ORNL/TM-2015/525
CRADA/NFE-11-03561

# Advanced Variable Speed Air-Source Integrated Heat Pump (AS-IHP) Development – Final Report


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**September 30, 2015**

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MANAGED BY UT-BATTELLE FOR THE US DEPARTMENT OF ENERGY

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Energy and Transportation Science Division

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Date Published: September 30, 2015

Prepared by
OAK RIDGE NATIONAL LABORATORY
Oak Ridge, Tennessee 37831-6283
managed by
UT-BATTELLE, LLC
for the
US DEPARTMENT OF ENERGY
under contract DE-AC05-00OR22725

**CRADA NFE-11-03561**
**with**
**Nordyne, LLC**

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## Executive Summary

Between August 2011 and September 2015, Oak Ridge National Laboratory (ORNL) and Nordyne, LLC (now Nortek Global HVAC LLC, NGHVAC) engaged in a Cooperative Research and Development Agreement (CRADA) to develop an air-source integrated heat pump (AS-IHP) system for the US residential market. Two generations of laboratory prototype systems were designed, fabricated, and lab-tested during 2011-2013. Performance maps for the system were developed using the latest research version of the DOE/ORNL Heat Pump Design Model, or HPDM, (Rice 1991; Rice and Jackson 2005; Shen et al 2012) as calibrated against the lab test data. These maps were the input to the TRNSYS (SOLAR Energy Laboratory, et al, 2010) system to predict annual performance relative to a baseline suite of equipment meeting minimum efficiency standards in effect in 2006 (combination of 13 SEER air-source heat pump (ASHP) and resistance water heater with Energy Factor (EF) of 0.9). Predicted total annual energy savings, while providing space conditioning and water heating for a tight, well insulated 2600 $ft^2$ (242 $m^2$) house at 5 U.S. locations, ranged from 46 to 61%, averaging 52%, relative to the baseline system (lowest savings at the cold-climate Chicago location). Predicted energy use for water heating was reduced 62 to 76% relative to resistance WH.

Based on these lab prototype test and analyses results a field test prototype was designed and fabricated by NGHVAC. The unit was installed in a 2400 $ft^2$ (223 $m^2$) research house in Knoxville, TN and field tested from May 2014 to April 2015. Average overall cooling season efficiencies (with the water heating set point at 120°F) were 5.14 for space cooling (SC), 4.39 for water heating (WH, neglecting a small amount of backup element usage due to data monitoring and system control issues), and 5.03 for the overall average. The SC COP equates to an average site-measured SEER of about 17.5 Btu/Wh. Average overall heating season efficiencies were 2.06 for space heating (SH), 2.16 for WH, and 2.07 for the overall average. The SH COP equates to an average site-measured HSPF of about 7.01 Btu/Wh. It must be noted that the thermal envelope of the test house is less efficient than that of the house used for the analytical predictions. This contributed to a field SH load that was considerably higher than those predicted for all of the five analysis locations except Chicago.

Based on the demonstrated field performance of the AS-IHP prototype and estimated performance of a baseline system operating under the same loads and weather conditions,

it was estimated that the prototype would achieve ~40% energy savings relative to the minimum efficiency suite. The estimated WH savings were >60% and SC mode savings were >50%. But estimated SH savings were only about 20%. It is projected that had the test house been better insulated (more like the house used for the savings predictions noted above) and the IHP system nominal capacity been a bit lower that the energy savings estimate would have been closer to 45% or more (similar to the analytical prediction for the cold climate location of Chicago).

The major items impacting the field measured SH energy use for the prototype AS-IHP were 1) heavy reliance on back up electric elements for SH and defrost tempering, and 2) higher indoor blower and water pump energy usage as compared to lab measured blower and simulated pump performance. In addition the daily hot water (HW) usage for the field test averaged 56.3 gallons/day – about 13%, lower than that assumed for the analytical performance simulations (64.5 gal/d). The lower HW use coupled with the higher space conditioning loads due to the poorer thermal envelope of the test house cause the WH load during the test year to be a smaller fraction of the total load on the IHP. This reduced the weighting of the WH mode energy savings with concomitant negative impact on the total annual energy savings estimate.

Nortek Global HVAC LLC is actively pursuing plans to introduce a heat pump product that implements the features of the AS-IHP field test prototype. The major development need remaining is to finalize the controls and control software and convert the field prototype control system to a solid-state hardware design (pc board, etc.) more suitable for production line use. They are assessing AS-IHP product introduction against other new product priorities.

The demonstrated cooling season performance results of the residential prototype under this CRADA indicate that application to commercial buildings with thermal loads dominated by WH and SC needs (e.g., restaurants, commercial laundries, health/fitness centers, lodging facilities, etc.) would have higher annual energy savings potential. A follow on CRADA project is planned with a goal to develop and field test a prototype AS-IHP aimed at such commercial building applications. It is planned to start work on this effort in FY2016/17.

# 1. Introduction

In late FY2011, a Cooperative Research and Development Agreement (CRADA) between UT-Battelle, LLC (ORNL) and Nordyne, LLC (now Nortek Global HVAC LLC, NGHVAC) was initiated to conduct the research and development needed to support development of a new residential heating, ventilating and air-conditioning (HVAC) & water heating (WH) product – an air-source integrated heat pump (AS-IHP). The goal was to introduce a new, highly efficient class of products for providing energy services (e.g. space heating and cooling, water heating, and indoor humidity control) to residential and small commercial buildings while consuming ~50% less energy than current minimum efficiency equipment.

The Department of Energy's (DOE) Building Technologies Program (DOE-BT) has a long term goal to maximize the energy efficiency of the US building stock by year 2020. To achieve this vision, a deep reduction of the energy used by the energy service equipment (equipment providing space heating and cooling, water heating, etc.) is required - 50% compared to today's best common practice. One approach to achieving this is to produce a single piece of equipment that provides multiple services. In FY05-07 ORNL developed a general concept for such an appliance, called the integrated heat pump (IHP) [Murphy, et al 2007]. Successful achievement of its goal requires that DOE not only develop the IHP concept, but must facilitate introduction of such equipment to the US building market. For this activity to have the best chance of success, collaboration with manufacturing partners with experience in developing and marketing HVAC products is critically required. NGHVAC expressed interest in the AS-IHP concept and agreed to partner with ORNL in this CRADA.

Project tasks were undertaken to design several system prototypes, produce lab test systems, refine the design and produce a prototype for field testing.

