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Enabling Electrification of Domestic Hot Water and Space Conditioning with Multi-function Heat Pumps

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Abstract

Heat Pumps, although imperative to eliminate building onsite emissions, are required to be both energy efficient and capable of retrofitting in existing homes. In this study, we focus on Multi-function Heat Pump (MFHP) system, that uses one outdoor unit to provide both domestic hot water (DHW) along with space cooling and heating. An air-to-air MFHP, demonstrated in this study, reduces barriers to home electrification by avoiding electric resistance and lowering peak demand. The MFHP system is retrofitted in a 2000 sq-ft residential home to evaluate ease of installation, reliability, energy efficiency, and thermal comfort in terms of the space and water temperatures. Performance metrics of the MFHP are monitored over the summer in various modes and reported in this study. Coefficient of Performance (COP) of the AC mode was observed to be about 2.8 for an outdoor dry bulb of 35°C and COP of the water heating mode was observed to be about 2.3 for an outdoor dry bulb of 19.7°C. MFHP system provides a unique opportunity to move the heat from space into the DHW tank while operating in a simultaneous space cooling and hot water heating mode. The simultaneous mode performed 36% better than individual two modes if operated separately. The study demonstrates energy savings potential with MFHP while adequately meeting multiple loads in residential buildings without electric resistance heating. Refrigerant charge optimization and advanced controls to increase the run-time of the simultaneous AC and DHW mode are the next steps to improve the air-to-air MFHP system.

Keywords: Heat Pump; Domestic Hot Water, Electrification, Residential Building

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1. Introduction

Residential buildings generate about 20% of the Unites States (US) energy-related greenhouse gas (GHG) emissions mainly from heating, cooling and hot water heating [1]. The main source of residential onsite emissions is from burning fossil fuels (mainly natural gas) for space and water heating which contributes about 323 million metric tons of GHG yearly[2]. Thus, residential heat pumps (HP) for domestic hot water (DHW) and space conditioning along with a clean electricity grid is the future of the decarbonized world. It is observed that even for the current predictions of the electricity grid, installation of heat pumps to replace natural gas furnaces for residential space heating leads to an average of 45% GHG emission reduction across the US over the lifetime of the equipment [3]. Electrifying existing space and water heating with heat pumps is a critical step in building decarbonization and poses a big challenge with electrical service upgrades. Requirements for home electrical service upgrades add cost and installation delays for retrofit customers who are considering HP for space conditioning and or hot water heating [4]. Around half of all homes are expected to need electrical service panel upgrades that cause system installation delays to fully electrify [5]. Furthermore, heat pump water heaters (HPWH) come with space constraints for closet installations and require air supply for the evaporator coil.

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Multi-Function Heat Pumps (MFHP) use one efficient compressor and outdoor heat exchanger coil to provide space cooling, space heating, and domestic hot water heating. Air-to-air MFHP systems (shown in Figure 1) use refrigerant to provide end-use heating and cooling services and have the potential to be easily retrofitted in ducted central air-conditioned (AC) homes and eliminate the need for electric resistance backup heaters. Eliminating electric resistance backup heaters can greatly reduce the maximum power requirements and eliminate the need for electrical service upgrades, which is one of the major barriers to electrification. The DHW tank of the MFHP obviates the need for exhaust connections, inlet air flow, or electrical power and has a smaller footprint. Uniquely, air-to-air MFHP systems can also operate in a simultaneous space cooling and DHW heating mode, which utilizes the heat from space cooling to heat DHW. The simultaneous AC-DHW mode usually has higher efficiency and can meet both space cooling and water heating loads flexibly [6].

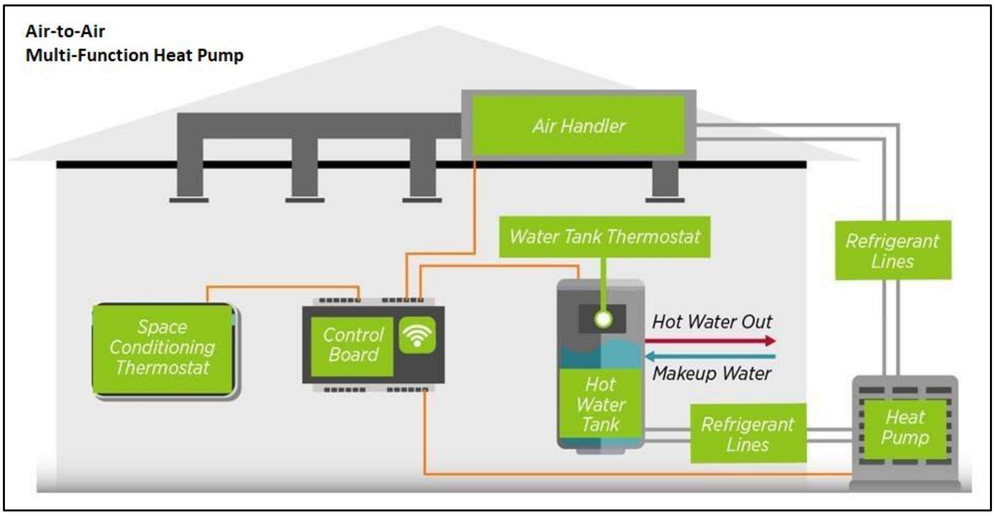

Figure 1 Air-to-air MSHP using refrigerant to provide end-use heating and cooling

