12th IEA Heat Pump Conference 2017

Prototype air-source integrated heat pump field evaluation results

Jeffrey Munk[a]*, Van Baxter[a], Anthony Gehl[a], Moonis Ally[a], Bo Shen[a], Keith Rice[a], and Robert B. Uselton[b†]

[a]*Oak Ridge National Laboratory, P.O. Box 2008, Oak Ridge, TN 37831, U.S.A.*
[b]*Lennox Industries Inc., 1600 Metrocrest Drive, Carrollton, TX 75006, U.S.A.*

Abstract

This paper presents results from a field evaluation of a prototype air-source integrated heat pump (AS-IHP) system. The system consists of a high-efficiency, variable-speed (VS) air-source heat pump (ASHP) coupled with a prototype water heating/dehumidification (WH/DH) module. Major operating modes for the AS-IHP system include space cooling (SC), space heating (SH), water heating (WH), and combined SH+WH and SC+WH. In addition the WH/DH module provides dedicated year round dehumidification (DH) and whole house ventilation capability. The AS-IHP design is particularly amenable to home HVAC/WH retrofit applications as well as to new construction. Design of the prototype was completed in spring 2015 and it was installed in an unoccupied single-family research home in June-July 2015. Field monitoring began in August 2015 and continued through September 2016. Analyses described in this paper indicate measured, average, system coefficient of performances (COPs) of 4.7 for cooling, 2.24 for heating, and average monthly WH COPs of ~2.3-2.5 and ~1.9-2.3 in summer and winter, respectively. Total energy savings relative to a minimum efficiency ASHP and electric water heater are estimated at 29% for the 2015-2016 test year. In addition, on-demand DH maintained the indoor space relative humidity (RH) level at ≤55% throughout the evaluation period. This feature will be critical to ensuring occupant comfort in low-load, high performance homes located in humid climates.

© 2017 Stichting HPC 2017.
Selection and/or peer-review under responsibility of the organizers of the 12th IEA Heat Pump Conference 2017.

Keywords: heat pump, water heating, dehumidificaiton, energy savings

1. Introduction

The Building Technologies Office of the U.S. Department of Energy (DOE) has established a goal to reduce by 2025 the energy use for space conditioning and water heating in single-family homes by 40% from 2010 levels. The baseline equipment for the energy savings comparison was an air-source heat pump with rated performance of 3.8 cooling seasonal coefficient of performance (SCOPc) and 2.3 heating seasonal coefficient of performance (SCOPh), a 0.90 Energy Factor (EF) resistance water heater, and a 1.4 EF dehumidifier. In support of this goal Oak Ridge National Laboratory and an industry partner demonstrated the performance of a near-production prototype air-source integrated heat pump (AS-IHP) through a controlled field evaluation. The AS-IHP was developed through a collaborative research and development agreement with a heat pump manufacturer

* Corresponding author. Tel.: +1-865-241-8226; fax: +1-865-241-4152.
E-mail address: munkjd@ornl.gov.

† *This paper has been coauthored by UT-Battelle, LLC under Contract No. DE-AC05-00OR22725 with the U.S. Department of Energy. The United States Government retains and the publisher, by accepting the article for publication, acknowledges that the United States Government retains a non-exclusive, paid-up, irrevocable, world-wide license to publish or reproduce the published form of this manuscript, or allow others to do so, for United States Government purposes. The Department of Energy will provide public access to these results of federally sponsored research in accordance with the DOE Public Access Plan (http://energy.gov/downloads/doe-public-access-plan).*

partner; the development process is documented in a paper by Rice et al [1]. The system that was field evaluated consisted of a commercially available variable-speed heat pump combined with a prototype heat pump that provides water heating, dehumidification, and ventilation. The system was evaluated in an unoccupied research house with simulated occupancy located in Knoxville, Tennessee.

2. Equipment description

The AS-IHP system tested consisted of a commercially available variable-speed (VS) air-source heat pump (ASHP) and a prototype unit (referred to as WH/DH) that provided water heating, dehumidification, and ventilation. The ASHP was rated at 10.1 kW (34.4 kBtu/h) of cooling with a $SCOP_c$ of 6.3 and DOE Climate Region IV $SCOP_h$ of 2.9 [2]. The major components of the prototype WH/DH are a single-speed compressor, single-speed water pump, VS fan, fin-and-tube refrigerant-to-air evaporator, brazed plate refrigerant-to-water condenser, and fin-and-tube refrigerant-to-air condenser with refrigerant circuiting as depicted in Figure 1. The WH/DH prototype unit uses a solid-state microcontroller to manage competing requests for service with water heating having priority over dehumidification. The VS blower uses the same speed for water heating and dehumidification but slows down for controlled fresh-air ventilation.

