

Concurrent Space Cooling and Hot water Heating through Compact Heat Pumps for All-electric Residential Buildings

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ABSTRACT

Electrification is key to reaching decarbonization goals and curbing the impacts of climate change. Multi-function integrated heat pumps provide opportunities to reduce installation costs and space required for all-electric buildings. Multi-function heat pumps can use the compressor heat during a space cooling operation, which is otherwise wasted to the ambient through the condenser coil, to heat up water for domestic use. This project tested a multi-function heat pump that combines space cooling, dehumidification, ventilation, and domestic hot water (DHW) in a compact system (POD). Evaluation of the POD was conducted in environmental chambers simulating various indoor and outdoor conditions. Air flow and ventilation rates were calibrated, and the system was operated as per manufacturer recommendations. Performance metrics were compared across two operational modes of the POD: space cooling only and space cooling with DHW production. Operational mode decisions of the system controls impact the performance, for instance, cooling COP increases during operation with DHW heating compared to space cooling-only mode. In the combined mode, it was also observed that the cooling COP diminishes by 0.1 with every 3°F rise in tank temperature. The results indicate that a multi-function heat pump performance is not only dependent on the indoor and outdoor conditions but also on the DHW tank water temperatures. This study describes the performance of an integrated space conditioning and water heating system, while shedding light on the system metrics that would be useful to evaluate and rate such systems.

Introduction

Residential buildings generate about 20% of the United States (US) energy-related greenhouse gas (GHG) emissions mainly from heating, cooling and powering appliances (EIA 2021). Currently, the majority of residential households continue to use fossil fuels (natural gas, propane, and fuel oil) for space and water heating. With a predicted increase of 62 -105 million homes in the US by 2050 (Goldstein, Gounaridis, and Newell 2020), the electrification of end uses along with a lower carbon electricity grid will be key to achieving decarbonization goals and curbing the impacts of climate change. It is observed that even for the current 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 (Pistochini et al. 2022). Furthermore, new construction of all-electric buildings are shown to reduce homeowner's cost in all four cities evaluated with various utility rate structures around the US over the lifetime of the appliances (Billimoria et al. 2021). Electrifying residential space and water heating is a critical step in the transition to all-electric buildings, and poses several challenges such as space constraints for heat pump water heaters and overall increased cost and maintenance. These challenges are more pronounced for multi-family residences where the building owners who incur the retrofit costs do not directly benefit from the energy and operational cost savings. Rising construction costs has also added to the burden for multi-family

building owners. The state of California has set ambitious targets for building energy use, including SB350, which requires doubling of energy efficiency implementation by 2030 and expanding energy efficiency offerings in state-designated disadvantaged communities. People in existing older homes suffer not only from increased energy costs but also on the limited capacity of their electric panels, which makes it harder to electrify space and water heating.

This study was funded by the California Energy Commission as part of a bigger initiative, REALIZE, led by Rocky Mountain Institute (RMI). The goal of REALIZE is mass deployment of electrification and eliminating direct fossil fuel use in buildings, especially in affordable housing communities. This program aims at developing easily deployable retrofit packages that include both envelope and mechanical upgrades. One technology concept being evaluated is a compact multi-function heat pump to reduce the number of mechanical systems required in the package. This paper discusses the laboratory testing of one multi-function compact heat pump that can provide heating, cooling, ventilation, and domestic hot water (DHW). Previous studies have focused on heat pumps utilizing carbon-dioxide (CO₂) as a refrigerant that can provide high temperature water for both space and water heating (Stene 2005; Eklund and Banks 2016). However, these systems did not provide cooling to the household during hot summer months which is a requirement in most climate zones (Janowitz et al. 2020). 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 (Wemhoener, Afjei, and Dott 2008). Although the testing procedure developed by IEA encompassed a multi-function heat pump, the scope was limited to space heating, DHW and ventilation. Multi-function heat pumps can operate in simultaneous space cooling and DHW heating mode and can ideally have higher efficiencies. In this combined mode, the heat pump can utilize the compressor heat from space cooling, which is otherwise wasted to the ambient through the condenser coil, to heat up water for domestic use. There has been a lack of understanding of multi-function heat pump with the simultaneous cooling and DHW heating, especially for the US market. This study explores the cooling mode of a multi-function heat pump in depth and provides insight into the advantages and disadvantages of a compact integrated system.

Equipment Design

This study evaluated a multi-function all-electric residential heat pump technology made by an Italian manufacturer and currently not available for sale in the United States. This technology, called POD, provides heating, cooling, dehumidification, ventilation, and domestic hot water (DHW). Typically, homes use separate unitary systems for each of these functions. By combining the functionality there is an opportunity to reduce cost and space required for installation. To accomplish all the functions, the POD runs a complex refrigerant loop with valves and accumulators to direct the flow based on the desired function. The POD system (Figure 1) has a small footprint (32"×18"×86") that can fit into a tall closet space. The system provides balanced ventilation by removing stale air from inside the building and replacing with outdoor air. The maximum supply air flow of the POD is 240 CFM with a max ventilation air flow of 60 CFM.

The POD consists of high efficiency inverter-driven fans and variable speed compressor capable of adjusting capacity according to the system requirements. The unit has a total 48-gallon DHW storage (equally distributed between tank 1 and tank 2) and uses a combination of

electrical resistance of 1.2 kW capacity and refrigerant-water plate heat exchanger for water heating. The unit is currently manufactured for the European market and is rated for 230V and 50Hz AC power and uses R-410A refrigerant. On top of the unit there are five air flow paths (Figure 1) with each providing critical functionality. Outdoor air enters the unit, where the majority is combined with the extraction air and blown over the outdoor coil (condenser coil in cooling mode) before exhausting out of the building. A portion of the outdoor air, ventilation air, is mixed with recirculation air and blown over the indoor coil (evaporator coil in cooling mode) before supplying back to the conditioned space. Exhausting stale indoor air through the outdoor coil improves the thermodynamic efficiency of the heat pump cycle when heating or cooling is required in the building.