Previous studies have focused on heat pumps utilizing carbon-dioxide (CO_2) as a refrigerant that can provide high temperature water for both space and water heating [7], [8]. However, these systems did not provide cooling to the household during hot summer months which is a requirement in most climate zones [9]. Heat pumps are best known for their ability to switch between cooling and heating, which is an opportunity to reduce equipment redundancies. Integrated heat pump systems have been available in Europe and a relevant testing standard was made available by International Energy Agency (IEA) through Annex 28 [10]. Although the testing procedure developed by IEA encompassed a multi-function heat pump, the scope was limited to space heating, DHW and ventilation. There has been a lack of understanding of MSHPs in space cooling (AC) with DHW mode and its overall potential to save costs and energy for the typical single-family house in the US. This study attempts to fill that gap by evaluating an air-to-air MSHP in a 2000 sq-ft household in Davis, California. Results from the study about the ease of retrofit installation, performance of the MSHP during summer, and ability of the MSHP to meet space comfort and DHW requirements are presented in this paper. Insights gained from this study will pave the path for further research in MSHP systems for easy electrification while saving energy and costs for existing residential buildings in the US.

2. Site and Installation

MSHP system uses an off-the-shelf standard split system heat pump outdoor unit with refrigerant line set serving an indoor direct expansion (DX) air handler and adds a second refrigerant line set serving a DHW tank (Figure 1). The custom made DHW tank consists of both refrigerant-to-water and water-to-water heat exchangers. City water supply flows separately through the water-to-water heat exchanger and exchanges heat with the thermal storage water from the tank. With this innovative design the DHW tank stays at ambient pressure and does not need to be certified as a pressure vessel with a relief valve. This MFHP product has been certified for capacity and efficiency for water heating by third party tests at Intertek.

As seen in Figure 2, the site before retrofit consisted of a 12.3kW (3.5 ref ton) AC, a 29.3kW (100 MBH/hr) natural gas furnace with a blower, and a 189-liter (50 gallon) tank with natural gas boiler of 11.7 kW (40

MBH/hr) capacity. The house electrical panel was limited to maximum 100 amp with the line to the utility transformer limited to maximum 125 amp. An electrification of the space and water heating would be replacing the AC with HP but would require strip heaters in the 220V air-handler requiring additional 30 amp. Additionally, a unitary hybrid HPWH installation in the closet would require 30 amp more in the electrical panel. This was avoided by electrifying the space conditioning and water heating with an air-to-air MFHP.