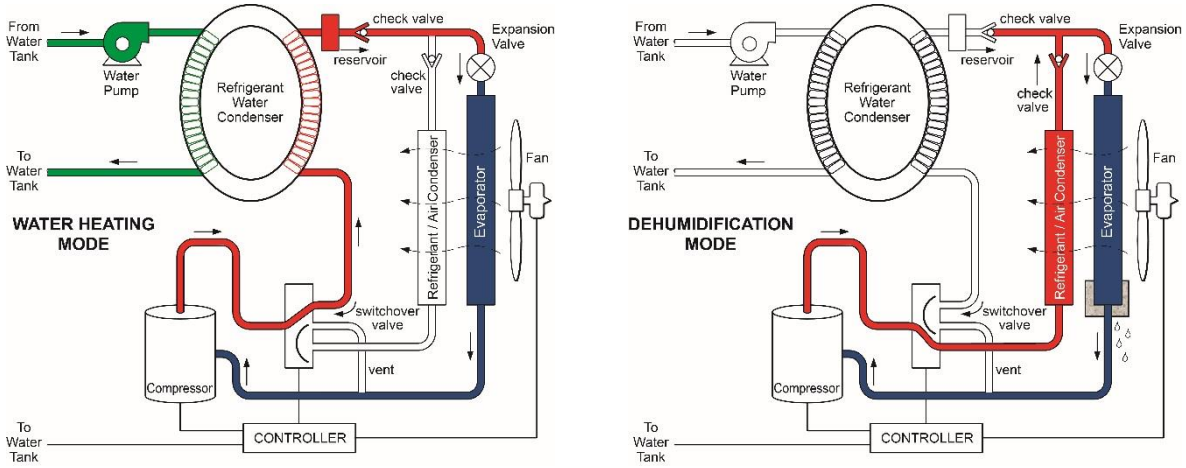


Figure 1: Refrigerant circuiting of WH/DH module for the water heating (left) and dehumidification (right) operating modes

3. Field test setup

3.1. House characteristics and simulated occupancy

The field test house was built in 2013 and is of typical construction for the region. The unoccupied, 2-story home had 223 m^2 (2,400 ft^2) of finished floor area and a slab foundation. Electric heaters and ultrasonic humidifiers were cycled to simulate internal sensible and latent heat gains due to occupants, appliances, lighting, etc. based on the Building America House Simulation Protocol [3]. The average daily internal sensible and latent gains were 20.0 kWh/day and 8.3 L/day respectively.

The hot water draw schedule was created using the Building America Domestic Hot Water Event Schedule Generator detailed in a report by Hendron et al [4]. This tool generates a year-long schedule of hot water draws based on realistic probability distributions of hot water draws, providing significant variation from day to day.

3.2. Equipment setup

The space conditioning system included zone controls and dampers that allowed the upstairs and downstairs zone temperatures to be controlled independently. The zoning system also controlled the ASHP airflow based on fixed airflow values that were assigned to each zone during commissioning of the system. The thermostat set points were 21.7°C (71.0°F) and 24.4°C (76.0°F) for the heating and cooling seasons respectively. The operating mode of the ASHP was switched manually between heating only and cooling only as needed.

The WH/DH was connected to a standard electric storage water heater with copper pipe and a concentric fitting that was inserted in place of the typical drain at the bottom of the water heater. The power to the lower thermostat/element was disconnected and rewired to provide a low voltage signal to the WH/DH when water heating is required.

The return air for the WH/DH was ducted from the return plenum of the heat pump. The supply air was ducted separately from the WH/DH with one duct terminating on the upstairs level of the house, and the other terminating on the lower level. The WH/DH supply air can also be ducted into the supply air plenum of the heat pump as long as the supply fan of the heat pump is wired to operate at the same time as the WH/DH fan in order to prevent recirculation, Figure 2. Controlled fresh-air intake is one difference from the baseline equipment. A constant 76 m^3/h (45 ft^3/m) was provided to the house. Homes with tightly sealed envelopes need mechanical fresh-air ventilation in order to maintain acceptable indoor air quality.

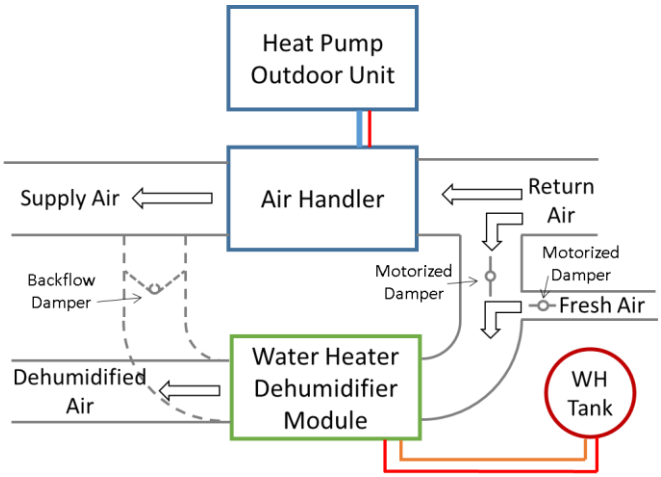


Figure 2: Schematic of equipment layout with dotted lines showing optional supply duct connections with backflow damper

3.3. Instrumentation

The ASHP was instrumented for air-side heating and cooling capacity measurements as well as additional measurements of refrigerant-side pressures and temperatures. The condensate drained from the evaporator coil was also measured to provide a check on the air-side latent capacity measurement. The WH/DH was instrumented for water-side water heating capacity measurements as well as air-side capacity measurements for dehumidification and the cooling by-product from the water heating mode. Like the ASHP, the condensate drained from the WH/DH was also measured to provide a check on the air-side latent capacity. Solid-state W/Wh transducers were used to measure the total and component energy use of the ASHP and WH/DH.

