DB_o}^2 + f \cdot WB_i \cdot DB_o$$

Where:

$a \dots f$ are regression curve fit coefficients.

4) EIR modifier curve (function of air flow fraction):

$$EIRAirFlowModFac = a + b \cdot ff_a + c \cdot {ff_a}^2 + d \cdot {ff_a}^3$$

Where:

$a \dots d$ are regression curve fit coefficients. If no data is available for correction, the user can simply use a = 1.0 and set the other coefficients to 0.0.

Performance Calculations

BYPASS FACTOR

Page 851 and 894 of the E+ Engineering Reference: The rated total capacity and rated SHR are first used to determine the ratio of change in air humidity ratio to air dry-bulb temperature:

$$SlopeRated = \frac{\omega_{in} - \omega_{out}}{T_{db,in} - T_{db,out}}$$

(15.133)

where:

ω_{in} is the humidity ratio of the air entering the cooling coil at rated conditions (kg/kg)

ω_{out} is the humidity ratio of the air leaving the cooling coil at rated conditions (kg/kg)

$T_{db,in}$ is the dry-bulb temperature of the air entering the cooling coil at rated conditions (°C)

$T_{db,out}$ is the dry-bulb temperature of the air leaving the cooling coil at rated conditions (°C).

Along with the rated entering air conditions, the algorithm then searches along the saturation curve of the psychrometric chart until the slope of the line between the point on the saturation curve and the inlet air conditions matches *SlopeRated*. Once this point, the apparatus dewpoint is found on the saturation curve and the coil bypass factor at rated conditions is calculated.

The bypass Factor is only used when the coil is wet. In the dry coil condition, the SHR is equal to 1.0. The Bypass Factor is also only used if SHR Modifier Curves are not utilized. For variable speed the following calculation is made on the lowest speed setting.

$$BF_{rated} = \frac{h_{out,rated} - h_{ADP}}{h_{in,rated} - h_{ADP}}$$

where:

$h_{out,rated}$ is the enthalpy of the air leaving the cooling coil at rated conditions (J/kg)

$h_{in,rated}$ is the enthalpy of the air entering the cooling coil at rated conditions (J/kg)

h_{ADP} is the enthalpy of saturated air at the coil apparatus dewpoint (J/kg).

The coil bypass factor is analogous to the "ineffectiveness" (1-ε) of a heat exchanger and can be described in terms of the number of transfer units (NTU).

$$BF = e^{-NTU} = e^{-\frac{\left(\frac{UA}{cp}\right)}{\dot{m}}} = e^{-\frac{A_o}{\dot{m}}}$$

For a given coil geometry, the bypass factor (BF) is only a function of air mass flow rate. The model calculates the parameter A_o based on BF_{rated} and the rated air mass flow rate. With A_o known, the coil BF can be determined for non-rated air flow rates.

For each simulation time step when the DX air conditioner operates to meet a cooling load, the total cooling capacity at the actual operating conditions is calculated using Equation 15.127, and the coil bypass factor is calculated based on Equation 15.135. The coil bypass factor is used to calculate the operating sensible heat ratio (SHR) of the cooling coil using Equations 15.136 and 15.137.

$$h_{ADP} = h_{in} - \frac{\dot{Q}_{total}/\dot{m}}{1 - BF}$$

$$SHR = Min\left(\left(\frac{h_{Tin,wADP} - h_{ADP}}{h_{in} - h_{ADP}}\right), 1\right)$$

where:

h_{in} is the enthalpy of the air entering the cooling coil (J/kg)

h_{ADP} is the enthalpy of air at the apparatus dewpoint condition (J/kg)

$h_{Tin,wADP}$ is the enthalpy of air at the entering coil dry-bulb temperature and humidity ratio at ADP (J/kg)

$\dot{m}$ is the air mass flow rate (kg/s).

With the SHR for the coil at the current operating conditions, the properties of the air leaving the cooling coil are calculated using the following equations:

$$h_{out} = h_{in} - \frac{\dot{Q}_{total} \cdot PLR}{\dot{m}}$$

$$h_{Tin,wout} = h_{in} - (1 - SHR)(h_{in} - h_{out})$$

where:

h_{out} is the enthalpy of the air leaving the cooling coil (J/kg)

$h_{Tin,wout}$ is the enthalpy of air at the entering coil dry-bulb temperature and leaving air humidity ratio (J/kg)

For higher speed operation, the BF factor is calculated as a relation to the speed ratio.