## 2. Background – AS-IHP Concept Development

Full details of the AS-IHP concept development can be found in the report by Murphy, et al (2007) and are briefly summarized here to provide a context for the subsequent system development activities under the CRADA. This system concept (Figure 1, conceptual installation; Figure 2, schematic) uses one variable-speed (VS) modulating compressor, a VS indoor blower and outdoor fan, and a multi-speed pump for hot water circulation. A 50 gallon (~189 l) WH tank is included. This original concept included a dedicated dehumidification mode and a humidifier option. The concept analyses were based on a relatively small (1800 $ft^2$, 167 $m^2$) and very well insulated house with nominal space cooling design loads of 1-1.5 tons (3.5-5.3 kW) depending upon location (e.g., insulation and space heating (SH)/space cooling (SC) load levels needed to reach net zero energy home, nZEH, performance). The NGHVAC system is of a 3-ton (~10.5 kW) nominal size designed for somewhat larger residences typical of new construction practice. For such homes, the fraction of the total load due to WH is reduced some from the original concept.

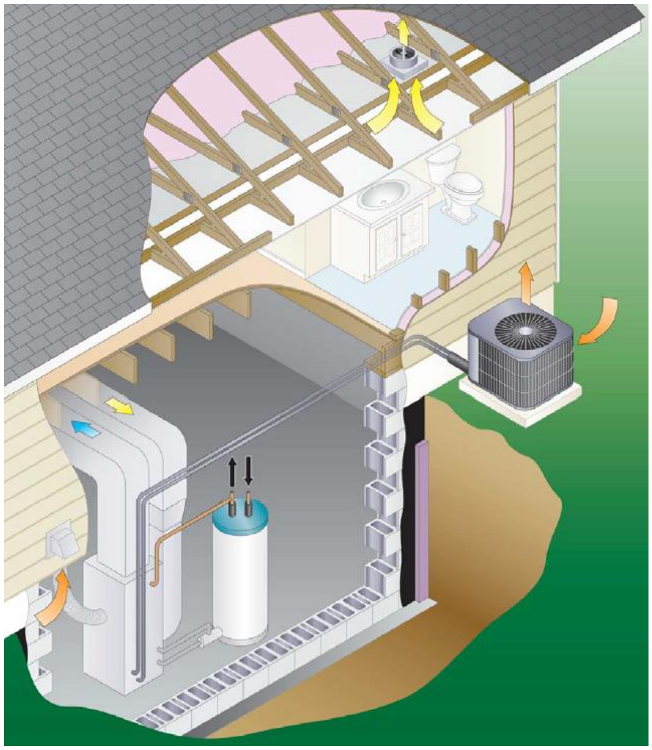

**Fig. 1. Conceptual installation of the residential air-source integrated heat pump.**

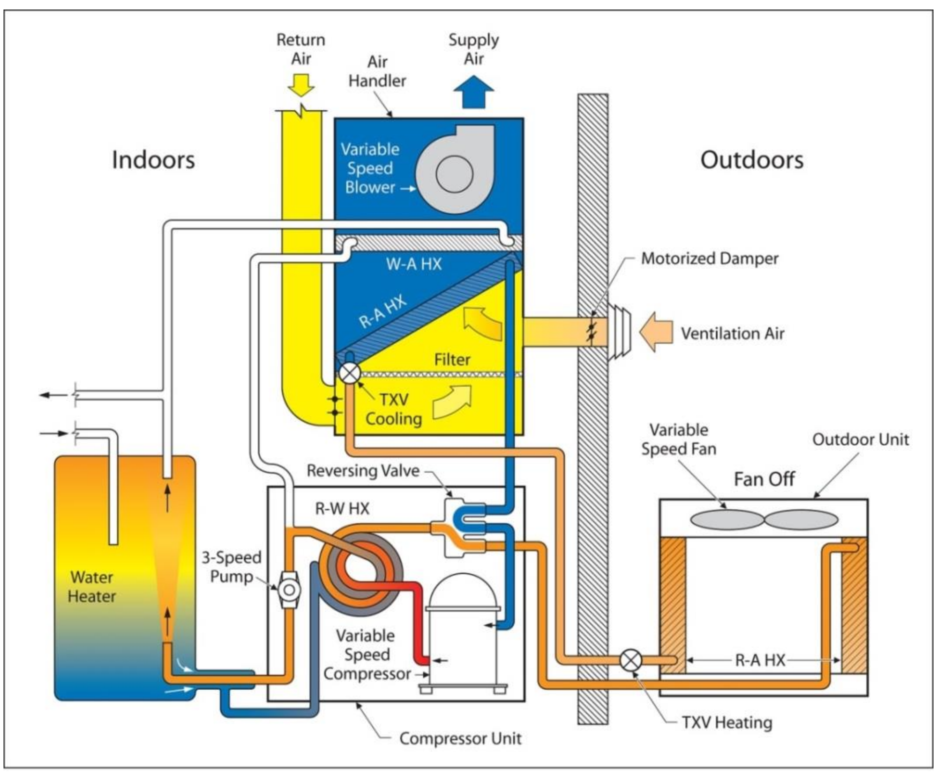


**Fig. 2. AS-IHP system schematic; SC plus "on-demand" WH mode shown.**

Annual energy use simulations for a baseline suite of individual systems (13 SEER/7.7 HSPF heat pump, 0.90 EF electric WH, standalone dehumidifier representative of average units available in 2006, the humidifier option, and ventilation per ASHRAE standard 62.2 (ASHRAE 2007) requirements) and the AS-IHP were performed using the TRNSYS 16 platform (Solar Energy Laboratory, et al. 2010). Annual, sub hourly simulations were performed for the baseline system and the IHP for five locations (Atlanta, mixed-humid type climate; Houston, hot-humid; Phoenix, hot-dry; San Francisco, marine; and Chicago, cold). Simulating the IHP systems required that the ORNL heat pump design model (HPDM) (Rice and Jackson 2005) be utilized to develop detailed performance maps for each operating mode which were then input to TRNSYS. Set points for space heating and cooling were 71°F and 76°F (21.7°C and 24.4°C), respectively. The water heating set point was 120°F (48.9°C) and total daily hot water use of ~64.5 gallons (~245 L) was assumed using the schedule shown in Table 1. The systems' humidity control set points (dehumidifier and humidifier for the baseline; dedicated dehumidification mode and humidifier for the IHP) were set to maintain indoor relative humidity (RH) ≤60% in summer, fall, and spring; and ≥30% in winter.