4. Field evaluation results

4.1. Dehumidification Performance

The WH/DH is called to dehumidify when a low voltage alternating current (AC) signal is supplied. In a typical installation, this would be provided by a humidistat. However, since the home was already instrumented with humidity sensors, the data logger was used to provide the contact closure functionality of a humidistat. The call for dehumidification was supplied to the WH/DH when either the level 1 or level 2 humidity sensors read over 55% relative humidity (RH), and was removed when both sensors read below 51% RH. The WH/DH did an excellent job of maintaining the humidity in the home, with the highest hourly average humidity measurement being 54.8% during the study.

One issue observed during the operation of the WH/DH was evaporation of condensate remaining on the evaporator coil during the ventilation mode. Both the dehumidification and the water heating modes condense moisture from the air on the evaporator coil. This phenomenon can be seen in the top plot of Figure 3. The blue highlighted sections indicate the unit operating in dehumidification mode. In this mode it can be seen that the unit is providing positive latent cooling and negative sensible cooling (i.e., heating). The house humidity is

reduced as moisture is removed from the air. The pink highlighted sections indicate operation in the ventilation mode. In this mode the unit is providing negative latent cooling (i.e., evaporating moisture into the air) and sensible cooling due to the evaporative cooling effect. This causes an increase in the house humidity and negates part of the work done during the dehumidification mode. Based on a comparison of the air-side latent capacity during the dehumidification mode and the latent capacity calculated based on the measured condensate leaving the unit, approximately 33% of the condensed moisture was being evaporated during the ventilation mode. This results in an effective dehumidification efficiency that is one third lower than its steady state efficiency. The first attempt to mitigate this effect was to reduce the airflow through the unit during the ventilation mode. The initial equipment setup required the ventilation airflow to be similar to that of the water heating and dehumidification modes, ~115 L/s (300 cfm), in order to ensure the proper ventilation rate of 76 m^3/h (45 ft^3/m). This was due to the small size of the fresh air intake duct relative to the return duct of the WH/DH. In June of 2016, a damper was added to the return duct of the WH/DH upstream of the fresh air intake. This damper was closed during the ventilation mode allowing the airflow to be reduced down to the required ventilation rate since the unit was now pulling 100% fresh ventilation air instead of a mixture of fresh air and house air. The bottom plot in Figure 3 shows dehumidification and ventilation cycles of the WH/DH after the installation of the damper and reduced airflow composed of 100% fresh air for ventilation. During the ventilation mode, the evaporation of condensate is significantly reduced as indicated by the latent capacity being only slightly negative. The frequency of dehumidification cycles is also reduced and the humidity in the home increases at a much slower rate (although the outdoor conditions were slightly drier for the data shown in the bottom plot of Figure 3). Once again a comparison of the air-side latent capacity measured during the dehumidification mode to the condensate collected from the WH/DH indicates that only 5% of the condensed moisture was evaporated back into the air during the ventilation mode. This is a significant reduction when compared to the 33% evaporation rate seen prior to installing the return air damper. Additional modifications to further reduce evaporation that can occur during the ventilation mode are under investigation.

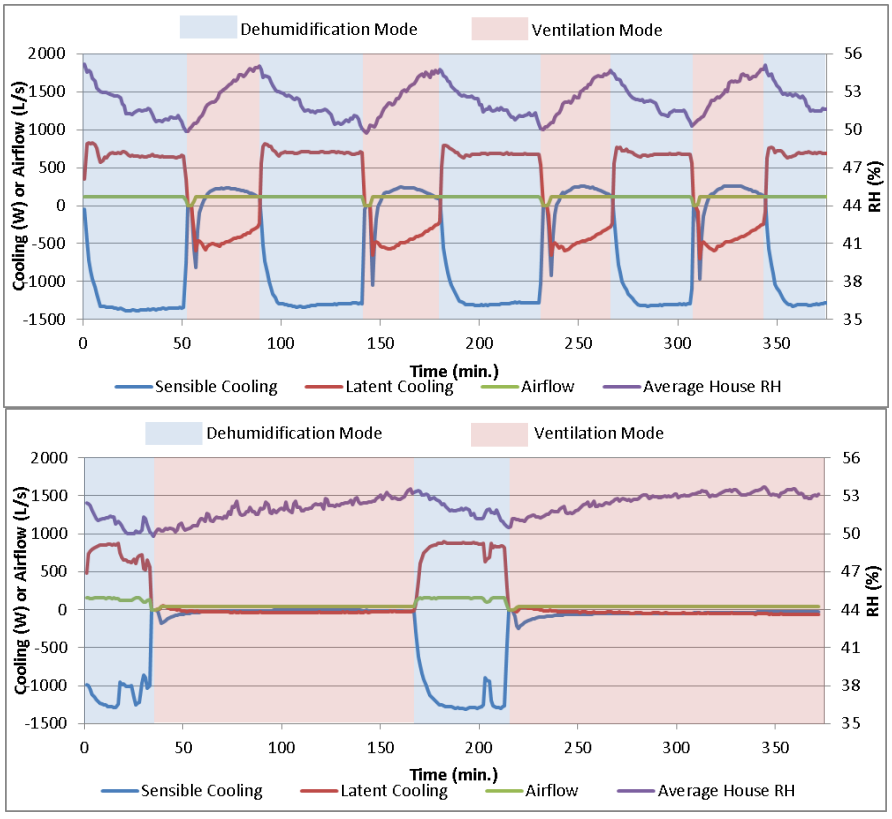


Figure 3: WH/DH cycling between dehumidification mode and ventilation mode with condensate evaporation during ventilation, for equal ventilation and dehumidification air flow rates (top plot) and with reduced airflow during ventilation (bottom plot)

The monthly runtime and average efficiency based on the measured air-side latent capacity are shown in Figure 4. As noted earlier, evaporation of condensate in the ventilation mode likely resulted in increased dehumidification runtime for all months prior to and including June of 2016. However, July and August of 2016 still showed significant, greater than 100 hours per month, dehumidification runtime. For the months with significant runtime, the efficiency for dehumidification ranged from ~1.5-2.1 L/kWh. There were measurement issues related to the air-side capacity of the WH/DH for the months of Aug.-Oct. 2015, so this time period was excluded from the calculation of the average dehumidification efficiency of 1.7 L/kWh.