LOWEST SPEED OPERATION

$$GrossCoolingCapacity_{1,cond} = RatedTotalCoolingCapacity_1 \cdot TotCapTempModFac \cdot TotCapAirFlowModFac_1$$

$$EIR_{1,cond} = \frac{1.0}{COP_{referance\ 1}} EIRTempModFac_1 \cdot EIRAirFlowModFac_1$$

Power excluding the fan:

$$Power_1 = GrossCoolingCapacity_{1,cond} \cdot EIR_{1,cond} \cdot RTF$$

Where:

RTF is the Runtime Fraction

RATED SENSIBLE HEAT RATIO

At the lowest speed, the dehumidification calculation is the same as the single-speed DX coil, and uses the rated SHR and the design air flow rate at the lowest speed to calculate rated bypass factor of $BF_{rated,1}$, and the corresponding effective surface area of $A_{o,1}$. With $A_{o,1}$ known, the coil BF can be adjusted for non-rated air flow rates.

The part load ratio for sensible cooling is:

$$PLR = \frac{SensibleCoolingLoad}{GrossCoolingCapacity_{1,cond} \cdot SHR_1}$$

For latent cooling:

$$PLR = \frac{LatentCoolingLoad}{GrossCoolingCapacity_{1,cond} \cdot (1.0 - SHR_1)}$$

HIGHER SPEED OPERATION

As shown on page 894 of the E+ Engineering Reference, at the speed level between the lowest and the highest, there is no part-load loss, meaning there is no Runtime Fraction. A parameter of speed ratio (SpeedRatio) is used to define the capacity partition between Speed x-1 and Speed x.

The design air flow rate at the speed ratio is given as follows:

Design Air Flow Rate Speed Ratio

$$\begin{aligned} DesignAirFlowRateSpeedRatio &= ReferenceUnitAirMassFlowRate@SpeedLevel(x-1) \cdot CapacityScaleFactor \\ &\cdot CapacityScaleFactor \cdot (1 - SpeedRatio) \\ &+ ReferenceUnitAirMassFlowRate@SpeedLevel(x) \ \cdot CapacityScaleFactor \\ &\cdot SpeedRatio \end{aligned}$$

Airflow Fraction

$$ff_{a,x-1} = ff_{a,x} = \frac{actualairmassflowrate}{DesignAirFlowRateSpeedRatio}$$

Cooling Capacity

$$\begin{aligned} Q_{total,x-1} = RatedTotalCoolingCapacity@SpeedLevel(x-1) \cdot TotCapempModFac_{x-1} \\ \cdot TotCapAirFlowModFac_x \end{aligned}$$

$$Q_{total,x} = RatedTotalCoolingCapacity@SpeedLevel(x) \cdot TotCapTempModFac_x \cdot TotCapAirFlowModFac_x$$

EIR

$$EIR_{x-1} = \frac{1.0}{ReferenceUnitCOPSpeed(x-1)} EIRTempModFac_{x-1} \cdot EIRAirFlowModFac_{x-1}$$

$$EIR_x = \frac{1.0}{ReferenceUnitCOPSpeed(x)} EIRTempModFac_x \cdot EIRAirFlowModFac_x$$

Total Cooling Capacity

$$Q_{total,SpeedRatio} = (1.0 - SpeedRatio) * Q_{total,x-1} + SpeedRatio \cdot Q_{total,x}$$

Total Power

$$Power_{speedRatio} = (1.0 - SpeedRatio) \cdot Q_{tot,x-1} \cdot EIR_{x-1} + SpeedRatio \cdot Q_{total,x} \cdot EIR_x$$

Net Heat Discharged from the Condenser

$$Q_{cond,SpeedRatio} = Power_{SpeedRatio} + Q_{total,SpeedRatio}$$

Effective Surface Area

$$A_{o,SpeedRatio} = (1.0 - SpeedRatio) \cdot A_{0,x-1} + SpeedRatio \cdot A_{o,x}$$

Heating Coil

The rated conditions for obtaining the capacities and COPs are at indoor dry-bulb temperature of 21.1°C (70°F) and the source-side entering air temperature of 8.3°C (47°F).