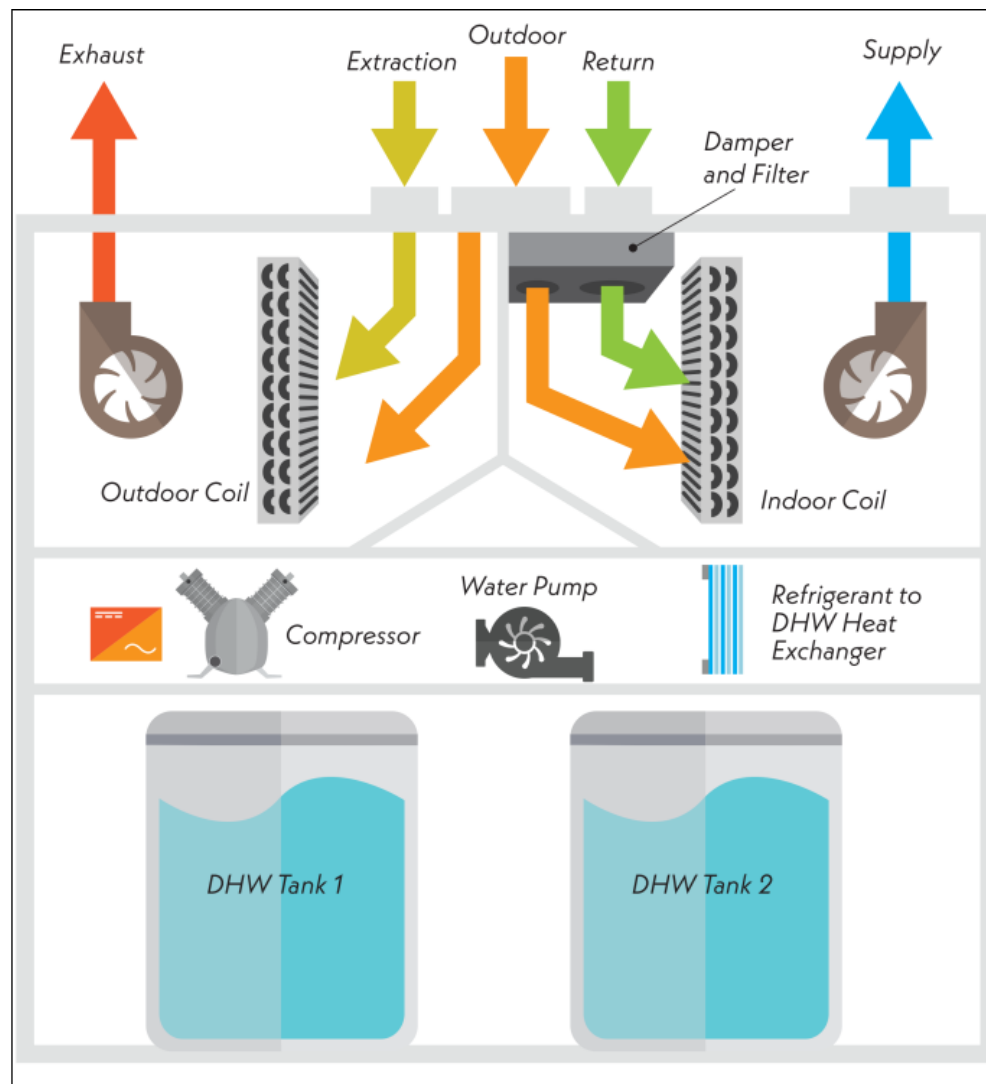


Figure 1: POD in cooling mode with air flows and internal components

Methodology

Environmental Chamber and Test Conditions

Western Cooling Efficiency Center (WCEC) at University of California, Davis, has environmental control chambers specifically designed to test HVAC equipment. The POD evaluation made use of two conditioned chambers to produce outdoor and indoor air conditions. Steady state laboratory tests of the POD were used to evaluate performance under several modes of operation. The tests focused mainly on cooling performance due to limitation of the laboratory equipment to simulate cold winter air in the outdoor chamber and because of the gap in literature on multi-function heat pump in cooling mode. As shown in Table 1, the POD was tested at four outdoor ambient conditions in cooling mode, once with the DHW function turned off and once with the DHW heating on. Performance metrics including space cooling capacity, DHW heating rate, and power draw of the unit were measured for each of the tests.

Table 1: Multi-function heat pump mode and test conditions

HP Mode	Outdoor chamber	Indoor Chamber		From inside		To inside	From outside	To outside
	Temp (°F)	Temp (°F)	Wetbulb (°F)	Extraction air (CFM)	Return air (CFM)	Supply air (CFM)	Outdoor air (CFM)	Exhaust air (CFM)
Cooling only	105	78	64	60	180	240	240	240
	95	78	64	60	180	240	240	240
	82	78	64	60	180	240	240	240
	75	78	64	60	180	240	240	240
Cooling w/ DHW	105	78	64	60	180	240	240	240
	95	78	64	60	180	240	240	240
	82	78	64	60	180	240	240	240
	75	78	64	60	180	240	240	240

Unit Configuration and Instrumentation

The POD was setup inside the indoor chamber as shown in Figure 2. The aeraulic connections for the outdoor air, supply air, and exhaust air were made using insulated 8-in flex ducting. The elbows were sealed and insulated to minimize any air and heat losses. The extraction and return air inlet were left open to the indoor chamber environment, and external dampers were installed on each to regulate the flows during commissioning. The water inlet was connected to the district water source and a pressure transducer was used to measure the inlet

pressure. Throughout the tests the water pressure was measured to be about 60 psi and was above the POD low water pressure alarm threshold. The condensate drain line was connected to a condensate pump to drain any water buildup. A frequency converter was used to convert the voltage and frequency to the required 230V and 50 Hz for the POD. All the instrumentation used for the POD testing and evaluation are listed in Table 2 along with their accuracy and signal type. The locations of the instrumentation that were installed inside the chambers are shown with respective location reference numbers in Figure 2. Critical system information such as DHW tank temperatures, valve operation, compressor speed, status of electric resistance heat, and DHW pump speed were also collected through a Modbus connection with the POD.