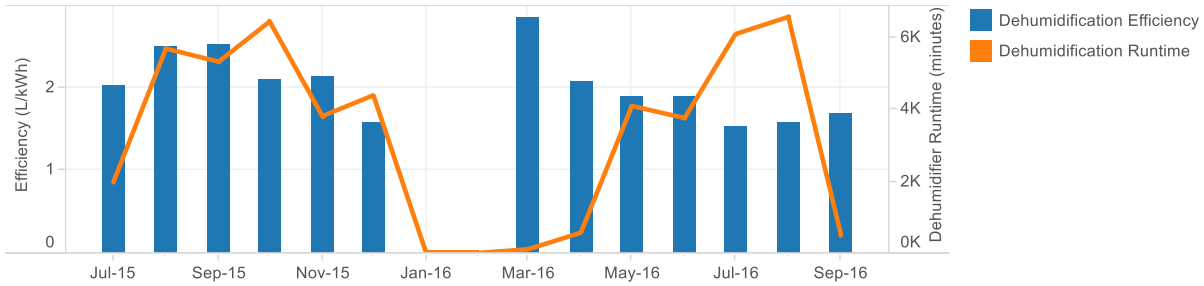


Figure 4: Monthly dehumidification efficiency and runtime

4.2. Water Heating Performance

As noted in section 3.1 the hot water draw schedule varied from day to day based on realistic probability distributions of hot water draws. Figure 5 shows a histogram of the daily hot water use for the period of October 2015 to September 2016. The average daily hot water draw for this period was 209 L/day (55.2 gal/day), slightly below the target of 220 L/day (58.1 gal/day). This difference is due to an issue that occurred with the hot water draw system in January of 2016 resulting in 20 inadvertent days of no hot water use.

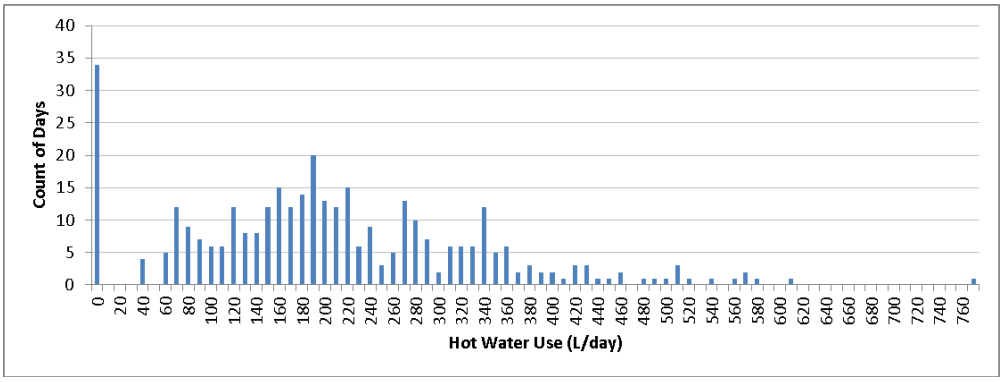


Figure 5: Histogram of daily hot water use at the research house

Two factors that impact the efficiency of a heat pump are the temperature of the heat source and the temperature of the heat sink. For the case of water heating, the heat source is the air entering the WH/DH unit, and the heat sink is the entering water. Figure 6 shows a color map, based on 1 minute resolution data, of the water heating coefficient of performance (COP) for various entering air and entering water temperatures experienced during the field evaluation. As expected, the COP of the system increases with an increase in entering air temperature (heat source) and decreases with an increase in entering water temperature (heat sink) as seen by the color trend of red to blue from the lower right of the map to the upper left. The percentages located within each cell indicate the percentage of the total water heating runtime at the given conditions. Most of the operating time occurs at higher entering water temperatures, 35°C (95°F) and up, indicating that the bottom of the tank is staying warm the majority of the time. During heavy hot water use, the lower portion of the tank can become saturated with cold water resulting in entering water temperatures near the temperature of the cold water supply.

The WH/DH was setup to pull a mixture of house air and fresh air for ventilation during water heater operation. This approach maintains continuous ventilation in the home and provides warmer, more humid return air during summer months. However, when it is cool or cold outside, the fresh air reduces the efficiency of the water heating mode. During extremely cold weather, it is also possible for the evaporator to freeze. This occurred early in the heating season and is represented by the low entering air temperatures and corresponding efficiencies (dark red) in Figure 6. A control change was made during the heating season to close the fresh air damper during water heating when the outdoor temperature was below 15.6°C (60.0°F) in order to maintain higher water heating efficiencies and eliminate the risk of the evaporator freezing.