**Table 1. Daily hot water draw schedule assumed for analysis**

| Event | Start time (h) | Duration (min) | Fraction of daily consumption |
|---|---|---|---|
| Shower | a.m. 6:00 | 12 | 0.172 |
| Shower | 6:15 | 12 | 0.172 |
| Shower | 6:30 | 12 | 0.172 |
| Lavatory | 6:00 | 1 | 0.014 |
| Lavatory | 6:15 | 1 | 0.014 |
| Kitchen sink | 6:45 | 2 | 0.029 |
| Kitchen sink | 7:30 | 2 | 0.029 |
| Clothes wash cycle | 9:00 | 3 | 0.204 |
| Lavatory | p.m. 12:15 | 1 | 0.014 |
| Kitchen sink | 12:30 | 1 | 0.014 |
| Lavatory | 4:45 | 1 | 0.014 |
| Lavatory | 5:15 | 1 | 0.014 |
| Dishwasher (1st wash) | 7:30 | 1.5 | 0.048 |
| Dishwasher (2nd wash) | 8:00 | 1.5 | 0.048 |
| Lavatory | 9:45 | 1 | 0.014 |
| Lavatory | 10:15 | 1 | 0.014 |
| Lavatory | 10:30 | 1 | 0.014 |

Table 2 shows the annual loads for a 167 $m^2$ (1800 $ft^2$) very well insulated house (nZEH ready) obtained from the TRNSYS simulations reported by Murphy et al (2007) for the five US climate locations. Table 2 also shows the nominal design SC capacity necessary for each city, and the fraction of the total IHP system load (neglecting the demand dehumidification (DH) loads) due to WH.

**Table 2. Annual SH, SC, WH and demand DH loads for a 167 $m^2$ (1800 $ft^2$) very well insulated house in five US locations**

| Location | Space heating load, kWh | Space cooling load, kWh | Water heating load, kWh (% of total SH+SC+WH load) | Demand dehumidification load, kWh | Heat pump design SC capacity, kW (tons) |
|---|---|---|---|---|---|
| Atlanta | 4775 | 5735 | 3032 (22) | 158 | 4.40 (1.25) |
| Houston | 1766 | 9927 | 2505 (18) | 704 | 4.40 (1.25) |
| Phoenix | 1580 | 9759 | 2189 (16) | - | 5.28 (1.50) |
| San Francisco | 2881 | 88 | 3387 (53) | 42 | 3.52 (1.00) |
| Chicago | 11475 | 2550 | 3807 (21) | 94 | 4.40 (1.25) |

Table 3 provides summary results from annual performance simulations for the baseline HVAC system for the five locations. Table 4 provides the annual results for the AS-IHP including hourly integrated peak demand. For both systems, maximum peaks generally occurred in the winter. Summer peaks are somewhat lower and generally occurred in July or August. Detailed results from the simulations are given in Table 5. The total energy consumption and consumption by individual modes for the baseline system are from the TRNSYS simulations. For the AS-IHP the total energy consumption, that of the ventilation fan, and for the electric backup water heating and space heating are from the detailed TRNSYS simulations. Breakdowns for the other modes for the AS-IHP were taken from the hourly simulations as well but with adjustments to fairly charge the water pump power in combined modes to the water heating function.

**Table 3. Annual site HVAC/WH system energy use and hourly peak demand for a 167 $m^2$ (1800 $ft^2$) very well insulated house with Baseline HVAC/WH system**

| Location | Heat pump cooling capacity kW (tons) | Site energy use, kWh | Hourly peak kW demand (W/S/SA)* |
|---|---|---|---|
| Atlanta | 4.40 (1.25) | 7657 | 8.6/4.6/2.1 |
| Houston | 4.40 (1.25) | 8349 | 6.1/4.4/2.2 |
| Phoenix | 5.28 (1.50) | 7165 | 6.1/3.9/2.1 |
| San Francisco | 3.52 (1.00) | 4937 | 5.7/5.6/1.6 |
| Chicago | 4.40 (1.25) | 10726 | 9.7/6.1/2.4 |

* W – winter morning; S – summer maximum; SA – summer mid-afternoon.

**Table 4. Estimated annual site HVAC/WH system energy use and hourly peak demand with AS-IHP system (winter humidification active)**

| Location | Heat pump cooling capacity (tons) | Site energy use, kWh | Hourly peak kW demand (W/S/SA)* | % energy savings vs. Baseline HVAC |
|---|---|---|---|---|
| Atlanta | 4.40 (1.25) | 3349 | 2.2/1.5/1.2 | 53.7 |
| Houston | 4.40 (1.25) | 3418 | 1.9/1.1/1.1 | 53.7 |
| Phoenix | 5.28 (1.50) | 3361 | 2.1/1.7/1.7 | 48.4 |
| San Francisco | 3.52 (1.00) | 1629 | 1.8/1.6/0.8 | 67.2 |
| Chicago | 4.40 (1.25) | 5865 | 7.3/1.6/1.0 | 45.6 |

* W – winter morning; S – summer maximum; SA – summer mid-afternoon.

The results summarized in Tables 4 and 5 show that the AS-IHP exceeded 50% savings over the baseline system in three of the locations (almost reaching 70% in the mild San Francisco climate). The summer cooling performance of the concept system design at extreme hot outdoor conditions seen in Phoenix is not quite high enough to enable reaching 50% annual savings in this SC dominated climate. In Chicago the energy service loads are dominated by heating — SH and WH together constitute ~84% of the total load — and the AS-IHP heating performance suffers during the extremely cold temperatures encountered in this climate.

Winter peak demand ranged from about 25 to 75% lower for the AS-IHPs than for the baseline. Maximum summer peaks usually occurred in the morning (during peak domestic hot water (DHW) demand periods) and were about 55% to 75% lower vs. the baseline. Summer mid-afternoon peaks were ~20 to 60% lower than those of the base system, depending upon location.