Performance Factor Curves

Total Heat Capacity Modifier Curve $f(DB_i, DB_o)$

$$TotCapTempModFrac = a + b \cdot dB_i + c \cdot {DB_I}^2 + d \cdot {DB_o}^2 + f \cdot DB_i \cdot DB_o$$

Total Heat Capacity Modifier Curve $f(ff_a)$

$$TotCapAirFlowModFrac = a + b \cdot ff_a + c \cdot {ff_a}^2 + d \cdot {ff_a}^3$$

Energy Input Ratio (EIR) Modifier Curve $f(DB_i, DB_o)$

$$EIRTempModFunc = a + b \cdot DB_i + c \cdot {DB_i}^2 + d \cdot DB_o + e \cdot {DB_o}^2 + f \cdot DB_i \cdot DB_o$$

Energy Input Ratio (EIR) Modifier Curve $f(ff_a)$

$$EIRAirFlowModFunc = a + b \cdot ff_a + c \cdot {ff_a}^2 + d \cdot {ff_a}^3$$

Water Heating Coil

"Coil:WaterHeating:AirToWaterHeatPump:VariableSpeed"

Rating conditions:

- Rated Water Heating Capacity
- Rated Evaporator Inlet Air Dry-Bulb Temperature
- Rated Evaporator Inlet Air Wet-Bulb Temperature
- Rated Condenser Inlet Water Temperature
- Rated Condenser Water Flow Rate

For each speed at rated conditions:

- Reference Unit Rated Water Heating Capacity
- Reference Unit Rated Heating COP
- Rated SHR
- Reference Unit Rated Air Flow Rate
- Reference Unit Rated Water Flow Rate
- Reference Unit Rated Water Pump Input Power

Curves

Total WH Capacity Function of Temperature Curve

$$WHCapacityFuncTemp = a + b \cdot T_e + c \cdot {T_e}^2 + d \cdot EWT + e \cdot EWT^2 + f \cdot T_e \cdot EWT$$

Where:

T_e is either the evaporator wet-bulb or dry-bulb temperature

EWT is the entering water temperature of the condenser

COP Function of Temperature Curve

$$WHCOPFuncTemp = a + b \cdot T_e + c \cdot {T_e}^2 + d \cdot EWT + e \cdot EWT^2 + f \cdot T_e \cdot EWT$$

Where:

T_e is either the evaporator wet-bulb or dry-bulb temperature

EWT is the entering water temperature of the condenser

Water flow fraction: Set to 1 because it is manufacturer controlled and varies over a tank heating cycle

Outdoor coil air flow fraction: Set to 1 because it is manufacturer controlled and varies over a tank heating cycle

Select the options to set the water and air flow fraction factors, as included in the capacity and COP curves in EnergyPlus. Additionally, capacity and COP are functions of the temperature curve at each speed.

Sim SCWH Coil

Simultaneous space cooling and domestic hot water heating with full refrigerant condensation in the refrigerant to water heat exchanger. It is similar to Water Heating but uses the indoor temperature instead of the outdoor coil temperatures.

"Coil:WaterHeating:AirToWaterHeatPump:VariableSpeed"

Rating conditions:

- Rated Water Heating Capacity
- Rated Evaporator Inlet Air Dry-Bulb Temperature
- Rated Evaporator Inlet Air Wet-Bulb Temperature
- Rated Condenser Inlet Water Temperature
- Rated Condenser Water Flow Rate

For each speed at rated conditions:

- Reference Unit Rated Water Heating Capacity
- Reference Unit Rated Heating COP
- Rated SHR
- Reference Unit Rated Air Flow Rate
- Reference Unit Rated Water Flow Rate

- Reference Unit Rated Water Pump Input Power

Curves

Total WH Capacity Function of Temperature Curve

$$WHCapacityFuncTemp = a + b \cdot T_e + c \cdot {T_e}^2 + d \cdot EWT + e \cdot EWT^2 + f \cdot T_e \cdot EWT$$

Where:

T_e is either the evaporator wet-bulb or dry-bulb temperature

EWT is the entering water temperature of the condenser

COP Function of Temperature Curve

$$WHCOPFuncTemp = a + b \cdot T_e + c \cdot {T_e}^2 + d \cdot EWT + e \cdot EWT^2 + f \cdot T_e \cdot EWT$$

Where:

T_e is either the evaporator wet-bulb or dry-bulb temperature

EWT is the entering water temperature of the condenser

Water flow fraction: Set to 1 because we will not control it—equipment controls water pump to achieve an EWT to LWT temperature rise of 8°C.