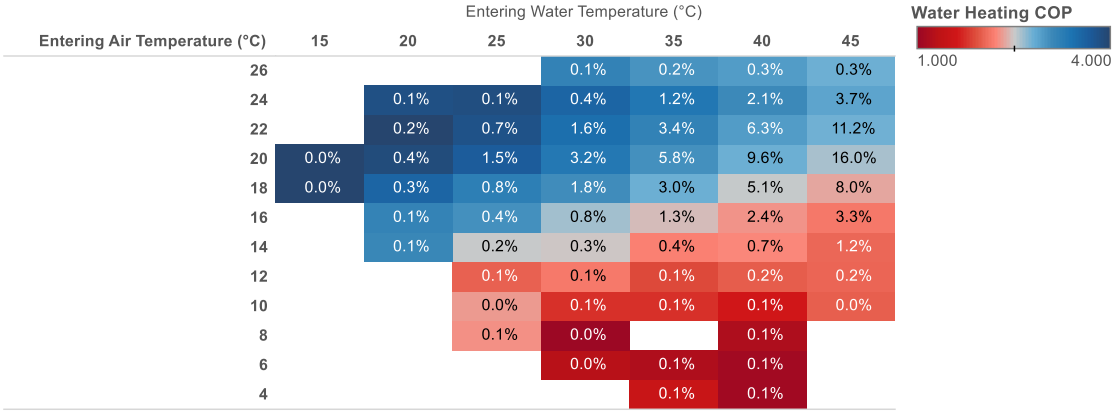


Figure 6: Water heating efficiency color map with percentage of total water heating operating hours labeled for each set of conditions

The monthly water heating efficiencies for the WH/DH are shown in Figure 7. The different lines indicate the performance of the system at various points as heat is generated by the water heating heat pump (HP), transferred to the storage tank, and leaves the storage tank for use in the house. The blue line shows the COP of the HP only, which does not account for backup resistance heat use and losses associated with the interconnecting lines and storage tank. These COPs range from a low of 2.3 in January to a high of 3.1 in August. As mentioned earlier, there were 20 days with no hot water draws in January. Without the flow of cold makeup water into the tank, the entering water temperatures seen by the WH/DH will be at least as high as the lower thermostat “make” temperature of approximately 44.4°C (112°F). The higher entering water temperatures seen by the WH/DH when recovering from standby losses compared to recovering from hot water use resulted in lower than average COPs for the month of January.

The orange line in Figure 7 shows the COP of the WH/DH when accounting for heat loss in the water lines that connect the WH/DH to the storage tank. Immersion temperature sensors located on both ends of the interconnecting water lines allowed for the measurement of the heat loss. Despite the water lines being insulated, the measured heat loss from these lines averaged 9.8% of the heat provided by the heat pump. Examining a one week snapshot of data indicated that that, on average, the water lines lost 76% of their heat relative to the garage temperature between water heating cycles. With a total length of 10.9 m (35.8 ft) of 1.9 cm (0.75 in.) diameter copper pipe and 2007 water heating cycles, the heat loss between water heating cycles of the water in the pipes and the pipes themselves is estimated to be 216 kWh (738 kBtu) or 6.2% of the water heating delivered by the WH/DH. Using the average garage temperature, average water temperature in the lines, and insulation thickness, the heat loss from the lines during the water heating cycle was calculated to be 133 kWh (453 kBtu) or 3.8% of the water heat provided by the WH/DH. Combining the calculated off-cycle and water heating cycle line losses yields a calculated value of 10.0% heat loss, which agrees well with the measured line losses of 9.8%.

The green line in Figure 7 shows the COP of the system when including backup resistance heat use but excluding tank losses. After filtering the data for periods when the WH/DH was shut down for sensor maintenance or other issues, the backup resistance energy use for the year-long period was only 60.2 kWh or 5.4% of the total energy used for water heating.

Finally, the red line in Figure 7 shows the COP of the entire system and was calculated by dividing the measured water heating energy being delivered to the house at the outlet of the storage tank by the total energy use of the WH/DH and backup resistance elements. Based on the measured data, the tank losses are 9.9% of the water heating energy delivered to the tank (omitting data from January when there was inadvertently 20 days of

no hot water use). This value is in line with the performance expectations of a typical electric storage water heater tank having a rated EF of 0.9 (minimum allowable EF for electric storage WHs manufactured before April 2015 in the U.S.).

The annual COPs for the WH/DH were 2.75, 2.48, 2.40 and 2.19 for the HP only, HP with line losses, HP with line losses and backup resistance use, and entire WH/DH system including tank/line losses and backup resistance heat use respectively. In order to achieve the highest overall system water heating efficiencies, it is important to limit the length and, if possible, diameter of the water lines connecting the WH/DH to the storage tank, insulate these lines, and use a well-insulated storage tank.