**Table 5. Detailed AS-IHP performance vs. baseline system**

| Loads (1800-ft$^2$ highly efficient house from TRNSYS) | | Equipment | | |
|---|---|---|---|---|
| | | Baseline | AS-IHP | |
| **Source** | **kWh** | **Energy use, kWh ($I^2r$)** | **Energy use, kWh ($I^2r$)** | **Energy reduction compared to baseline** |
| **Atlanta** | | | | |
| **Space Heating** | 4775 | 1789 (51) | 1251 | 30.1% |
| **Space Cooling** | 5735 | 1643 | 1073 | 34.7% |
| **Water Heating** | 3032 | 3402 | 924 (142) | 72.8% |
| **Dedicated DH** | 158 | 208 | 82 | 60.4% |
| **Ventilation fan** | - | 189 | 20 | 89.6% |
| **Totals** | 13701 | 7230 | 3349 | **53.7%** |
| **Humidifier water use** | 499 kg | | 618 kg | |
| **Houston** | | | | |
| **Space Heating** | 1766 | 648 | 474 | 26.9% |
| **Space Cooling** | 9927 | 2853 | 1894 | 33.6% |
| **Water Heating** | 2505 | 2816 | 556 (91) | 80.2% |
| **Dedicated DH** | 704 | 875 | 482 | 44.9% |
| **Ventilation fan** | - | 189 | 12 | 93.7% |
| **Totals** | 14902 | 7380 | 3418 | **53.7%** |
| **Humidifier water use** | 75 kg | | 87 kg | |
| **Phoenix** | | | | |
| **Space Heating** | 1580 | 535 | 336 | 37.1% |
| **Space Cooling** | 9759 | 3317 | 2296 | 30.8% |
| **Water Heating** | 2189 | 2477 | 696 (19) | 71.9% |
| **Dedicated DH** | - | - | - | na |
| **Ventilation fan** | - | 189 | 33 | 82.7% |
| **Totals** | 13527 | 6518 | 3361 | **48.4%** |
| **Humidifier water use** | 170 kg | | 229 kg | |
| **San Francisco** | | | | |
| **Space Heating** | 2881 | 932 | 607 | 34.8% |
| **Space Cooling** | 88 | 26 | 23 | 12.5% |
| **Water Heating** | 3387 | 3767 | 957 (100) | 74.6% |

| | | | | |
|---|---|---|---|---|
| **Dedicated DH** | 42 | 54 | 11 | 80.3% |
| **Ventilation fan** | - | 189 | 32 | 83.2% |
| **Totals** | 6398 | 4968 | 1629 | **67.2%** |
| **Humidifier water use** | 34 kg | | 38 kg | |
| **Chicago** | | | | |
| **Space Heating** | 11425 | 5448 (1415) | 3686 (614) | 32.3% |
| **Space Cooling** | 2550 | 729 | 436 | 40.1% |
| **Water Heating** | 3807 | 4286 | 1644 (327) | 61.6% |
| **Dedicated DH** | 94 | 121 | 83 | 31.9% |
| **Ventilation fan** | - | 189 | 17 | 91.1% |
| **Totals** | 17877 | 10773 | 5865 | **45.6%** |
| **Humidifier water use** | 1369 kg | | 1639 kg | |

# 3. AS-IHP Prototype Equipment Design and Simulation Approach

The AS-IHP concept investigation summarized above led to collaboration with Nordyne, LLC (now NGHVAC), to develop a design suitable for residential applications typical of current construction practices using R-410A refrigerant. NGHVAC selected a nominal 10.6 kW (3-ton) design cooling size for development leading to the first lab prototype testing. The design used inverter-driven variable-speed brushless permanent magnet (BPM) rotary compressor, blower, and fan motors. Dual electronic expansion valves (EEVs) were used to provide a wide range of refrigerant flow control. A nominal 10.6 kW (3-ton) double-walled fluted tube-in-tube heat exchanger (HX) was used for the domestic hot water with tube-and-fin HXs for the indoor and outdoor coils of this initial prototype. One consequence of the larger unit size is that the fraction of the total load due to WH is reduced some from the original concept and so the potential percentage energy savings is expected to be lower.

Expected WH modes of operation are 1) dedicated WH (with the full condensing (FC) output of the IHP going to WH) using the outdoor coil as the heat source, 2) combined SC and WH (FC output), and 3) desuperheating (DS) WH along with SC or SH operation. The water-to-refrigerant HX is arranged in series with the air-to-refrigerant condenser in DS mode and in parallel in FC mode. A pump capable of at least two-speed operation is required to meet both FC and DS flow requirements.

The key design issues were to determine the optimal component operating speeds, flow controls, and refrigerant charge for the various operating modes. The compressor has ~20% over-speed capability in the space heating mode relative to the nominal cooling capacity speed. By using over-speed compressor operation in the heating mode, as proposed by Rice (1992), the heating balance point can be further reduced to minimize the need for supplemental resistance heat. The design must also keep the refrigerant operating conditions within the compressor manufacturer's allowable operating envelope of suction and discharge pressures and temperatures -- limits which vary to some degree with operating speed. An example of the condensing pressure operating limit with rotary compressors is shown in Figure 3.

One technical challenge for AS-IHP system designs is refrigerant charge management. This challenge is greater for air-source systems than for ground-source units because outdoor air coils have much larger internal volume than water-to-refrigerant HXs of similar capacity. When in combined space cooling and water heating mode, the condenser internal volume is somewhat less than in the space cooling mode. To deal with this AS-IHP issue, the manufacturer developed a proprietary design to manage charge between operating modes.