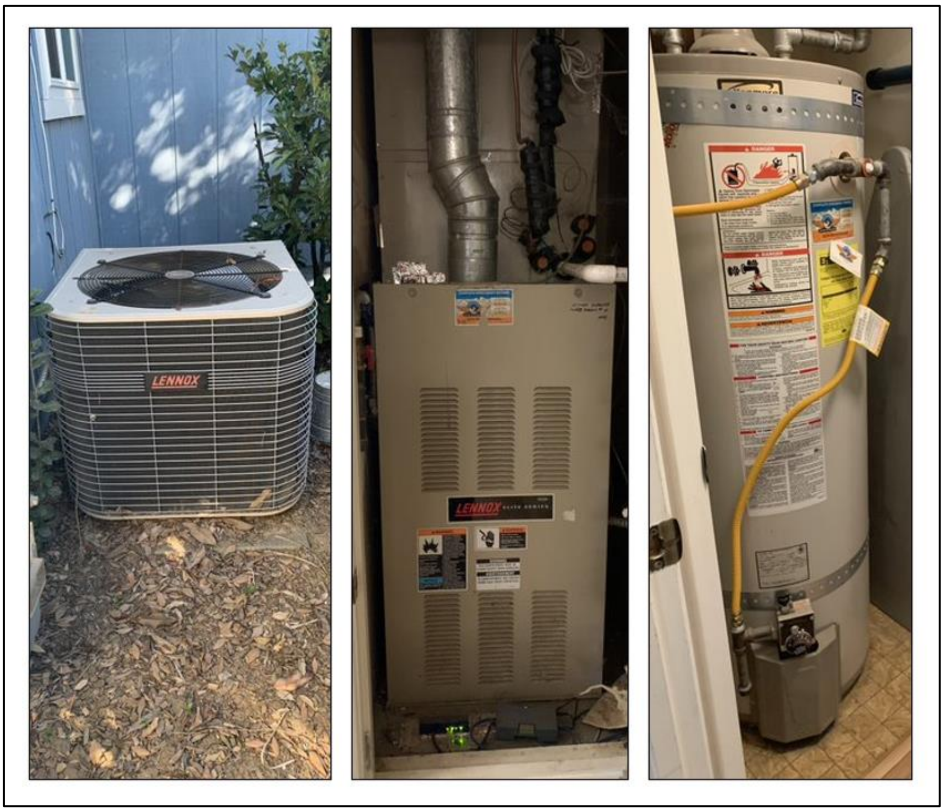

Figure 2: Pre-retrofit space conditioning and water heating equipment

The retrofitted air-to-air MFHP consisted of a 14 kW (4 ref ton) cooling capacity outdoor unit (16 SEER) connected to an air-handler in the attic, which circulates 2480 m3/hr (1460 CFM) of air through the house. The innovative refrigerant to water tank was placed in the same closet as the pre-retrofit natural gas DHW tank and did not need any additional air flow paths. Refrigerant (R410A) line sets were connected from the outdoor unit to the AHU and the DHW tank with proper temperature and sound insulation. The main control board of the MFHP is connected to the water tank thermostat and space conditioning thermostat. The main control board decides the mode of the MFHP based on the heating and cooling calls from each thermostat. For the summer monitoring period, the MFHP operated in three modes Space cooling (AC), DHW heating mode, and concurrent AC and DHW heating mode.

Figure 3: Post-retrofit MSHP components installed

3. Methodology

3.1. Instrumentation

To determine the overall performance of the MFHP system, the system was monitored using Frontier Energy's proprietary data logger. The following were instrumented and logged at 5-second intervals using the data logger:

- Detailed DHW monitoring: An Onicon ultrasonic BTU meter collected temperature, flow, and delivered energy between the hot water storage tank and the heat pump outdoor unit. Three Omega PR-10L RTDs were used to measure water temperature stratification within the DHW tank.
- Detailed HVAC system monitoring: Three ICP M-7059D modules were installed between the inputs and outputs of the MSHP controller to monitor every time the controller activates a new setting, i.e. dehumidification, cooling, heating, heat recycling, etc. Nine thermocouples were installed inside the primary supply duct leaving the supply plenum to create a 3x3 sensor averaging temperature grid to measure supply temperature. A Vaisala HMD62 was installed in the return plenum, and another in the supply plenum to measure temperature and relative humidity. A Wattnode WND-WR-MB1 True-RMS energy meter was installed on both the air handler and the outdoor unit to measure power. Five Omega PR-10L RTDs were mounted around the outdoor unit to monitor the airflow temperature entering and leaving the outdoor unit refrigerant coil.
- Weather monitoring: A Vaisala HMS110 humidity and temperature sensor was mounted outside to collect the local weather data.

In addition to the detailed DHW, HVAC, and weather monitoring, the indoor conditions of the home were monitored using an ecobee thermostat with five remote temperature sensors which provided the following data points in 5-minute intervals:

- Thermostat setpoint
- System mode
- Component operation time
- Current temperature
- Relative humidity at the ecobee
- Occupancy

The specifications for all sensors used are presented in Table 1 and Table 2.

Table 1: Data Logger Sensor Specifications.

Type	Locations	Mfr/Model	Signal	Span	Accuracy
Temperature/RH	Supply and Return Plenum	Vaisala HMD62	Modbus	-40-176°F	0.18°F
				0-100 %RH	1.5 %RH
Temperature/RH	Outside	Vaisala HMS100	Modbus	-40-140°F	0.36°F
				0-100 %RH	2 %RH
Temperature	Supply duct and return plenum	Type T	Analog	-200 to 200°C	±0.5°C
Temperature	Outdoor unit and Hot Water Tank	Omega PR-10L RTD	Analog	-50-250°C	Class A ± (0.15 + 0.002T)°C
AC Voltage Digital Input/Output State	AquaThermAire Controller	ICP DAS M-7059D	Modbus	10 – 70 V AC	N/A
RTD Analog Input Module	Outdoor unit and Hot Water Tank	ICP DAS M-7015P	Modbus	-13 – 167°F	0.05%
Power meter	Heat Pump Outdoor unit and Air Handler	Wattnode WND-WR-MB	Modbus	CT dependent	0.5%
BTU meter	Hot Water Tank	Onicon System 40	Modbus	0.15-15 gpm	±2%
					±0.18°C

Table 2. Ecobee sensor Specifications.

Type	Mfr/Model	Signal	Span	Accuracy
Temperature	ecobee and remote sensors	N/A	5 to 37 °C	±0.5 °C
Relative Humidity	ecobee and remote sensors	N/A	20 to 90% RH	±5% RH

The installation locations of the sensors and monitoring equipment are shown in Figure 4. The data logger was installed with the cell modem, in a lockable NEMA enclosure. Other equipment, such as remote sending units, were installed behind hatches or other removable coverings. The Ecobee thermostat is part of the HVAC system and property of the homeowner.