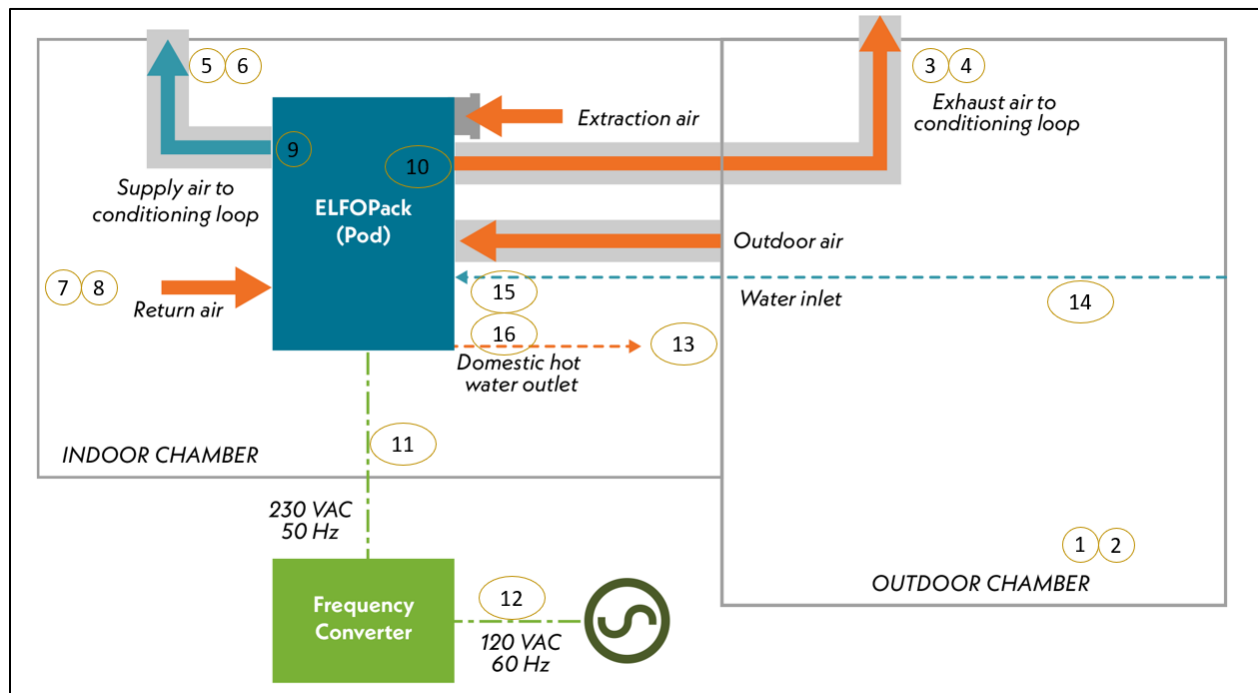


Figure 2: POD setup in the indoor chamber and instrumentation locations

Table 2: Measurement and Instrumentation

Location reference	Measurement Type	Manufacturer and Model #	Accuracy	Signal Type
1	Outdoor Air Temperature	GE OptiSonde	±0.3°F	RS-232
2	Outdoor Air Dew Point Temperature	GE OptiSonde	±0.4°F	RS-232
3	Exhaust Air Temperature	GE OptiSonde	±0.3°F	RS-232
4	Exhaust Outdoor Air Dew Point Temperature	GE OptiSonde	±0.4°F	RS-232
5	Supply Air Temperature	GE OptiSonde	±0.3°F	RS-232
6	Supply Air Dew Point Temperature	GE OptiSonde	±0.4°F	RS-232
7	Return and Extraction Air Temperature	GE OptiSonde	±0.3°F	RS-232
8	Return and Extraction Air Dew Point Temperature	GE OptiSonde	±0.4°F	RS-232
9	Delta P Supply Fan	Energy Conservatory APT	1% of reading	RS-232
10	Delta P Exhaust Fan	Energy Conservatory APT	1% of reading	RS-232

11	POD input power	Dent Powerscout 3+	1% of reading	RS-485
12	Frequency converter input power	Dent Powerscout 3+	1% of reading	RS-485
-	Delta P static Supply Duct	Energy Conservatory APT	1% of reading	RS-232
-	Upstream Flow Nozzle Pressure (Indoor Side)	Energy Conservatory APT	1% of reading	RS-232
-	Flow Nozzle Differential Pressure (Indoor Side)	Energy Conservatory APT	1% of reading	RS-232
-	Upstream Flow Nozzle Pressure (Outdoor Side)	Energy Conservatory APT	1% of reading	RS-232
-	Flow Nozzle Differential Pressure (Outdoor Side)	Energy Conservatory APT	1% of reading	RS-232
-	Indoor Chamber Static Pressure	Energy Conservatory APT	1% of reading	RS-232
-	Outdoor Chamber Static Pressure	Energy Conservatory APT	1% of reading	RS-232
-	Atmospheric Pressure	OMEGADYNE PX409-26BI	±0.08% BSL	4-20mA
-	POD parameters - DHW tank1 and tank2 temperatures - Compressor speed - DHW pump speed - DHW electric resistance - Supply and Exhaust fans speeds	Modbus	various	RS-485

Air Flow Calibration and Balance

The flow rate of the different air streams can vary based on the duct length and transitions used in any specific application. It is essential to commission the flow rates to balance the amount of air entering and exiting the conditioned space. The POD uses integrated differential pressure (DP) measurements for measuring the supply and exhaust flow for commissioning. Calibrated nozzle boxes in the outdoor and indoor chambers were used for air flow measurements and used in the analysis to calculate POD performance.

The system is intended to be setup with three ducted airflow paths that go to the building and two that go to outside. The supply air for the system was connected to the indoor chamber conditioning equipment which measured air flow rate and psychrometric conditions of supply air. The return air and stale extract air were left open to the indoor chamber conditions representing air from the building and dampers were used to simulate duct resistance for those pathways. The outdoor air and exhaust air were ducted to the outdoor chamber conditioning equipment (Figure 2).