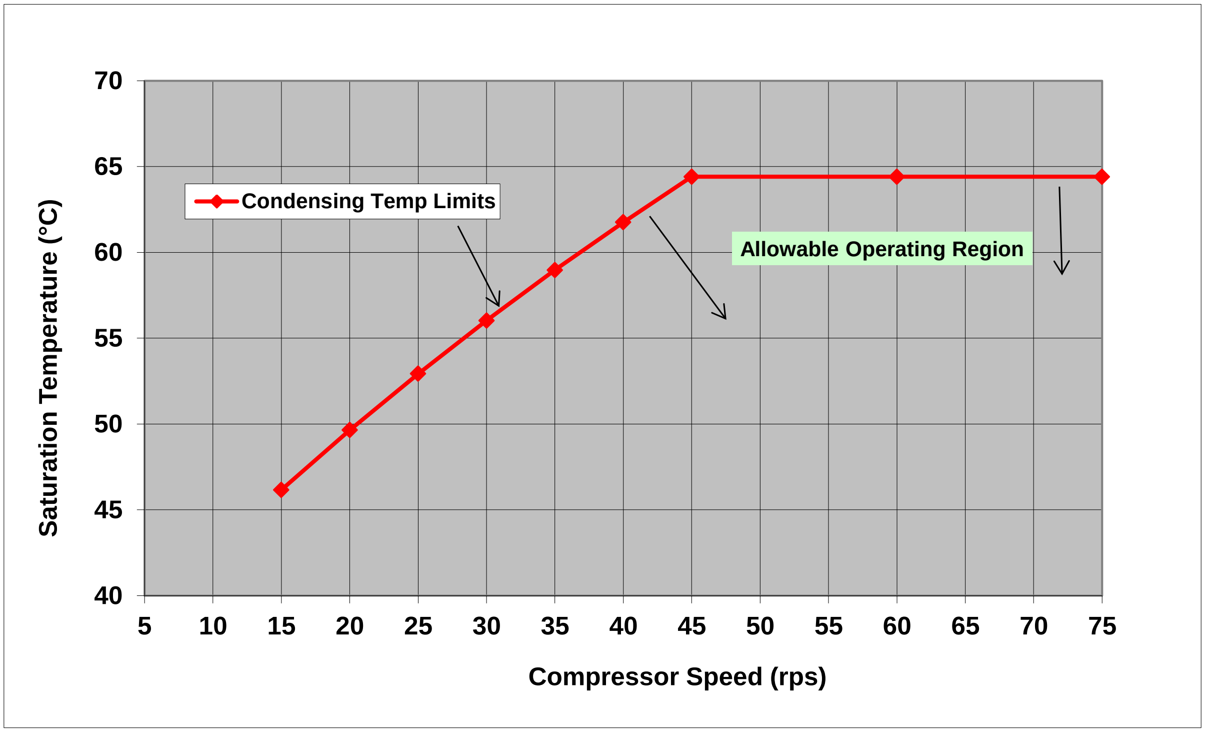


**Fig. 3. Condensing Temperature Limits vs. Compressor Speed for an example Rotary Compressor**

Another design challenge is in water heating. Variable-speed compressors typically can operate at maximum condensing temperatures only above a certain speed, with limits on condensing temperature dropping linearly below this speed. This constraint limits the minimum compressor speed for dedicated water heating operation. In addition, to reach maximum output water temperatures above about 50°C (122°F), higher speeds with output capacity of 10.6 kW (3 tons) or higher are required. As such, a pump capable of providing ~1.14 $m^3/h$ (5 gpm) or higher flow is required. Operation in desuperheating-only mode can also provide temperatures above 50°C (122°F).

## 4. Annual Energy Use Analysis and Savings Predictions for First Prototype Design

The first lab prototype design was assembled by the manufacturer and operationally tested in their laboratory at nominal conditions in each operating mode. This unit shown in Figure 4 was then tested at ORNL in a two-room environmental chamber over a range of steady-state air-source conditions in each of the operating modes.

We used the detailed lab measurements of refrigerant and source/sink conditions to calibrate the HPDM in each of four operating modes: SH, SC, SC+WH, and dedicated WH. The fluted-tube water-to-refrigerant component model in the HPDM (Rousseau 2003) requires internal geometry specifications which were obtained by direct measurements of a cutaway section as shown in Figure 5. We first obtained the refrigerant-side volume and other volume-related geometry information by successively filling the inner tube and annulus with water and comparing the weight of the assembly with that of an empty HX. Geometry details of the air-to-refrigerant HXs and compressor, blower, and pump performance maps were provided by the manufacturers.

**Fig. 4. First Lab Prototype AS-IHP System; (l to r) Water Heating Section with Tank, Indoor Blower and Coil Section, Compressor Section, and Outdoor Fan and Coil Section**

**Fig. 5. Fluted Tube-in-Tube Water-to-Refrigerant HX**

A software wrapper was developed to provide seamless coupling between a publicly available optimization program, GenOpt® (Wetter, 2009), and the HPDM. The GenOpt® wrapper program accepts objectives for optimizing, targeting, and bounding. It can be a flexible and powerful tool for model calibration, control strategy determination, and product configuration optimization. Also, it considers design constraints by setting bounding objectives. Furthermore, it facilitates parametric optimization runs over an extensive range, and helps achieve optimized design over an entire operation envelope. Manual calibration can be a time-consuming and error-prone practice. The GenOpt® wrapper provides an auto-calibration means by selecting targeting objectives. The auto-calibration function makes it possible to calibrate a system model against experimental data over a large range. With this approach, one can apply functional calibration curves to improve the model accuracy for wider ranges of operating conditions.

The HPDM was used with GenOpt® in this manner with the lab test data to auto-calibrate available HX adjustment factors as linear or quadratic functions of compressor speed and/or source/sink temperatures for best match to measured suction and discharge pressures. The test data were also used to determine compressor map power and mass flow corrections, compressor shell heat loss factors, line heat gains/losses and suction superheat levels as similar functions of compressor speed and/or other operating conditions, as well as the indicated active refrigerant charge in each mode. Examples of the heat transfer multipliers obtained from model calibration in combined space cooling and water heating mode (SC+WH) are shown in Figure 6. The evaporator multipliers are usually less than 1 due to airflow mal-distribution while the condenser multipliers are usually greater than 1 for the fluted tube HXs due to the simplified model of the annular refrigerant-side heat transfer. Differences between the calibrated model and the lab data in capacity and compressor–only COP for the dedicated WH mode averaged 1.3% with standard deviations of 3.0 and 4.6%, respectively.