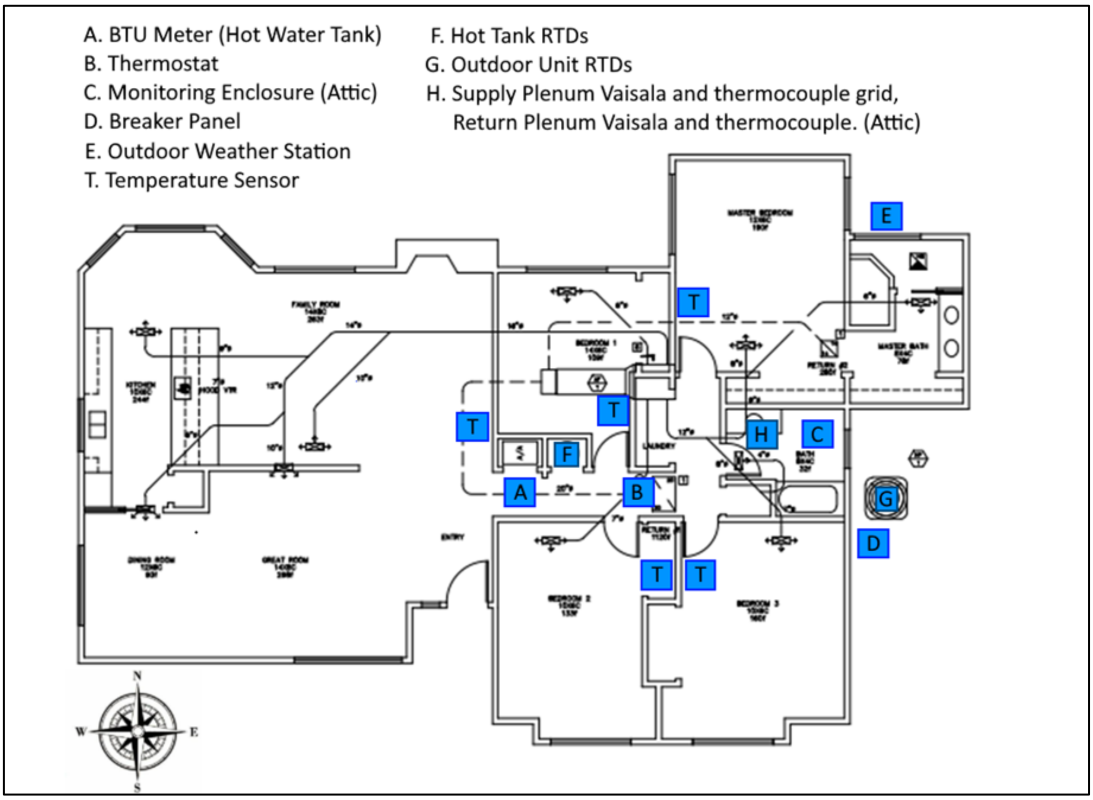

Figure 4. Monitoring System Install Locations in house floor plan

3.2. Data Analysis

3.2.1. DHW Performance

Detailed domestic hot water (DHW) system data was analyzed to calculate operating efficiencies. The BTU meters performed the necessary calculations for the amount of energy of the water delivered from the hot water tank. The standard heat balance equation follows:

$$Q_{Water} = c_{p_{water}} \rho_{water} \dot{V}_{water} \Delta T \tag{1}$$

The capacity measured from the BTU meter only accounted for the DHW energy leaving the tank as water was used throughout the home. To determine the capacity delivered to the tank during a water heating cycle, the following equation was used:

$$Q_{TES} = m_{TES} c_{p_{water}} \frac{\Delta T_{avg,Tank}}{\Delta t} \tag{2}$$

To determine the total capacity during a water heating cycle, the sum of the heat delivered to and leaving the tank was taken shown in the following equation:

$$Q_{DHW} = Q_{TES} + Q_{Water} \tag{3}$$

3.2.2. Heat Pump Performance

The output cooling capacity was calculated in real time using the enthalpies calculated using the Vaisala HMD62 sensors and Type T thermocouples in the supply and return plenums, with the one-time airflow measurement taken when commissioning the system.

$$Q_{cooling} = \rho_{air} \cdot \dot{V}_{air} \cdot \left(H_{return} - H_{supply}\right) \tag{4}$$

To determine the overall performance of the space cooling mode, the Coefficient of Performance (COP) was calculated:

$$COP = \frac{Q_{cooling}}{P_{total}} \tag{5}$$

3.2.3. Overall Performance

Due to the design of the tank (an atmospheric storage tank with two coincident coils, one carrying refrigerant and one carrying potable water), it is very difficult to measure the performance of the combined modes. For this reason, overall system efficiency was generalized using the ratio of the total daily delivered cooling and DHW energy to the total electrical energy consumed.