Part of the commissioning process required that the extraction air flow is properly balanced. A calibrated fan was connected as a powered flow hood to the return vent (Figure 3) to measure the return air flow rate drawn in by the POD from the conditioned space. According to the manual and requirements of the experimental testing, the recirculation or return air flow should be set to 75% of the supply air flow rate, resulting in a 25% outdoor air fraction. At first startup, the return air flow was low and was measuring around 66% of the total supply flow. The manual shutter in the POD was adjusted to have the maximum air intake through the return duct. Finally, an external damper placed on the extraction air intake was gradually closed to achieve the desired value of the recirculation air, i.e., 75% of the supply flow. The extraction air flows over the outdoor coil of the POD and provides a thermodynamic energy recovery. Lowering the

extraction air flow rate drops the system efficiency but reduces ventilation-related thermal loads introduced in the building.



Figure 3: Calibrated fan setup as a powered flow hood to measure return air flow from conditioned space

An initial flow map of the system was created by forcing the exhaust and supply fans to run at various speeds to determine the ventilation flow rate. The range of supply flow rates possible in the POD and the associated ventilation flow rates are shown in Figure 4. For simplicity and repeatability, as shown in Table 1, all the steady state tests were performed at 100% fan speed with the maximum supply flow (240 CFM) and ventilation flow (60 CFM). However, for a safe operation in all test conditions, manufacturer configured unit controller managed the compressor speed, DHW pump speed, and electronic expansion valve (EXV) instead of manual controls of these parameters.

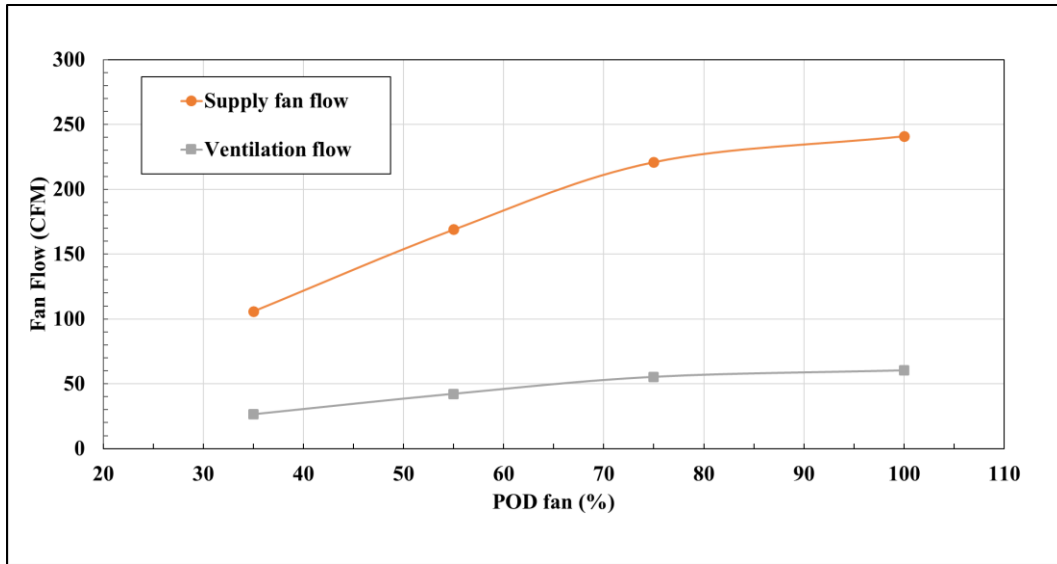


Figure 4: POD Supply flow and ventilation flow range

Performance Metrics and Associated Equations

Mixed air enthalpy. The mixed air refers to the air conditions entering the indoor coil (evaporator for cooling). Due to difficulty directly measuring this condition, the mixed air enthalpy is calculated based on the known amount of return air and outdoor air entering the indoor coil.

$$h_{MA} \left[\frac{btu}{lbm} \right] = \frac{Q_{OA}}{Q_{SA}} * h_{OA} \left[\frac{btu}{lbm} \right] + \frac{Q_{RA}}{Q_{SA}} * h_{RA} \left[\frac{btu}{lbm} \right]$$

Where:

h_{MA} is the mixed air enthalpy

Q_{OA} is the outdoor air flowrate for ventilation

Q_{SA} is the supply air flowrate

h_{OA} is the outdoor air enthalpy

Q_{RA} is the return air flowrate

h_{RA} is the return air enthalpy

Cooling capacity. The cooling capacity of the system was calculated based on the change in enthalpy between the mixed air and supply air.

$$Capacity \left[\frac{btu}{hr} \right] = \left(h_{SA} \left[\frac{btu}{lbm} \right] - h_{MA} \left[\frac{btu}{lbm} \right] \right) * \dot{m}_{SA} \left[\frac{lbm}{hr} \right]$$

Where:

h_{SA} is the supply air enthalpy

h_{MA} is the mixed air enthalpy

$\dot{m}_{SA}$ is the supply air mass flowrate

$$\text{Sensible Capacity} \left[\frac{\text{btu}}{\text{hr}} \right] = (T_{SA} [^{\circ}\text{F}] - T_{MA} [^{\circ}\text{F}]) * C_{p,air} \left[\frac{\text{btu}}{\text{lb } ^{\circ}\text{F}} \right] * \dot{m}_{SA} \left[\frac{\text{lbm}}{\text{hr}} \right]$$

Where:

T_{SA} is the supply air temperature

T_{MA} is the mixed air temperature

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

$\dot{m}_{SA}$ is the supply air mass flowrate

$$\text{Latent Capacity} \left[\frac{\text{btu}}{\text{hr}} \right] = \text{Capacity} - \text{Sensible Capacity}$$

DHW heating capacity. The DHW heating capacity was calculated based on the average of the DHW tank 1 and tank 2 water temperatures measured using probes installed by manufacturer at the mid-heights of the two tanks. The hot water is continuously recirculated by the water pump (shown in Figure 1) between the two tanks and the refrigerant to water heat exchanger. Although tank 1 and tank 2 temperatures followed the same trend at steady state, there was a small offset between the two, leading to the use of average in the calculations over the sampling duration.