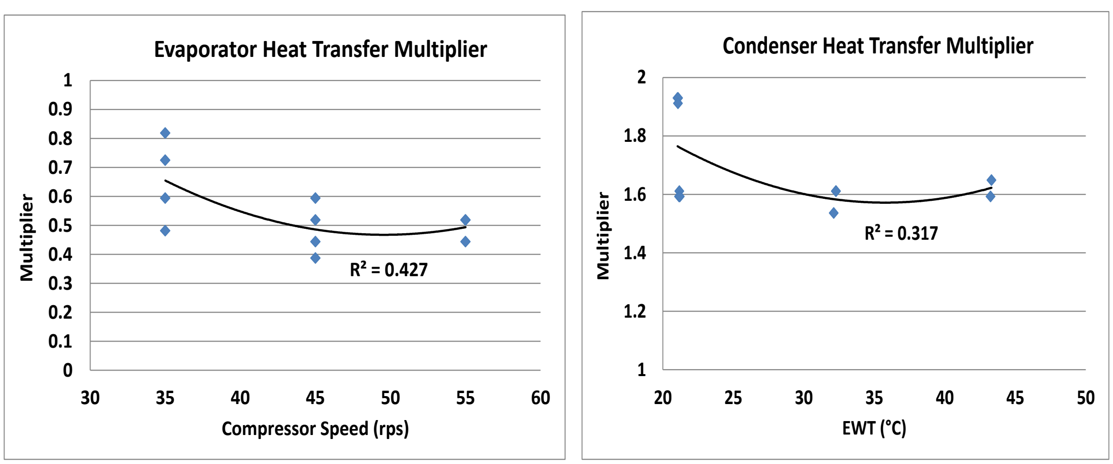


**Fig. 6. Example Calibration Equations for Heat Transfer Multipliers for SC+WH Mode**

Power versus airflow relationships were developed for the indoor blower and outdoor fan from test data. The HPDM was again used with GenOpt® to optimize airflow rates for maximum performance, within minimum allowable delivered air temperature constraints, over the range of appropriate compressor speeds and associated source/sink temperatures in each operating mode. One design control feature specific to an IHP in combined SC and WH mode is that, at entering water temperatures (EWTs) above 35°C (95°F) the indoor airflow needs to be lowered relative to that for SC only to maintain an acceptable sensible heat ratio. An example of the required airflow reduction is shown in Figure 7, for EWTs of 45 and 55°C (113 and 131°F) over a range of compressor speeds. Also, in combined SH and desuperheating WH mode, the indoor airflow needs to be lowered compared to that for the SH only mode to maintain acceptable supply air temperatures.

This information was applied by the manufacturer in developing suitable unit control tables for the four operating modes based on the unit inlet source and sink temperatures and thermostat calls.

Once the design control approaches and calibration equations were complete, we used the HPDM to generate performance maps (i.e., tables) of capacities, powers, and mass flow rates for each mode as a function of all relevant independent variables, e.g., compressor speed, indoor and outdoor DB, indoor or outdoor RH, and EWT from the DHW loop. The desuperheating operation mode was modeled in TRNSYS as a fixed HX effectiveness based on our laboratory test data.

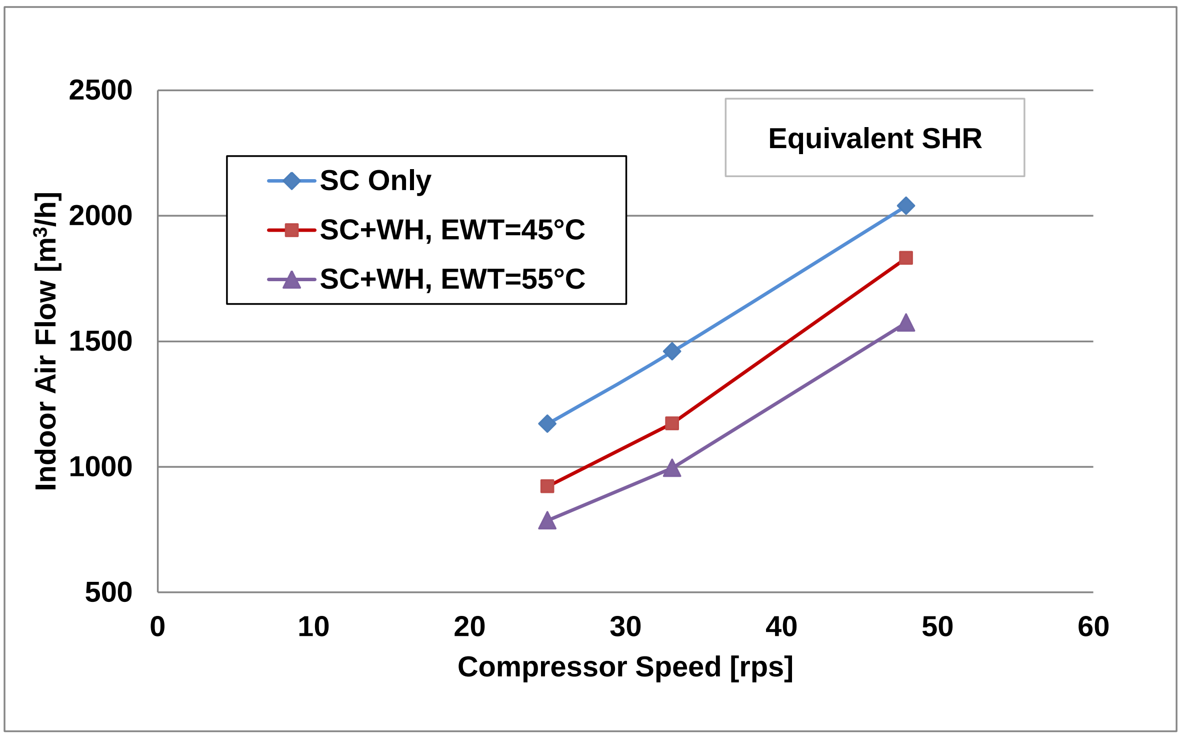


**Fig. 7. Required Indoor Airflows to Maintain Similar SHR Levels in SC and SC+WH Modes**

The HPDM performance maps were used as input to the TRNSYS model for sub-hourly annual AS-IHP performance simulation using a custom interface and thermostat control logic and linked with standard TRNSYS house, weather, and DHW tank models. The available house for the analysis was a tight, well-insulated 242 $m^2$ (2600 $ft^2$) threebedroom unit with 7 kW (~2-ton) design cooling load; as such we scaled the performance maps from 10.6 to 7 kW (3- to 2-tons) nominal capacity.

The DHW tank was a nominal 189 L (50 gallon) capacity. The DHW tank was modeled using a TRNSYS Type 534 module, which models a vertical cylindrical water tank. The tank is divided into 6 isothermal temperature nodes (to model stratification observed in storage tanks) where each constant-volume node is assumed to be isothermal and interacts thermally with the nodes above and below through several mechanisms; heat conduction between nodes and through fluid movement (either forced movement from inlet flow streams or natural destratification mixing due to temperature inversions in the tank). Mechanical ventilation per ASHRAE STD 62.2 (2007) was assumed to be provided by continuous operation of a bathroom ventilation fan.

DHW controls for heat pump dedicated WH operation in the analysis were set to operate until the lower tank temperature was 50°C (122°F) and the upper electric element was set to minimize electric element use while maintaining the upper tank delivery temperature above 41°C (105°F). The assumed daily use schedule shown in Figure 8 includes discrete tempered [41°C (105°F)] and untempered hot water draws totaling ~245 L/day (~64.5 gal/day), which is consistent with the Department of Energy (DOE 2010) daily hot water draw totals for electric resistance and HPWH Energy Factor testing.