$$COP_{combined} = \frac{Q_{DHW}+Q_{cooling}}{P_{total}} \tag{6}$$

3.2.4. Equation Nomenclature

c_p is the specific heat.
CFM is the total return airflow in cooling.
COP is Coefficient of Performance at AHRI standard conditions.
P_{total} is the total HVAC electrical power including AHU
η is the overall system efficiency.
H is supply and return enthalpies.
m_{TES} is the mass of water within the thermal storage tank.
$Q_{cooling}$ is the cooling delivery rate.
Q_{DHW} is the total rate of heat delivered to the tank during a water heating cycle.
Q_{TES} is the rate of heat added to the thermal storage tank.
Q_{Water} is the rate of heat delivered from the thermal storage tank to the point of use.
ρ is the density.
Δt is the time period of the cycle.
ΔT is the temperature difference between the hot and cold side.
$\Delta T_{avg,Tank}$ is the change in average tank temperature during a tank heating cycle.
$\dot{V}$is the volumetric flow rate.

4. Results and Discussion

The significant findings from the summer monitoring period for the MFHP are shown below as results. An hourly average of the indoor, outdoor, and DHW tank temperatures are shown in Figure 4 (top). The modes of the MFHP observed during the summer along with their respective runtimes during that three-day period are shown as bars on the secondary y-axis. It is evident from the indoor temperature that the MFHP is able to hold the indoor temperature steady across fluctuating outdoor conditions. The first 12 hours of the day had cold outdoor conditions and did not have AC runtime, however DHW heating runtimes were triggered by significant water draws. The concurrent AC-DHW mode was also seen to run during times when there is call for both space cooling and water heating. The frequency and run-times of the concurrent mode were low, which provides a scope of improvement for the MFHP with advanced controls. Figure 4(bottom) shows the performance of the MFHP over the three-day period in terms of cooling capacity, total power consumption, and volumetric flow of hot water. The volumetric flow is represented in liters of hot water provided per minute on the secondar y-axis. The average DHW tank temperature would drop due to water draws but would go up due the MFHP operation in water heating or the concurrent AC and DHW mode. As expected, the cooling capacity was seen to be larger than the total power draw, suggesting a COP higher than unity.

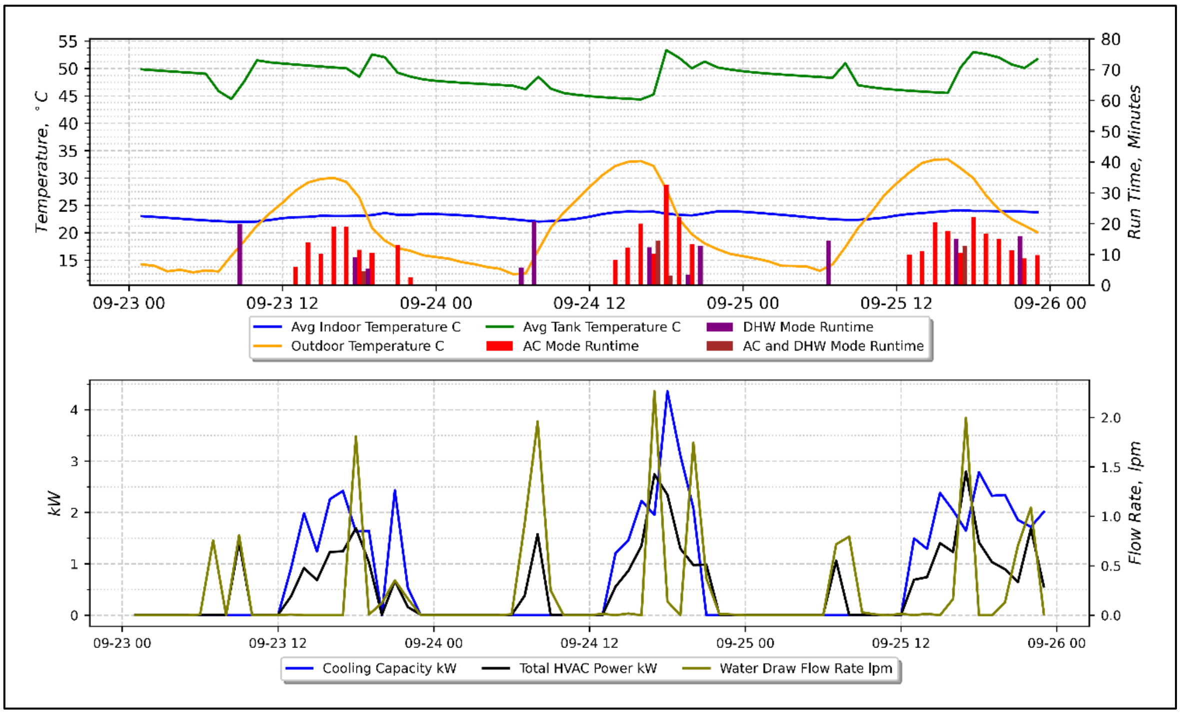

Figure 4: Hourly runtimes, average temperature, and performance of the MSHP system over a 3 day period in summer

A compressor cycle level analysis was performed to understand the efficiency of each mode of the MFHP. Coefficient of Performance (COP) for compressor cycles above four minutes in the duration between May 21 and October 21 with the outdoor dry-bulb temperature are shown in Figure 5. The cycle level COPs are sorted into three modes: AC mode (top left), DHW mode (top right), and concurrent AC and DHW mode (bottom).