Cooling coil air flow fraction: Controlled by AHU supply air fan speed – in cooling coil section

Select the options to set the water and air flow fraction factors as included in the capacity and COP curves in EnergyPlus. Additionally, capacity and COP are functions of the temperature curve at each speed.

Appendix B: Variable-Speed MFHP EnergyPlus Performance Curve Regressions

Space Cooling

At the rating conditions of:

Space Cooling $T_{OA,DB} = 95°\text{F}(35°\text{C})$ and $T_{RA,WB} = 67°\text{F}(17.22°\text{C})$.

For ordinary least squares fit to space cooling capacity, with a compressor speed of 44 Hz normalized to 1 at 7.178 kW of capacity, and a sensible heat ratio of 0.699, the regression terms are:

Term	Capacity Coefficients	COP Coefficients
Intercept	2.677E+00	3.477E+01
OA_db	3.136E-02	-1.631E-02
OA_db^2	-7.936E-05	4.630E-04
OA_db*RA_wb	-3.158E-04	-3.527E-03
OA_db*Speed	-6.320E-04	8.819E-02
RA_wb^2	1.206E-03	1.115E-02
RA_wb*Speed	2.354E-02	-1.148E-01
RA_wb	-1.263E-01	-9.020E-01
Comp_Speed^2	-4.295E-01	-1.160E+00

The complete regression statistics for a fit with compressor speed as an independent variable are:

```
                    OLS Regression Results (For Capacity)
==============================================================================
Dep. Variable:                      y   R-squared:                       0.987
Model:                            OLS   Adj. R-squared:                  0.982
Method:                 Least Squares   F-statistic:                     210.6
Date:                Thu, 02 Oct 2025   Prob (F-statistic):           5.74e-19
Time:                        14:19:46   Log-Likelihood:                 59.449
No. Observations:                  31   AIC:                            -100.9
Df Residuals:                      22   BIC:                            -87.99
Df Model:                           8
Covariance Type:            nonrobust
======================================================================================
                         coef    std err          t      P>|t|      [0.025      0.975]
--------------------------------------------------------------------------------------
Intercept              2.6774      5.990      0.447      0.659      -9.744      15.099
OA_DB                  0.0314      0.020      1.595      0.125      -0.009       0.072
OA_DB^2            -7.936e-05   4.22e-05     -1.880      0.073      -0.000     8.2e-06
OA_DB*RA_WB           -0.0003      0.000     -1.173      0.253      -0.001       0.000
OA_DB*Comp_Speed      -0.0006      0.002     -0.398      0.694      -0.004       0.003
RA_WB^2                0.0012      0.001      0.890      0.383      -0.002       0.004
RA_WB*Comp_Speed       0.0235      0.003      8.099      0.000       0.018       0.030
RA_WB                 -0.1263      0.178     -0.708      0.486      -0.496       0.244
Comp_Speed^2          -0.4295      0.056     -7.613      0.000      -0.546      -0.312
==============================================================================
Omnibus:                        8.118   Durbin-Watson:                   2.370
Prob(Omnibus):                  0.017   Jarque-Bera (JB):                6.553
Skew:                          -0.910   Prob(JB):                       0.0378
Kurtosis:                       4.327   Cond. No.                     9.68e+06
==============================================================================
```

```
                       OLS Regression Results (For COP)
==============================================================================
Dep. Variable:                      y   R-squared:                       0.936
Model:                            OLS   Adj. R-squared:                  0.913
Method:                 Least Squares   F-statistic:                     40.29
Date:                Thu, 02 Oct 2025   Prob (F-statistic):           2.20e-11
Time:                        13:19:53   Log-Likelihood:                -12.997
No. Observations:                  31   AIC:                             43.99
Df Residuals:                      22   BIC:                             56.90
Df Model:                           8
Covariance Type:            nonrobust
======================================================================================
                         coef    std err          t      P>|t|      [0.025      0.975]
--------------------------------------------------------------------------------------
Intercept             34.7685     61.991      0.561      0.581     -93.794     163.331
OA_DB                 -0.0163      0.204     -0.080      0.937      -0.438       0.406
OA_DB^2                0.0005      0.000      1.060      0.301      -0.000       0.001
OA_DB*RA_WB           -0.0035      0.003     -1.266      0.219      -0.009       0.002
OA_DB*Comp_Speed       0.0882      0.016      5.367      0.000       0.054       0.122
RA_WB^2                0.0111      0.014      0.795      0.435      -0.018       0.040
RA_WB*Comp_Speed      -0.1148      0.030     -3.816      0.001      -0.177      -0.052
RA_WB                 -0.9020      1.846     -0.489      0.630      -4.730       2.926
Comp_Speed^2          -1.1599      0.584     -1.987      0.060      -2.371       0.051
==============================================================================
Omnibus:                        5.757   Durbin-Watson:                   2.417
Prob(Omnibus):                  0.056   Jarque-Bera (JB):                4.070
Skew:                          -0.723   Prob(JB):                        0.131
Kurtosis:                       4.030   Cond. No.                     9.68e+06
==============================================================================
```