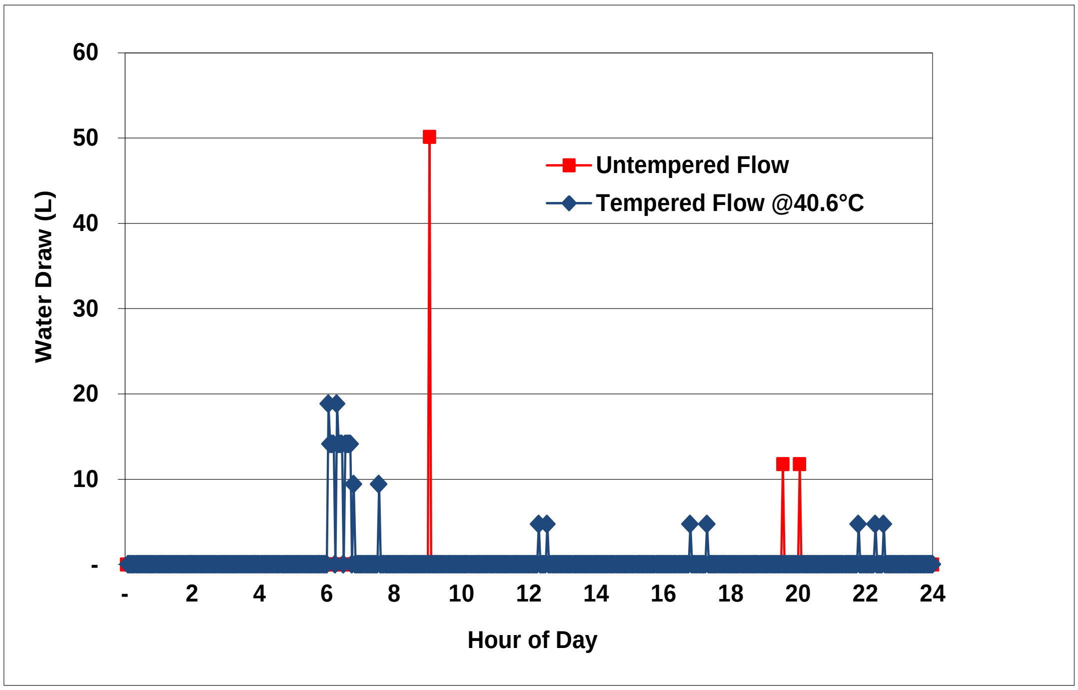


**Fig. 8. Assumed Daily Hot Water Draw Schedule from DHW Tank**

To determine the energy savings potential of the prototype AS-IHP design, a suitable baseline all-electric case was defined and its annual performance simulated in TRNSYS. This consisted of a 7 kW (2-ton) fixed capacity air-source heat pump (ASHP) with a 3.8 W/W cooling season performance factor (CSPF) (13 SEER Btu/Wh), a 2.3 W/W heating seasonal performance factor (HSPF) (7.7 HSPF Btu/Wh) in combination with a 0.90 W/W Energy Factor (EF) electric water heater. The ASHP performance was represented in TRNSYS as a function of ambient and indoor conditions based on a manufacturer's published data.

The time steps in TRNSYS for AS-IHP seasonal performance analysis were set at 3.0 minutes between thermostat call priority decisions. In our initial analysis, control logic rules were applied, as in the AS-IHP concept report by Murphy et al. (2007) to give priority to water heating when both space and water heating calls were active if the indoor DB was within 1.1°C (2°F) of the heating mode set point. Dedicated WH operation was however constrained to a specified minimum ambient due to refrigerant discharge temperature limits. Simulations were run for the same five Building America climate regions (U.S. DOE 2013) as used in the original concept analyses (see Tables 2-5 above). The HVAC-WH energy savings predictions for the reference house in the 5 climates averaged 52%, ranging from 46 to 61% for Chicago and San Francisco, respectively. The average space conditioning savings exceeded 40% while the average water heating savings were 67%. The detailed simulation results are given in Table 6. Table 7 compares the WH load fraction for the larger home used in the Table 6 analyses (2.2 ton SC design load and capacity) to that for the original smaller house used in the concept analyses (1-1.5 ton SC design load/capacity). The WH load fractions for the larger house are slightly lower than those for the smaller concept house.

**Table 6. Energy Use and Savings Predictions for AS-IHP Lab Prototype 1 Design**

| 2600 ft^2 BA House, 2.2 ton design cooling load and unit sizing | | Equipment Performance | | | |
|---|---|---|---|---|---|
| | | Baseline | ASIHP | | |
| Mode | Baseline Delivered Load, kWh | Energy Use, kWh ($I^2R$) | Energy Use, kWh ($I^2R$) | Savings from Base (%) | Comments |
| **Atlanta** | | | | | |
| space heating | 5949 | 2314 | 1300 | 43.8% | V0 MEEHP Prototype Map *Calibrated*, ded. WH allowed in SH+WH call if <2F below set point but only above 40F amb, desup operation to 130F EWT |
| resistance heat | | (42) | (0) | | |
| space cooling | 5670 | 1566 | 858 | 45.2% | |
| water heating | 2963 | 3293 | 1086 | 67.0% | |
| resistance heat | | (3293) | (263) | | |
| ventilation fan | | 189 | 189 | | |
| totals | 14582 | 7361 | 3433 | 53.4% | |
| **Houston** | | | | | |
| space heating | 2851 | 1062 | 576 | 45.7% | Same |
| resistance heat | | (3) | (0) | | |
| space cooling | 9001 | 2498 | 1419 | 43.2% | |
| water heating | 2438 | 2728 | 776 | 71.5% | |
| resistance heat | | (2728) | (115) | | |
| ventilation fan | | 189 | 189 | | |
| totals | 14290 | 6476 | 2960 | 54.3% | |
| **Phoenix** | | | | | |
| space heating | 2119 | 724 | 389 | 46.2% | Same |
| resistance heat | | (1) | (0) | | |
| space cooling | 10428 | 3395 | 2210 | 34.9% | |
| water heating | 2129 | 2392 | 754 | 68.5% | |
| resistance heat | | (2392) | (77) | | |
| ventilation fan | | 189 | 189 | | |
| totals | 14676 | 6700 | 3543 | 47.1% | |
| **San Francisco** | | | | | |
| space heating | 3964 | 1304 | 674 | 48.3% | Same |
| resistance heat | | (1) | (0) | | |
| space cooling | 72 | 21 | 12 | 44.0% | |
| water heating | 3315 | 3676 | 1144 | 68.9% | |
| resistance heat | | (3676) | (75) | | |
| ventilation fan | | 189 | 189 | | |
| totals | 7351 | 5189 | 2019 | 61.1% | |
| **Chicago** | | | | | |
| space heating | 13341 | 6287 | 3885 | 38.2% | Same |
| resistance heat | | (1037) | (629) | | |
| space cooling | 2254 | 623 | 316 | 49.3% | |
| water heating | 3716 | 4110 | 1677 | 59.2% | |
| resistance heat | | (4110) | (697) | | |
| ventilation fan | | 189 | 189 | | |
| totals | 19311 | 11209 | 6066 | 45.9% | |