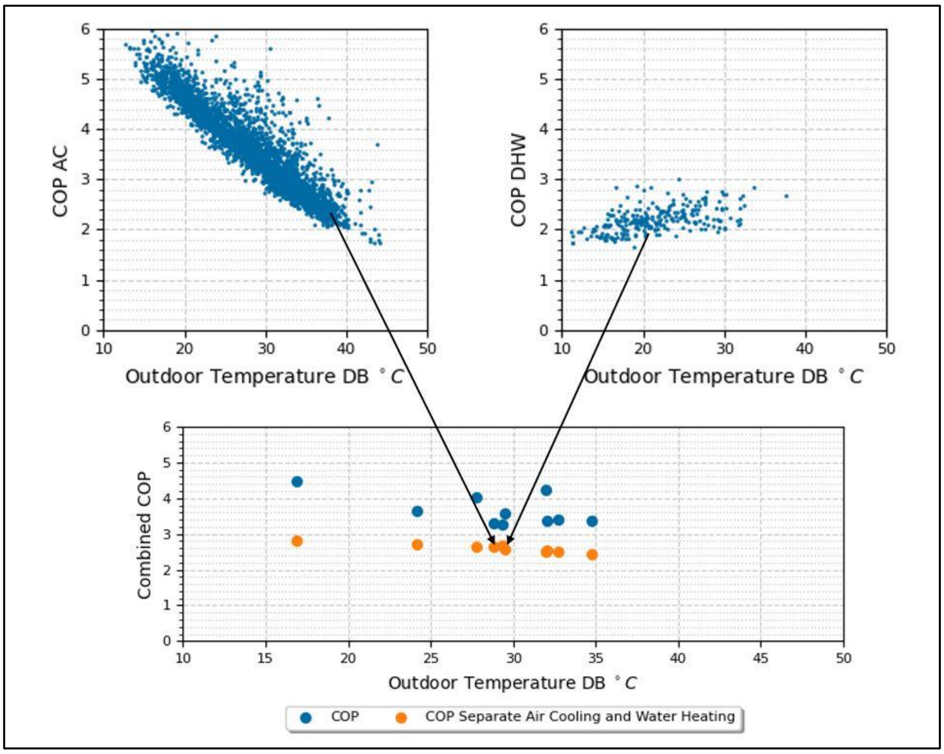

Figure 5: Compressor cycle level COP of each mode of the MFHP during the summer

Each blue dot in Figure 5 represents a compressor cycle in the respective mode. As shown, the MFHP ran very frequently in AC mode, followed by the DHW heating mode, and very infrequently in concurrent AC and DHW mode. This is because most of the concurrent mode occurrences throughout the monitoring period usually had lower than 4-minute runtimes, which was taken as the minimum amount of time for a cycle to reach steady state. The AC mode was observed to be drawing a power at 4.1 kW for a COP of 2.8 at 35°C outdoor dry bulb which was lower than the manufacturer published COP data (3.5) for a heat pump without the MFHP modifications. The reason for this difference in COP could be many. Firstly, there is higher refrigerant pressure drop expected from the addition of the three-way valves for changing modes and longer refrigerant line sets compared to the AHRI rated line set length of 25 feet. Secondly, return air conditions and supply air pressure drop in the current installation of the house are probably different than AHRI rating conditions. And lastly, the MFHP refrigerant charge is still being optimized by the manufacturer to accommodate wide range of conditions in the various modes. The DHW mode demonstrated a COP of 2.3 delivering 9.8 kW of thermal power into the water at 19.7°C outdoor dry bulb temperature, several times higher than the regular heat pump water heaters.

The bottom chart shows the measured combined COP of the concurrent mode (blue) and the combined calculated COP of the separate modes (orange), that is, if the MFHP were to run in AC and DHW mode separately to meet the same loads at the same outdoor condition. The calculated combined COP of the separate modes was based on data driven efficiency models of the AC and DHW modes separately. The measured COP of the concurrent AC and DHW mode was 36% better on average than two separate modes and thus, there is potential for energy savings by maximizing the runtime of the concurrent mode.