By setting the compressor speed variable to the desired fraction, one can obtain the coefficients needed for the EnergyPlus object. The compressor speed that delivers the rated capacity at rating conditions is defined as 100% compressor speed, full speed. The four speeds shown below are an example, and up to 10 may be entered in EnergyPlus.

Coefficients at compressor speed fractions				
Speed	25%	50%	75%	100%
OA_db^2	-8.800E-05	-8.800E-05	-8.800E-05	-8.800E-05
RA_wb^2	1.041E-03	1.041E-03	1.041E-03	1.041E-03
OA_db*RA_wb	-3.250E-04	-3.250E-04	-3.250E-04	-3.250E-04
OA_db	3.342E-02	3.327E-02	3.311E-02	3.296E-02
RA_wb	-9.896E-02	-9.308E-02	-8.720E-02	-8.132E-02
Intercept	1.897E+00	1.818E+00	1.687E+00	1.503E+00

Space Heating

At the rating conditions of:

Space Heating $T_{OA,DB} = 47°\text{F}(8.3°\text{C})$ and $T_{RA,DB} = 70°\text{F}(21.1°\text{C})$.

For ordinary least squares fit to space heating capacity, with a compressor speed of 48 Hz normalized to 1 at 7.10 kW of capacity:

Term	Capacity Coefficients	COP Coefficients
Intercept	3.396E-02	6.089E+00

Term	Capacity Coefficients	COP Coefficients
OA_db	-4.35E-03	4.704E-02
RA_db^2	8.55E-07	-9.146E-04
RA_db*Comp_Speed	-6.71E-03	8.039E-02
OA_db*2	7.30E-05	6.057E-04
OA_db*Comp_Speed	1.28E-02	-4.677E-02
Comp_Speed	8.75E-01	-4.778E+00

The complete regression statistics for a fit with compressor speed as an independent variable are:

```
==============================================================================
Dep. Variable:                      y   R-squared:                       0.995
Model:                            OLS   Adj. R-squared:                  0.993
Method:                 Least Squares   F-statistic:                     484.0
Date:                Tue, 02 Sep 2025   Prob (F-statistic):           2.71e-16
Time:                        17:35:17   Log-Likelihood:                 48.491
No. Observations:                  22   AIC:                            -82.98
Df Residuals:                      15   BIC:                            -75.35
Df Model:                           6
Covariance Type:            nonrobust
=====================================================================================
                        coef    std err          t      P>|t|      [0.025      0.975]
-------------------------------------------------------------------------------------
Intercept             0.0340      0.169      0.201      0.843      -0.326       0.394
OA_DB                -0.0043      0.003     -1.625      0.125      -0.010       0.001
RA_DB^2            8.547e-07   3.24e-05      0.026      0.979   -6.83e-05       7e-05
RA_DB*Comp_Speed     -0.0067      0.006     -1.100      0.289      -0.020       0.006
OA_DB*2            7.301e-05   3.13e-05      2.331      0.034    6.25e-06       0.000
OA_DB*Comp_Speed      0.0128      0.001     12.649      0.000       0.011       0.015
Comp_Speed            0.8751      0.431      2.030      0.060      -0.044       1.794
==============================================================================
Omnibus:                       15.128   Durbin-Watson:                   2.083
Prob(Omnibus):                  0.001   Jarque-Bera (JB):               16.070
Skew:                          -1.425   Prob(JB):                     0.000324
Kurtosis:                       6.067   Cond. No.                     3.59e+05
==============================================================================
```

```
                   OLS Regression Results (For COP)
==============================================================================
Dep. Variable:                      y   R-squared:                       0.936
Model:                            OLS   Adj. R-squared:                  0.911
Method:                 Least Squares   F-statistic:                     36.83
Date:                Thu, 02 Oct 2025   Prob (F-statistic):           3.81e-08
Time:                        14:25:53   Log-Likelihood:                -6.9097
No. Observations:                  22   AIC:                             27.82
Df Residuals:                      15   BIC:                             35.46
Df Model:                           6
Covariance Type:            nonrobust
=====================================================================================
                        coef    std err          t      P>|t|      [0.025      0.975]
-------------------------------------------------------------------------------------
Intercept             6.0895      2.096      2.906      0.011       1.623      10.556
OA_DB                 0.0470      0.033      1.417      0.177      -0.024       0.118
RA_DB^2              -0.0009      0.000     -2.272      0.038      -0.002   -5.65e-05
RA_DB*Comp_Speed      0.0804      0.076      1.063      0.305      -0.081       0.242
OA_DB*2               0.0006      0.000      1.559      0.140      -0.000       0.001
OA_DB*Comp_Speed     -0.0468      0.013     -3.734      0.002      -0.073      -0.020
Comp_Speed           -4.7780      5.349     -0.893      0.386     -16.178       6.622
==============================================================================
Omnibus:                        3.991   Durbin-Watson:                   2.259
Prob(Omnibus):                  0.136   Jarque-Bera (JB):                1.563
Skew:                          -0.214   Prob(JB):                        0.458
Kurtosis:                       1.766   Cond. No.                     3.59e+05
==============================================================================
```