$$\text{DHW Heating Capacity} \left[\frac{\text{btu}}{\text{hr}} \right] = \frac{(T_{avg,2} [^{\circ}\text{F}] - T_{avg,1} [^{\circ}\text{F}]) * C_{p,water} \left[\frac{\text{btu}}{\text{lb } ^{\circ}\text{F}} \right] * 397 [\text{lb}]}{t_2 [\text{hr}] - t_1 [\text{hr}]}$$

Where:

$T_{avg,1}$ is the tank1 and tank2 average temperature at the beginning of the sample period

$T_{avg,2}$ is the tank1 and tank2 average water temperature at the end of the sample period

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

t_1 is time at the beginning of the sample period

t_2 is time at the end of the sample period

Coefficient of Performance (COP). The coefficient of performance (COP) describes the efficiency of the system. Due to the multiple modes of operation, it is necessary to define multiple COPs depending on the particular mode. The space conditioning *COP* is used for tests where only cooling was performed. The *Combined COP* is used for tests where cooling coincided with DHW heating.

$$\text{COP} = \frac{\text{Capacity} [\text{W}]}{\text{Total Power} [\text{W}]}$$

Combined COP

$$= \frac{\text{Space Heating or Cooling Capacity} [\text{W}] + \text{DHW Heating Capacity} [\text{W}]}{\text{Total Power} [\text{W}]}$$

Results and Discussion

When a steady state condition was reached in the chambers, data was recorded over a 30-minute period for each of the test points. The POD fans were forced to run at 100% speed and corresponding chamber fan speeds were also held constant during these steady state tests. The compressor and the internal DHW pump, however, relied on the unit controller and would

modulate even during steady conditions. If the compressor speed was detected to be cycling during a particular test, caution was taken to ensure the steady state data was evaluated using full cycling periods to avoid bias.

Space cooling only

During space cooling only mode, the performance of the POD was characterized over a range of outdoor air temperatures as shown in Figure 5. The space cooling capacity of the heat pump (left y-axis) varied from about 7,500 Btu/hr to 6,200 Btu/hr as the outdoor temperature increase by 30°F. The COP (right y-axis) ranged from 2.05 at the coldest outdoor air conditions and 1.45 at the hottest outdoor condition. The POD-controlled average compressor speed is shown next to the capacity markers in Figure 5 below. The COP dropped significantly between 82°F and 95°F as the POD compressor increased speed in an attempt by controller to maintain cooling capacity. However, at very hot ambient temperatures of 105°F, the POD avoided further drop in COP by lowering the compressor speed to 64%, which led to a decrease of cooling capacity by 860 Btu/hr. This limitation of the POD in delivering capacities at higher ambient temperatures could be due to an undersized outdoor coil to reject heat. At the 95°F outdoor air condition the COP was 1.45 which is lower than Department of Energy requirement of 3.22 for packaged air conditioners in the southwestern region, which includes California.

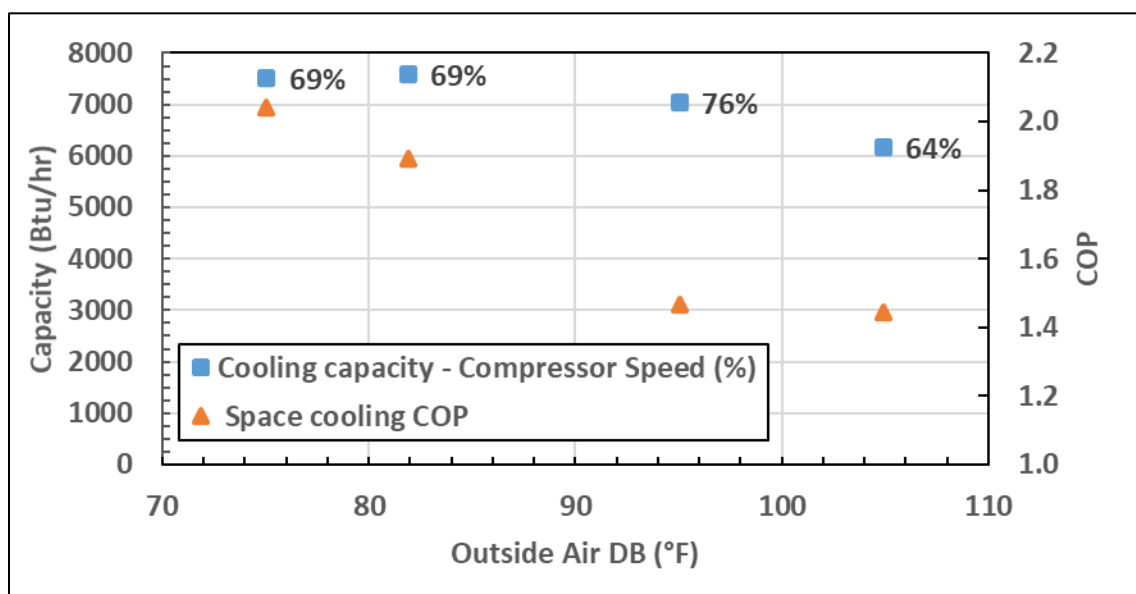


Figure 5: POD performance during space cooling only mode

Since the system brings in outdoor air for ventilation and mixes that with return air, a portion of the cooling capacity is used for cooling that ventilation air. The inlet of the supply coil was 75% recirculated indoor air and 25% outdoor air for ventilation. The indoor air temperature was fixed at 78°F dry bulb while the outdoor air varied depending on the specific test condition. As outdoor air temperatures increase, more cooling capacity is required to cool the ventilation air. Figure 6 shows the relative cooling load for the recirculated air from the building and the outdoor air for ventilation. The results show that at the lowest temperature condition of 75°F the

ventilation-related cooling only accounted for 15% of the total load, while at the hottest condition the ventilation cooling accounted for 37% of the load.