Figure 6 shows the transient response curve of the temperature in home to a setpoint change. The thick blue line represents the thermostat cooling setpoint which was initially set to 26.7°C as the occupants were away and was dropped to 23.3°C at 16:00 hours. The thick orange line is the temperature as measured by the thermostat and started to drop as the MFHP ran the AC mode. The thin lines with the shading in between are the individual room temperatures and shows the spread in room temperatures across the house. The dotted red line is the outdoor temperature, plotted on the secondary y-axis, stayed relatively constant over the time period. The thermostat temperature was seen to drop by 1.5°C in the first 30 minutes for the 2000 sq-ft house and was able to meet the comfort needs of the occupants. The span temperatures in all the rooms of the house denoted by the shaded region increases as the system operates due to heat gains on the house envelope and unequal distribution of air in rooms.

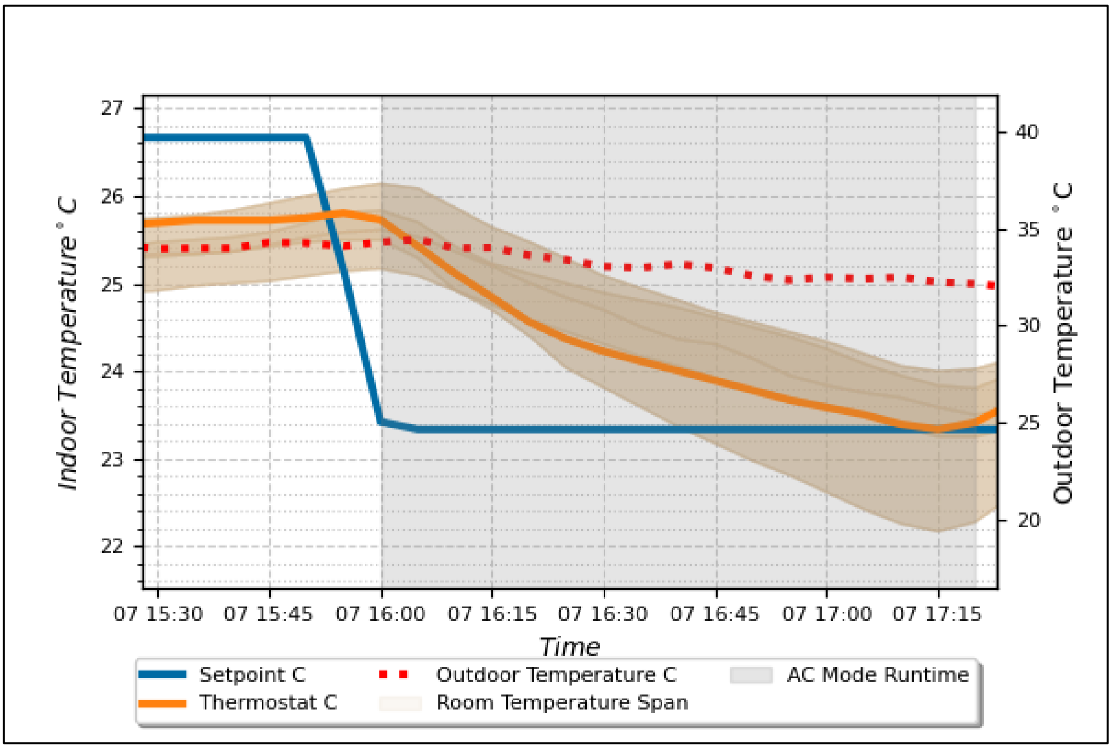

Figure 6: Indoor temperature transient response to a cooling call in AC mode

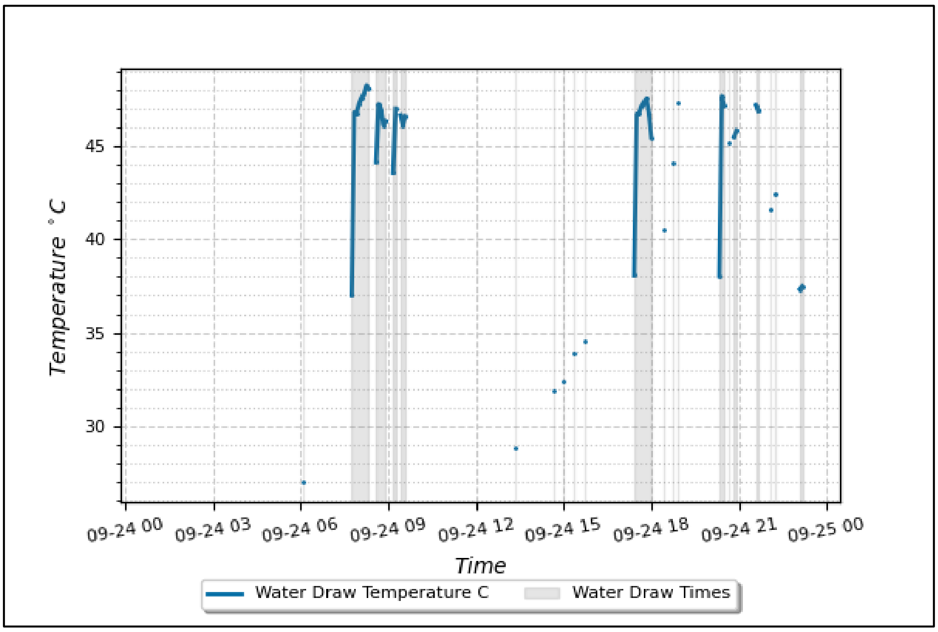

Figure 7: DHW tank delivered temperature during hot water draws

Figure 7 shows the water draw temperature over a day measured at the exit of the DHW tank during times of hot water draws, denoted by grey shading. The water temperature data was filtered on this basis to better illustrate the water delivered to point of use since the water at the tank exit cools down and does not heat up until there is flow. At start of every long draw the water temperature was seen to have a lower temperature and quickly increasing in temperature to about 45°C. This is due to the thermal mass in and around the outlet temperature sensor, which takes time to heat up to give a more accurate reading of the water flow temperature. The blue dots denote temperatures at water draws with a duration lasting less than 5 minutes. These points generally have lower temperatures similar reason to the starting measurements of the longer draws. The temperature profiles of the longer draws shows drop in temperature after a peak temperature value is achieved indicating depletion of thermal storage in the DHW tank.

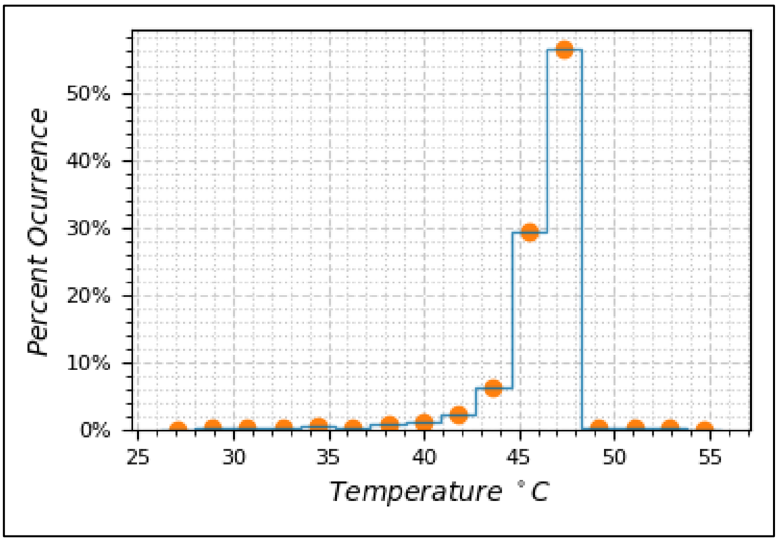

Figure 8: Hot Water temperature distribution of draws lasting above 5 minutes

The water temperature distribution shown in