By setting the compressor speed variable to the desired fraction, one can obtain the coefficients needed for the EnergyPlus object. The four speeds shown below are an example, and up to 10 may be entered in EnergyPlus.

Coefficients at compressor speed fractions				
Speed	25%	50%	75%	100%
Intercept	3.396E-02	3.396E-02	3.396E-02	3.396E-02

	Coefficients at compressor speed fractions			
OA_db	3.193E-03	6.385E-03	9.578E-03	1.277E-02
RA_db^2	8.547E-07	8.547E-07	8.547E-07	8.547E-07
OA_db*2	7.301E-05	7.301E-05	7.301E-05	7.301E-05
RA_db	-1.677E-03	-3.354E-03	-5.030E-03	-6.707E-03

Water Heating

At the rating conditions of:

Water Heating $T_{OA,DB} = 47°\text{F}(8.3°\text{C})$

For ordinary least squares fit to space heating capacity, with a compressor speed of 48 Hz normalized to 1 at 8.0 kW of capacity:

Term	Capacity Coefficients	COP Coefficients
Intercept	-2.007E+00	2.18E+01
OA_db	4.579E-03	-0.151557
OA_db^2	3.323E-05	0.001744
HX_in	1.199E-02	-0.099745
HX_in^2	-8.935E-05	0.000479

Term	Capacity Coefficients	COP Coefficients
OD_db*HX_in	1.350E-05	-1.22E-03
Comp_Speed	3.212E+00	6.34E+00
Comp_Speed^2	-1.028E+00	-7.00E+00