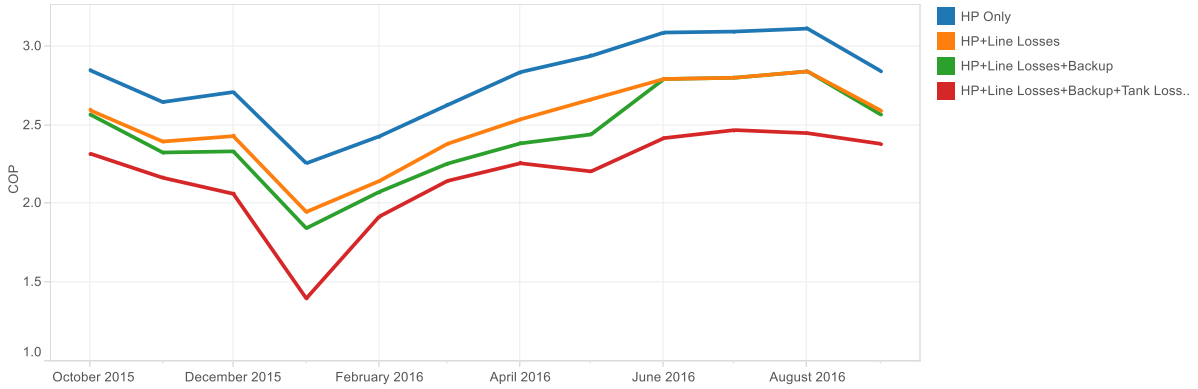


Figure 7: Monthly average water heating COPs of the water heating heat pump with and without backup resistance heat use and heat losses from the storage tank and water lines connecting the WH/DH to the storage tank

4.3. Space Cooling Performance

The monthly and seasonal space cooling performance of the ASHP and the impact of the WH/DH operation on space cooling are summarized in Table 1. The average monthly cooling COPs of the ASHP were between 4.32 and 5.59 with a seasonal average of 4.44. The water heating mode of the WH/DH provides space cooling as a by-product of its operation. The dehumidification mode also generates sensible heating in addition to latent cooling with a net heating effect (indicated by negative values in the table). The "free cooling" effect that the water heating operation has on the overall system (ASHP and WH/DH combined) space cooling efficiency results in monthly cooling COPs for the system between 4.46 and 10.84. The very high system COPs during April and October are indicative of a larger ratio of "free cooling" from the WH/DH to cooling supplied by the ASHP. However, it is likely that much of the cooling delivered by the WH/DH in these months was not satisfying a demand for cooling and was actually over-cooling the house. Because of this, the WH/DH cooling effect for these two months was not included in the seasonal average cooling COP for the system. With this consideration, the seasonal average COP of the system was 4.72, 6.3% higher than the COP of the ASHP alone. For the cooling season, this 6.3% increase in efficiency results in estimated space cooling energy savings due to operation of the WH/DH of 122 kWh.

Table 1: Space cooling data for the ASHP including the cooling and heating by-products of the WH/DH

Month	April 2016	May 2016	June 2016	July 2016	Aug. 2016	Sept. 2016	Oct. 2015	Totals
System SC delivered, kWh	191	674	1819	2317	2304	1812	271	9189[a]
ASHP	122	526	1697	2242	2233	1680	141	8641
WH/DH DH Mode	-4[b]	-30	-23	-56	-50	-5	-23[b]	191
WH/DH WH Mode	73[b]	178	144	132	121	137	153[b]	938
ASHP SC energy use, kWh	22	99	385	517	517	383	25	1948
ASHP Avg. COP	5.57	5.34	4.41	4.34	4.32	4.39	5.59	4.44
System Avg. COP	8.68	6.84	4.72	4.48	4.46	4.77	10.84	4.72[a]

Avg. OD temp, °C	15.7	18.9	25.1	26.4	26.3	23.7	15.1	21.6
while ASHP cooling	25.0	25.3	28.1	28.4	28.0	26.9	23.4	27.6
ASHP Run hours	22.5	95.0	300.5	400.3	409.7	329.3	31.4	1588.7

[a] Total system space cooling delivered and average system COP do not include WH/DH cooling/heating effects for the months of April and October, because the cooling demand was very low for these months and it is likely that the WH/DH operation did not significantly affect the cooling load experienced by the ASHP.

[b] Only includes days of the month when the ASHP was in the cooling mode

4.4. Space Heating Performance

The monthly and seasonal space heating performance of the ASHP and the impact of the WH/DH operation on space heating are summarized in Table 2. As noted in section 4.3, when the WH/DH operates in water heating mode, it provides cooling as a by-product. However, for the heating season, the latent cooling provides no energy benefit or penalty, so the data only shows the sensible cooling. Similarly for the limited runtime in the dehumidification mode, only the sensible heating is accounted for in the table. The ASHP monthly heating COPs are between 2.00 and 3.43. The lowest COPs correspond to months with high backup resistance heat use. For the month of January 2016, backup resistance heat use accounted for approximately one third of the total heating energy use. The average heating COP of the ASHP during the evaluation period was 2.38. When the cooling and heating by-products of the water heating and dehumidification modes of the WH/DH are accounted for, the overall system heating COP is reduced to 2.23, a 6.3% reduction. For the heating season, this 6.3% reduction in overall efficiency results in an estimated space heating penalty due to operation of the WH/DH of 330 kWh.