Figure 8 was filtered for data when there was a water draw in the 5 minutes prior to the current water draw. This was done to filter out temperature data at the beginning of a draw before the sensor has time to heat up which was not deemed representative of hot water delivered by MFHP to meet comfort. About 15% of the total water draw occurrences, or 3.2% of the total hot water volume flow, had a hot water delivery temperature of below 45°C, which is the considered the lowest acceptable value for a shower. This shows that the MFHP system reasonably meets the hot water needs of the household during the summer while meeting the space cooling loads.

5. Conclusion

Electrification of space conditioning and hot water heating in homes require installation of separate air source heat pump equipment with each having electrical resistance heaters, which leads to increase in demand and ampacity of the electrical panel. Additional cost and complexity of electrical panel upgrades is one of the main challenges of electrification of older homes in the US. An air-to-air R410A MFHP system, capable of providing DHW and space conditioning, is investigated as a retrofit in a single-family home in Davis, California without electric resistance heaters. The system is monitored over the summer (May 21 to October 21) to analyze efficiencies of the MFHP in each mode and to evaluate its capability in meeting the space cooling and hot water loads. The household was instrumented with several sensors to calculate the metrics necessary for assessing the performance of the MFHP system and the thermal comfort of the occupants. The average COP of the MFHP AC mode was observed to be 2.8 (at 35°C Outdoor dry bulb), lower than heat pump without MFHP modifications. The reasons could be sub-optimal refrigerant charge and higher pressure drop due to longer lineset and refrigerant valving. The measured COP of the concurrent cooling and DHW mode was better than the COP estimated for meeting those loads in separate cycles by an average of 36%. Altering the controls to favor the concurrent mode rather than tackling the space cooling and DHW loads separately, this system can provide major energy and cost savings to the homeowner. Continued monitoring of the operation of MFHP and evaluating the heating performance of the system in the winter is ongoing. The MFHP space heating mode is demonstrated an average COP of 3.25 (at 8.3°C Outdoor dry bulb). Future work will assess the year-round performance and capability of MFHP in providing required household space conditioning and domestic hot water.

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