The complete regression statistics for a fit with compressor speed as an independent variable are:

```
                            OLS Regression Results  (For COP)
==============================================================================
Dep. Variable:                      y   R-squared:                       0.998
Model:                            OLS   Adj. R-squared:                  0.998
Method:                 Least Squares   F-statistic:                     1757.
Date:                Thu, 02 Oct 2025   Prob (F-statistic):           1.80e-26
Time:                        12:43:40   Log-Likelihood:                 27.822
No. Observations:                  28   AIC:                            -39.64
Df Residuals:                      20   BIC:                            -28.99
Df Model:                           7
Covariance Type:            nonrobust
================================================================================
                 coef    std err          t      P>|t|      [0.025      0.975]
--------------------------------------------------------------------------------
Intercept     21.8362      4.726      4.620      0.000      11.977      31.695
OA_DB         -0.1516      0.068     -2.226      0.038      -0.294      -0.010
OA_DB^2        0.0017      0.000      4.960      0.000       0.001       0.002
HX_in         -0.0997      0.051     -1.946      0.066      -0.207       0.007
HX_in^2        0.0005      0.000      1.841      0.081   -6.38e-05       0.001
OD_DB*HX_in   -0.0012   5.91e-05    -20.591      0.000      -0.001      -0.001
comp_speed     6.3374      2.952      2.147      0.044       0.180      12.495
comp_speed^2  -7.0034      1.931     -3.627      0.002     -11.031      -2.975
==============================================================================
Omnibus:                        2.663   Durbin-Watson:                   2.291
Prob(Omnibus):                  0.264   Jarque-Bera (JB):                1.384
Skew:                           0.183   Prob(JB):                        0.501
Kurtosis:                       1.974   Cond. No.                     3.41e+06
==============================================================================
```

```
                          OLS Regression Results (For Capacity)
==============================================================================
Dep. Variable:                      y   R-squared:                       0.995
Model:                            OLS   Adj. R-squared:                  0.994
Method:                 Least Squares   F-statistic:                     610.5
Date:                Thu, 02 Oct 2025   Prob (F-statistic):           6.75e-22
Time:                        12:46:01   Log-Likelihood:                 82.724
No. Observations:                  28   AIC:                            -149.4
Df Residuals:                      20   BIC:                            -138.8
Df Model:                           7
Covariance Type:            nonrobust
================================================================================
                 coef    std err          t      P>|t|      [0.025      0.975]
--------------------------------------------------------------------------------
Intercept     -2.0069      0.665     -3.017      0.007      -3.394      -0.619
OA_DB          0.0046      0.010      0.478      0.638      -0.015       0.025
OA_DB^2     3.323e-05   4.95e-05      0.671      0.510      -7e-05       0.000
HX_in          0.0120      0.007      1.661      0.112      -0.003       0.027
HX_in^2    -8.935e-05   3.66e-05     -2.439      0.024      -0.000   -1.29e-05
OD_DB*HX_in  1.35e-05   8.31e-06      1.624      0.120   -3.84e-06    3.08e-05
comp_speed     3.2117      0.415      7.730      0.000       2.345       4.078
comp_speed^2  -1.0282      0.272     -3.783      0.001      -1.595      -0.461
==============================================================================
Omnibus:                        2.509   Durbin-Watson:                   2.232
Prob(Omnibus):                  0.285   Jarque-Bera (JB):                1.249
Skew:                          -0.049   Prob(JB):                        0.535
Kurtosis:                       1.970   Cond. No.                     3.41e+06
==============================================================================
```

By setting the compressor speed variable to the desired fraction, one can obtain the coefficients needed for the EnergyPlus object. The four speeds shown below are an example for the capacity coefficients, and up to 10 may be entered in EnergyPlus.

	Capacity coefficients at compressor speed fractions			
Speed	25%	50%	75%	100%

	Capacity coefficients at compressor speed fractions			
Intercept	-1.268E+00	-6.580E-01	-1.764E-01	1.767E-01
OA_db	4.579E-03	4.579E-03	4.579E-03	4.579E-03
OA_db^2	3.323E-05	3.323E-05	3.323E-05	3.323E-05
HX_in	1.199E-02	1.199E-02	1.199E-02	1.199E-02
HX_in^2	-8.935E-05	-8.935E-05	-8.935E-05	-8.935E-05
OD_db*HX_in	1.350E-05	1.350E-05	1.350E-05	1.350E-05

Heat Recovery

At the rating conditions of $T_{IA,DB} = 80°\mathrm{F}(26.7°\mathrm{C})$, $T_{IA,WB} = 67°\mathrm{F}(19.5°\mathrm{C})$, and compressor speed of 35 Hz

For ordinary least squares fit of total capacity, sum of SC and WH capacity, normalized to 1 at 14.358 kW:

Term	Capacity Coefficients	COP Coefficients
Intercept	-8.227E+00	4.17E+04
RA_WB	2.598E-01	-2.23E+03

Term	Capacity Coefficients	COP Coefficients
RA_WB^2	-1.848E-03	2.55E+01
HX_In	7.621E-03	6.07E+02
HX_In^2	-3.530E-05	-1.02E+00
RA_WB*HX_In	-6.441E-05	-7.26E+00