Table 2: Space heating data for the ASHP including the cooling and heating by products of the WH/DH

Month	Oct. 2015	Nov. 2015	Dec. 2015	Jan. 2016	Feb. 2016	March 2016	April 2016	Totals
Total Sensible Heating delivered, kWh	172	1344	1687	4029	2723	1192	384	11651[a]
ASHP	171	1431	1764	4158	2974	1408	505	12411
WH/DH DH Mode	43[b]	77	77	0	0	1	1[b]	169
WH/DH WH Mode	-42[b]	-164	-154	-129	-251	-216	-122[b]	-1078
Space Heating energy use, kWh								
Total	50	502	677	2076	1289	478	153	5225
backup	0	0	120	684	299	32	1	1136
defrost	0	0	19	54	24	8	0	105
Avg. ASHP COP	3.43	2.85	2.61	2.00	2.31	2.94	3.29	2.38
Avg. System COP	3.44	2.68	2.49	1.94	2.11	2.49	2.51	2.23
Avg. OD Temp, °C	15.1	11.5	10.6	1.5	5.4	12.5	15.7	10.3
While ASHP heating	10.2	5.9	5.3	0.2	2.1	5.8	7.6	3.3
Run hours	27.0	260.2	289.2	592.0	449.0	241.3	87.9	1946.6
Defrost hours	0.0	0.0	4.0	10.7	5.1	2.1	0.2	22.1

[a] Total system space heating delivered and average system COP do not include WH/DH cooling/heating effects for the months of April and October, because the heating demand was very low for these months and it is likely that the WH/DH operation did not significantly affect the heating load experienced by the ASHP.

[b] Only includes days of the month when the ASHP was in the heating mode

5. Predicted energy savings

A bin-hour approach is used to predict the energy savings of the AS-IHP system relative to baseline equipment. In this approach, the hourly heating, sensible cooling, and latent cooling loads delivered by the AS-IHP were averaged within 2.0°C (3.6°F) outdoor air temperature bins. This provides the average house heating

and cooling loads as seen in Figure 8. It should be noted that the heating load line for this house is about 50% higher at -15°C than the heating load line used for $SCOP_h$ ratings calculations. The performance of the baseline equipment (an air-source heat pump with rated performance of 3.8 $SCOP_c$, 2.3 $SCOP_h$, and cooling capacity matching the AS-IHP system for space conditioning, a 0.9 EF water heater, and a 1.4 EF dehumidifier) was calculated using performance curves obtained from BEopt [5] and underlying equations used in EnergyPlus [6]. The performance curves and equations accounted for increased field-installed fan power, frosting and defrosting effects, tempering heat during defrost cycles, cycling losses, and return air conditions matching those of the AS-IHP system. For bins where the baseline heat pump alone could not meet the latent load, the dehumidifier and heat pump ran together to provide the necessary latent capacity and a net sensible capacity of zero. For bins where the baseline heat pump had excess latent capacity, the performance of the heat pump was modeled in two phases. The first phase was operation to meet the latent load on the house at return air conditions matching those of the AS-IHP system. The second phase simulated the baseline heat pump running to meet the remaining sensible load, with the excess latent capacity reducing the return air humidity. The reduction in indoor humidity was correlated to delivered latent cooling by spot checking the relationship during dehumidification cycles of the WH/DH module. The average power of the equipment required to meet house loads was then calculated for each bin. The seasonal energy use and total delivered loads were calculated by multiplying the number of hours in each bin, Figure 9, by the corresponding power and required capacity. For this approach, the latent cooling provided by the WH/DH during the heating season was ignored.

The results of the bin-hour analysis are shown in Table 3. The AS-IHP system shows significant savings in all modes of operation except for space heating (due to the atypical heating load and zone control discussed in the next paragraph). An annual savings of 29%, 3531 kWh, was estimated for the AS-IHP system relative to the baseline equipment for the 2015-2016 evaluation year. This level of savings is similar to those that could be achieved with a high efficiency heat pump, heat pump water heater, whole house dehumidifier, and ventilation system today. However, the AS-IHP system offers several non-energy related benefits including a shared evaporator and compressor for water heating and dehumidification reducing cost and a fully ducted system eliminating localized cooling and noise sometimes associated with heat pump water heaters.

While the zoning system ensured that both levels of the house were maintained at the desired conditions, the method of controlling the ASHP airflow based on the zone(s) calling for conditioning may have reduced the ASHP efficiency. Based on the data, it appears that the ASHP ran at a compressor speed corresponding to the airflow set for the zone(s) calling for space conditioning. For example, when both zones were calling for conditioning, the ASHP ran at high airflow and correspondingly high compressor speeds regardless of the actual load on the house. Therefore, there were times when the ASHP ran at a higher capacity than necessary to meet the load on the house. This reduces the operating time of the ASHP at low capacity where it is most efficient and increases cycling and associated losses. Due to this, the space heating and cooling estimated savings are likely conservative and additional savings are expected of a system not operating with zone controls.

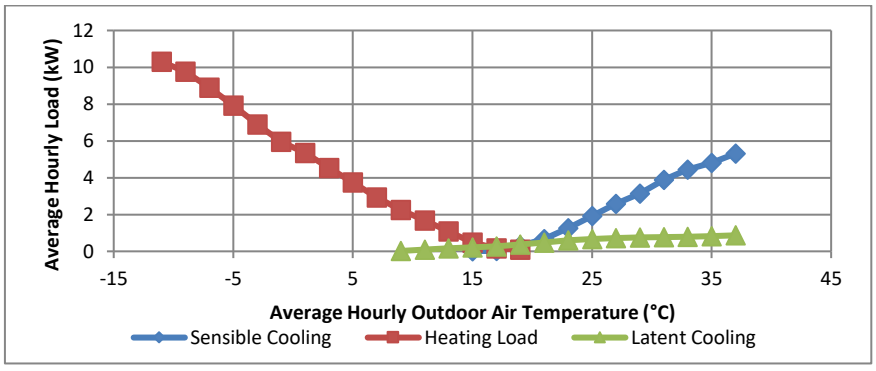


Figure 8: Average hourly heating, sensible cooling, and latent cooling loads of the field test house