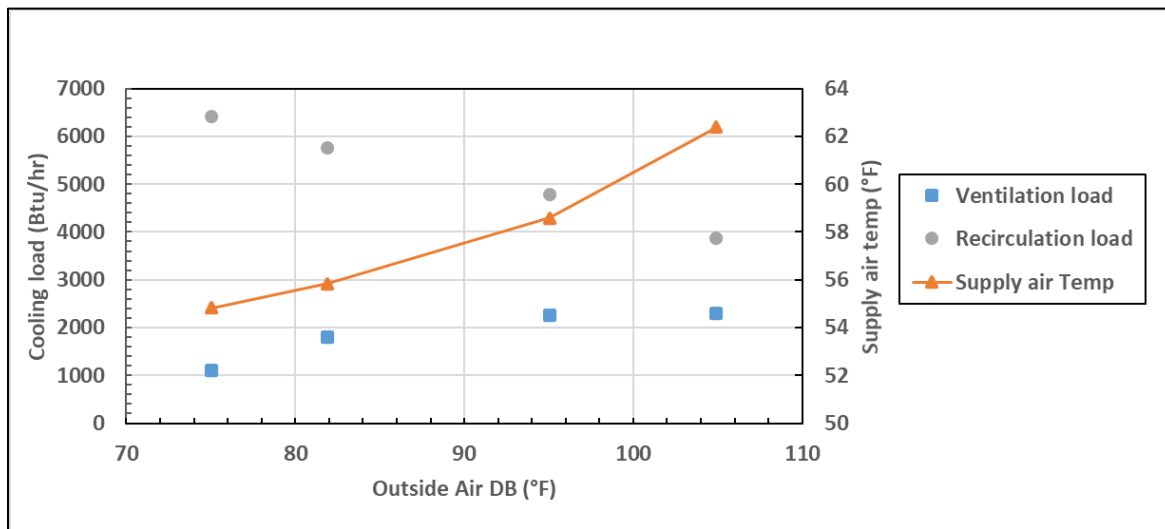


Figure 6: Cooling loads and supply air temperature during space cooling only mode

The capacity of the POD can also be divided into sensible and latent cooling loads as shown in Figure 7. The sensible cooling load stayed fairly constant while the latent load dropped. This is because as the outside air temperature increased, the mixed inlet temperature and enthalpy to the supply fan coil also increased. This leads to less condensate formation on the supply coil, and thus a lower latent cooling load.

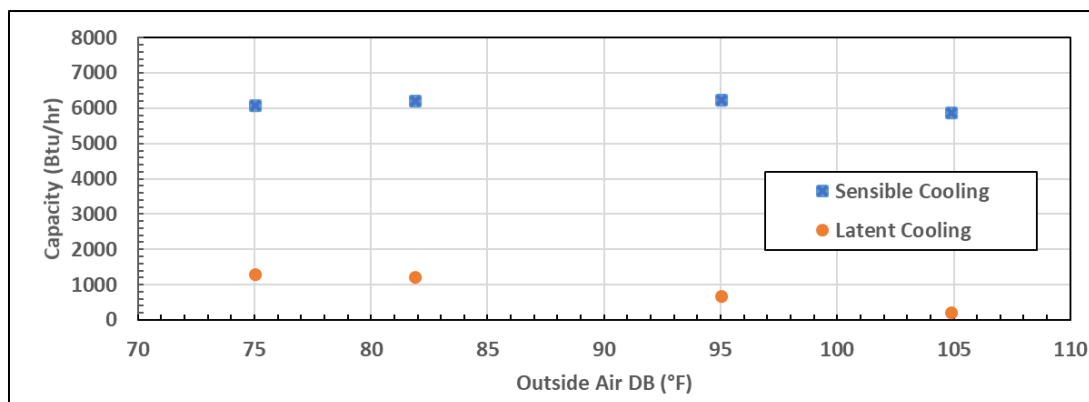


Figure 7: Sensible and latent cooling loads during space cooling only mode

Space cooling with DHW

During the space cooling and DHW heating mode, the performance of the POD was characterized over a range of outdoor air temperatures. Data was collected as the POD heated the tank 1 and tank 2 water, without any water draw, in the heat pump mode using the refrigerant to water plate heat exchanger. Electric resistance was not used by the POD controller in the DHW production as there was free heat available while the unit is cooling the space. The POD would

run the internal DHW pump to recirculate the water from the tanks to the plate heat exchanger, acting as desuperheater, and back to tanks. For the steady state characterization, a 24 to 30 min snapshot of the data was analyzed when the tank 1 and tank 2 average temperature was between 110°F and 130°F. The tank temperature probes are located at mid-heights and well mixed conditions could be assumed based on the continuous pumping operation. Similar to the space cooling only tests, it was ensured that the snapshot took whole periods of the compressor cycling.

The space cooling performance of the POD in the space cooling with DHW heating mode is quantified and illustrated in Figure 8. It shows the system efficiency and cooling capacity were improved compared to the cooling only mode tests. Furthermore, the cooling capacity increased as outdoor air temperatures increased because of the increase in compressor speed. Although this increase in compressor speed led to higher capacities, the system showed a steady decline in COP with increase in outdoor temperatures. It had become apparent during testing that the outdoor coil of the POD was undersized for the compressor capacity which resulted in the compressor running at lower speeds when only providing cooling. This is likely due to the intent for compact design of the multi-function heat pump incorporating both space conditioning and water heating. When operating with DHW production, the plate heat exchanger extends the condenser area for the system allowing the compressor to ramp to full speed and providing higher cooling capacities. When evaluating the cooling efficiency without considering the DHW capacity provided, the COP of the cooling system (right y-axis Figure 8) ranged from 2-3 across all outdoor test conditions which is about 37-50% higher than what was observed for space cooling only tests.