In Heat Recovery mode the thermal energy removed from air in SC and a significant fraction of the electrical power consumed are delivered as WH capacity. For Heat Recovery capacity performance curves, the ratio of capacity delivered to WH over SC capacity is set at 1.10, the average of the measured runs, with standard deviation of 0.062. For example, if 1 kw of space cooling is delivered, 1.1 kw of water heating will be delivered. It is likely that the EnergyPlus Energy Management System will be needed to augment or replace the built-in objects to accurately calculate performance.

```
                            OLS Regression Results (For Capacity)
==============================================================================
Dep. Variable:                      y   R-squared:                       0.985
Model:                            OLS   Adj. R-squared:                  0.977
Method:                 Least Squares   F-statistic:                     127.2
Date:                Thu, 02 Oct 2025   Prob (F-statistic):           1.02e-08
Time:                        18:40:50   Log-Likelihood:                 57.111
No. Observations:                  16   AIC:                            -102.2
Df Residuals:                      10   BIC:                            -97.59
Df Model:                           5
Covariance Type:            nonrobust
================================================================================
                   coef    std err          t      P>|t|      [0.025      0.975]
--------------------------------------------------------------------------------
Intercept       -8.2271      1.866     -4.409      0.001     -12.384      -4.070
RA_WB            0.2598      0.055      4.692      0.001       0.136       0.383
RA_WB^2         -0.0018      0.000     -4.228      0.002      -0.003      -0.001
HX_in            0.0076      0.009      0.849      0.416      -0.012       0.028
HX_in^2       -3.53e-05   2.97e-05     -1.187      0.263      -0.000     3.1e-05
RA_WB*HX_in -6.441e-05      0.000     -0.622      0.548      -0.000       0.000
==============================================================================
Omnibus:                        0.550   Durbin-Watson:                   1.305
Prob(Omnibus):                  0.760   Jarque-Bera (JB):                0.181
Skew:                           0.253   Prob(JB):                        0.913
Kurtosis:                       2.876   Cond. No.                     1.28e+07
==============================================================================
```

```
                            OLS Regression Results (For COP)
==============================================================================
Dep. Variable:                      y   R-squared:                       0.942
Model:                            OLS   Adj. R-squared:                  0.913
Method:                 Least Squares   F-statistic:                     32.53
Date:                Thu, 02 Oct 2025   Prob (F-statistic):           7.10e-06
Time:                        18:40:50   Log-Likelihood:                -109.86
No. Observations:                  16   AIC:                             231.7
Df Residuals:                      10   BIC:                             236.4
Df Model:                           5
Covariance Type:            nonrobust
===============================================================================
                  coef    std err          t      P>|t|      [0.025      0.975]
-------------------------------------------------------------------------------
Intercept    4.174e+04    6.35e+04      0.657      0.526   -9.98e+04    1.83e+05
RA_WB       -2230.9037    1885.532     -1.183      0.264   -6432.131    1970.324
RA_WB^2        25.4554      14.887      1.710      0.118      -7.714      58.625
HX_in         606.8221     305.668      1.985      0.075     -74.248    1287.892
HX_in^2        -1.0190       1.013     -1.006      0.338      -3.275       1.237
RA_WB*HX_in    -7.2572       3.528     -2.057      0.067     -15.118       0.604
==============================================================================
Omnibus:                        6.044   Durbin-Watson:                   2.493
Prob(Omnibus):                  0.049   Jarque-Bera (JB):                3.110
Skew:                          -0.885   Prob(JB):                        0.211
Kurtosis:                       4.237   Cond. No.                     1.28e+07
==============================================================================
```

Bi-Heating

In Bi-Heating mode the ratio of capacity delivered to the water tank over the capacity delivered to the air handler is not constant and was found to depend on outdoor and indoor conditions. Thus, this mode requires an additional fitted curve to accurately describe the performance. The curves below do not include the 2nd degree polynomial terms because fewer tests were performed, and adding those terms may risk over-fitting. The regressions have a very good R^2 value between 0.993 and 0.995.

At the rating conditions of:

Water Heating $T_{OA,DB} = 47°\mathrm{F}(8.3°\mathrm{C})$

For ordinary least squares fit to total capacity, with a compressor speed of 78 Hz normalized to 1 at 11.2 kW of capacity:

Term	Capacity Coefficients	COP Coefficients	Capacity Ratio Coefficients
Intercept	-8.494E+01	2.84E+05	-8.494E+01
RA_DB	1.214E+00	-3.92E+03	1.214E+00
HX_in	7.545E-01	-2.63E+03	7.545E-01
RA_DB*HX_in	-1.115E-02	3.71E+01	-1.115E-02
OD_db	2.930E-02	1.12E+01	2.930E-02
Comp_Speed	2.531E-02	-4.92E+01	2.531E-02