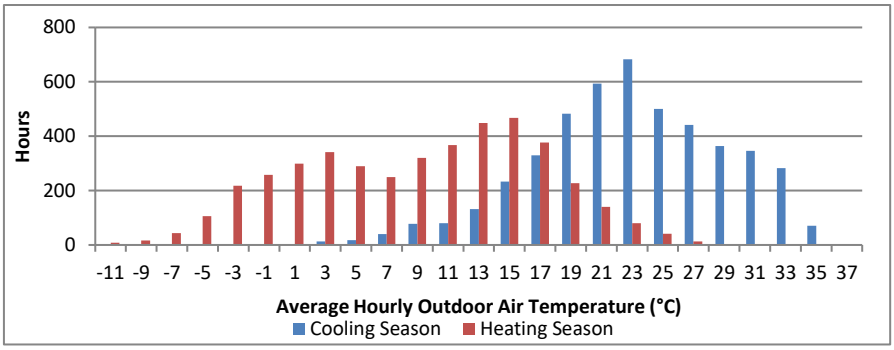


Figure 9: Cooling and heating season hours falling within outdoor air temperature bins

Table 3: Estimated annual energy savings for the AS-IHP relative to baseline equipment

	Baseline	AS-IHP	Savings over baseline
Space heating, kWh	5889	5225	11%
Space cooling and dehumidification, kWh	3475	2201	37%
Water Heating, kWh	2739	1146	58%
TOTAL, kWh	12103	8572	29%

6. Conclusions

A field evaluation of an AS-IHP system was conducted in Knoxville, Tennessee in an unoccupied house with simulated occupancy. The AS-IHP was composed of an ASHP and a prototype WH/DH module providing the home with space cooling, space heating, water heating, dehumidification, and ventilation. A few issues were identified over the course of the field evaluation. During the field evaluation it was observed that evaporation of condensate remaining on the WH/DH evaporator coil was occurring during the ventilation mode and adversely affecting indoor humidity. This was addressed by adding dampers to limit the ventilation air to 100% fresh air and thereby allowing significant reduction of the ventilation airflow and resulting evaporation. There are additional solutions under investigation that could be implemented to address this issue. The heat loss from water lines connecting the WH/DH to the storage tank resulted in ~10% reduced water heating system efficiency. In order to mitigate this effect, care should be taken to locate the WH/DH as close to the storage tank as possible and insulate water lines to the maximum extent reasonably possible. The data also indicates that the space heating and cooling efficiency of the ASHP were reduced due to operation at higher capacities than necessary to meet the load. This was the result of the zoning system controlling the ASHP airflow, and corresponding compressor speed, based on the zone(s) calling for conditioning instead of the heating or cooling load.

The energy consumption of the AS-IHP system was compared to the estimated efficiency of a suite of baseline equipment including an air-source heat pump with rated performance of 3.8 $SCOP_c$ and 2.3 $SCOP_h$, a 0.90 EF resistance water heater, and a 1.4 EF dehumidifier. The annual savings were estimated at 3270 kWh, 29%, for the house and time period studied. Due to the issues mentioned above, these savings are likely conservative. The particular combination of ASHP and WH/DH evaluated in this study would have the highest savings for homes located in cool-humid to hot-humid climates that can take full advantage of the dedicated dehumidification mode and the cooling by-product provided when the WH/DH is heating water.

Acknowledgements

This study was based upon work supported by the U.S. Department of Energy, Office of Energy Efficiency and Renewable Energy (EERE), specifically the Building Technologies Office (BTO). The authors would like to recognize Mr. Antonio Bouza, BTO HVAC, Water Heating, and Appliances Technology Manager, and Mr. Eric Werling, BTO Building America Program Coordinator, for their support of this research. Additionally we would like to recognize Lennox Industries, Inc. that worked with us throughout the development of the system.

References

[1] Rice, C. K., Shen, B., Baxter, V. D., Shrestha, S. S., and Uselton, R. B., 2014. "Development of an Air-Source Heat Pump Integrated with a Water Heating/Dehumidification Module", *Conference Proceedings of 11th International Energy Agency Heat Pump Conference*, Montreal, Quebec, Canada, May 12-16

[2] AHRI. Directory of Certified Product Performance. [Online] Air-Conditioning, Heating, and Refrigeration Institute. [Cited: 9 30, 2016.] https://www.ahridirectory.org/.

[3] Hendron, R., Engebrecht, C. Building America House Simulation Protocols. Golden, CO : National Renewable Energy Laboratory, 2010. TP-550-49426.

[4] Hendron, R., Burch, J., Barker, G. Tool for Generating Realistic Residential Hot Water Event Schedules: Preprint. United States : s.n., 2010. NREL/CP-550-47685.

[5] NREL. BEopt. Golden, CO : National Renewable Energy Laboratory. version 2.4.0.1.

[6] U.S. Department of Energy. EnergyPlus Version 8.6 Documentation: Engineering Reference. September 30, 2016. https://energyplus.net/sites/all/modules/custom/nrel_custom/pdfs/pdfs_v8.6.0/EngineeringReference.pdf.