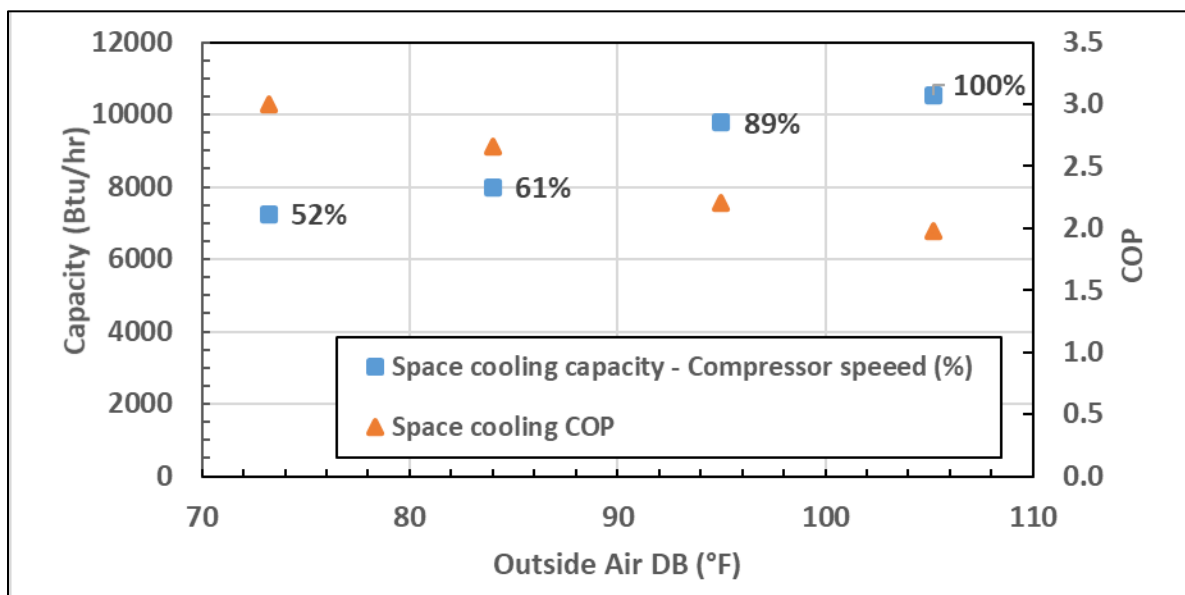


Figure 8: Space cooling capacity and COP during cooling and DHW heating mode

A closer look at the POD performance, when the compressor and the internal DHW pump were running steady at 100% speed, showed the power input to the POD increases and the cooling capacity decreases as the water temperature in the tanks rise. This is because the water temperature rise leads to higher condensing pressure of the refrigerant leading to higher lift for the compressor. To better understand the reduction in performance of the POD as DHW tank

temperatures increase, the space cooling COP at 95°F outdoor temperature is plotted against the average temperature of the tanks in Figure 9. It was observed that for every 3°F increase in water temperature the space cooling COP drops by about 0.1. The performance of the POD in Cooling and DHW mode approached that of the cooling only mode as the water temperature in the tank reached 135°F.

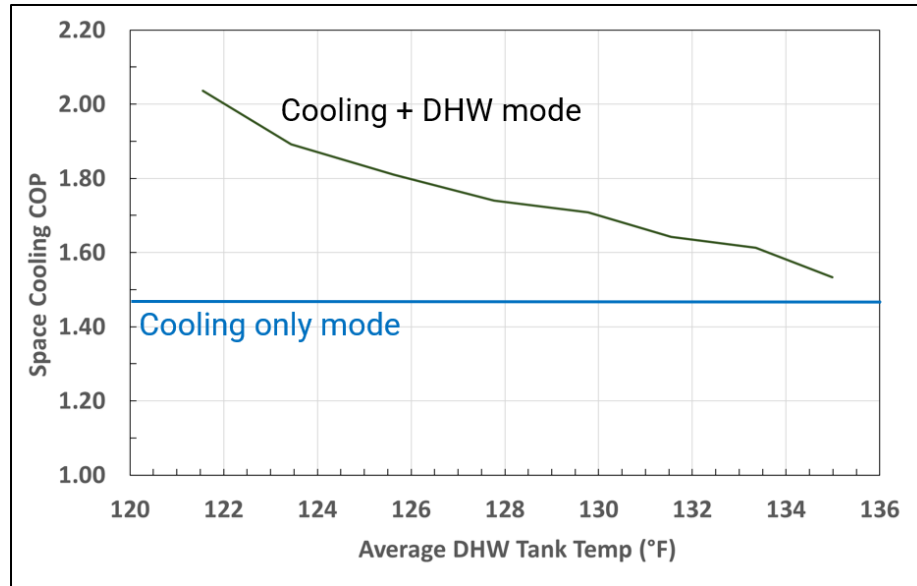


Figure 9: Space Cooling COP of POD during DHW heating

A combined COP was defined to account for both the useful heat added to the water and the space cooling load of the POD. The DHW heating capacity provided and the combined COP over the range of outdoor air temperatures are shown in Figure 10. It was observed that the performance of the POD was dependent on the DHW tank temperatures in this combined mode. Thus, the average DHW water temperature was kept at $120^{\circ}\text{F} \pm 5^{\circ}\text{F}$ for each test for comparison. The data labels in Figure 10 show the compressor speed of the unit at each condition. The DHW heating capacity and compressor speed both increase as outdoor air temperatures increase. This was because the compressor at higher speeds and higher lifts would produce hotter refrigerant temperatures resulting in increased heat transfer in the refrigerant-to-water plate heat exchanger. The combined COP is relatively flat across all outdoor air conditions ranging between 3.5 and 4 which is about double the COP for cooling only tests.

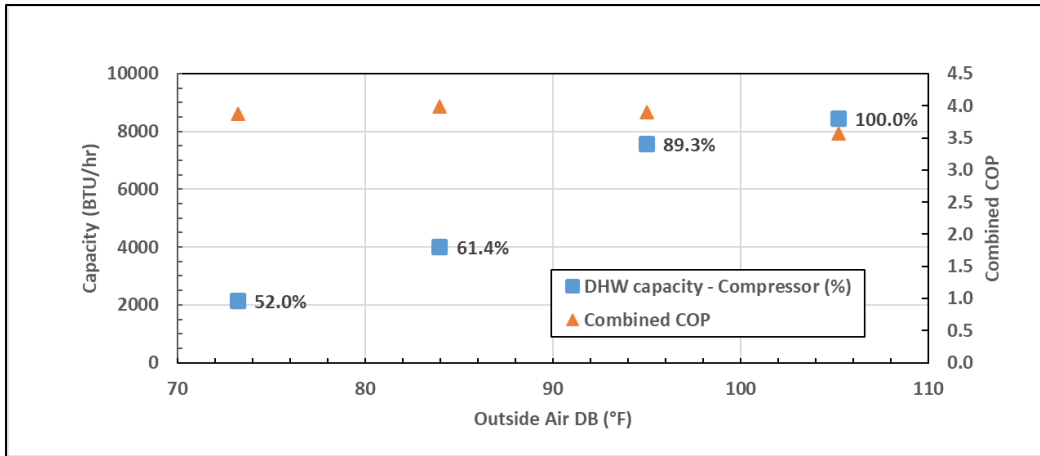


Figure 10: Water heat rate and combined COP during space cooling and DHW heating mode (data labels indicate compressor speed under each condition) with DHW water temperature at 120°F ±5°F

Conclusion

Integrated systems, that utilize a single compressor to achieve multiple functions for a building, have the potential to improve efficiency, decrease equipment redundancy and space requirements, and reduce both first cost and maintenance costs. These systems can also help the US building sector as it pursues electrification of space conditioning and water heating by reducing connected electric load and peak power draw. The multi-function heat pump (POD) tested for this project is one system that integrates these functions, as well as the ability to provide balanced ventilation in a single package. The POD uses a desuperheater to utilize the compressor heat available during space cooling to simultaneously heat DHW. The heat recovery feature of the POD allows stale air extracted from the building to be directed over the outdoor coil, lowering compressor discharge temperature and associated power draw.

The performance testing of the POD system demonstrates a tradeoff between efficiency and space constraints. The form-factor of the system was much smaller than the equipment this system would replace. The system is about the size of a residential water heater but contains the functionality of a 1-ton heat pump, a 48-gallon heat pump water heater, and an energy recovery ventilator. The lower capacity and supply flow rate suggest that the POD was manufactured with the intention to serve smaller floor areas, relative to ducted systems typically used in US. The condenser design appeared to be undersized for the compressor capacity as indicated by the reduced compressor speed when running in cooling-only conditions. The plate heat exchanger used for DHW heating effectively expands the condenser area allowing the system to take advantage of the full compressor capacity.

The measured cooling efficiency was relatively low with COPs between 1.4-2.0 for the range of test conditions. The space cooling performance improved to 2.0-3.0 when operating in cooling and DHW heating mode which shows the advantage of combining these processes. The results indicated that the POD performance is not only dependent on the outdoor conditions but also on the DHW tank water temperature. The cooling COP was shown to drop by 0.1 for every 3°F rise in water temperature in the tank. This alludes to the necessity of a rating standard and testing protocol for multi-function heat pump which is not available in the US. A combined COP metric, introduced in this study, could be used to characterize these multi-function systems.

References

- Billimoria, S., L. Guccione, M. Henchen, and L. Louis-Prescott. 2021. “The Economics of Electrifying Buildings: How Electric Space and Water Heating Supports Decarbonization of Residential Buildings.” In , 297–304. https://doi.org/10.1142/9789811213960_0033.
- DOE. 2011. "Energy Conservation Program: Energy Conservation Standards for Residential Furnaces and Residential Central Air Conditioners and Heat Pumps; Direct final rule. [EERE-2008-BT-STD-0006][EERE-2009-BT-STD-0022]" Energy Efficiency and Renewable Energy Office
- Energy Conservation Program: Energy Conservation Standards for Residential Furnaces and Residential Central Air Conditioners and Heat Pumps; Direct final rule. [EERE-2008-BT-STD-0006][EERE-2009-BT-STD-0022]
- EIA. 2021. “U.S. CO₂ Emissions from Energy Consumption by Source and Sector.” Total Energy Monthly Data - U.S. Energy Information Administration (EIA). April 2021. <https://www.eia.gov/totalenergy/data/monthly/>.
- Eklund, K., and A. Banks. 2016. “Combined Space and Water CO₂ Heat Pump System Performance Research.”
- Goldstein, B., D. Gounaridis, and J.P. Newell. 2020. “The Carbon Footprint of Household Energy Use in the United States.” *Proceedings of the National Academy of Sciences of the United States of America* 117 (32): 19122–30. https://doi.org/10.1073/PNAS.1922205117/SUPPL_FILE/PNAS.1922205117.SAPP.PDF.
- Janowitz, K., C. Dymond, C. Roos, and M. Jerome. 2020. “Is the CO₂ Heat Pump Ideal for Combination Space and Water Heating ? Field Data from Seven CO₂ Heat Pump-Based Combined Space and Water Heating Systems.” In , 126–41. ACEEE Summer Study on Energy Efficiency in Buildings. https://www.aceee.org/files/proceedings/2020/event-data/pdf/catalyst_activity_10614/catalyst_activity_paper_20200812131014166_52f12a7f.
- Pistochini, T., M. Dichter, S. Chakraborty, N. Dichter, and A. Aboud. 2022. “Greenhouse Gas Emission Forecasts for Electrification of Space Heating in Residential Homes in the US.” *Energy Policy* 163 (April): 112813. <https://doi.org/10.1016/J.ENPOL.2022.112813>.
- Stene, J. 2005. “Residential CO₂ Heat Pump System for Combined Space Heating and Hot Water Heating.” *International Journal of Refrigeration* 28 (8): 1259–65. <https://doi.org/10.1016/j.ijrefrig.2005.07.006>.
- Wemhoener, C., T. Afjei, and R. Dott. 2008. “IEA HPP Annex 28 – Standardised Testing and Seasonal Performance Calculation for Multifunctional Heat Pump Systems.” *Applied Thermal Engineering* 28 (16): 2062–69. <https://doi.org/10.1016/J.APPLTHERMALENG.2007.12.003>.