**Table 7. WH load fraction for 2600 ft$^2$ house used in Table 6 and original nZEH house (Table 2)**

| Location | WH load fraction (% of total SC+SH+WH load) | |
|---|---|---|
| | Table 6 house | Table 2 house |
| Atlanta | 20 | 22 |
| Houston | 17 | 18 |
| Phoenix | 15 | 16 |
| San Francisco | 45 | 53 |
| Chicago | 19 | 21 |

## 5. Annual Energy Use Analysis and Savings Predictions for Second Prototype Design

Following these results, work proceeded to develop and test a second lab prototype design that would be closer to a production product configuration. The 2nd generation prototype also used compact microchannel and brazed-plate air- and water-to-refrigerant HXs. These compact HXs were expected to have similar performance as the tube-and-fin air coils and tube-in-tube water coil used in the first prototype. Figure 9 shows the 2nd generation compressor/WH package, the HW tank, and the indoor air handler for the 2nd prototype as set up in the ORNL test chamber.

**Fig. 9. 2nd generation AS-IHP lab prototype as set up in ORNL test chamber; compressor/WH package in front, WH tank just behind, and air handler in back (with air flow measurement set up).**

This unit was run through similar steady-state laboratory testing in the various operating modes and the data used to re-calibrate the HPDM and repeat the annual performance analyses. A more efficient DHW pump was assumed in this analysis. The pump power relationship as a function of water flow rate was developed based on matching manufacturer's performance curves for a brushless permanent-magnet (BPM) pump against manufacturer's system head curves for an assumed DWH loop head characteristic. The required power for the pump was lowered by ~ 60% at full flow and ~80% at reduced flow. A plot of the pump power and assumed head curve versus water flow for the two pump options is shown in Figure 10. With the slightly stronger BPM pump, the full flow level was increased from 1.2 to 1.36 $m^3/h$ (5.3 to 6 gpm). Note that the low flows below 0.15 $m^3/h$ (0.66 gpm) needed for desuperheater operation required an added flow restriction in the system water line as shown by the change in the system head curve below 0.75 $m^3/h$ (3.3 gpm) .

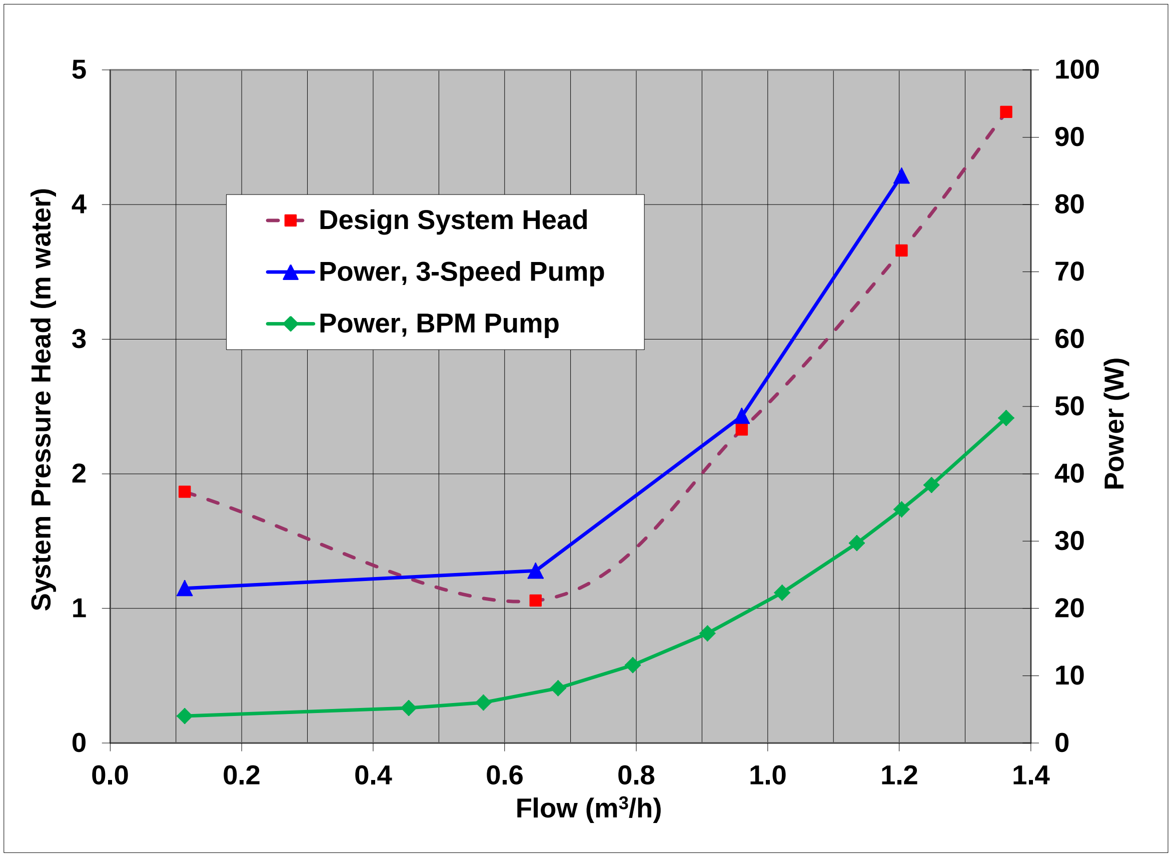


**Fig. 10. Power Comparisons vs. Flow Between 3-Speed Induction and BPM Pumps for Design Head Requirements**

Parametric runs varying condenser subcooling and water flow with the BPM pump were used to determine the optimal levels for the full condensing water heating modes. Figure 11 shows that the combined EER in SC+WH mode (both cooling and WH outputs / input power) has a distinct peak near the maximum water flow rate available from the BPM pump, as shown by the **bold 'X'.**

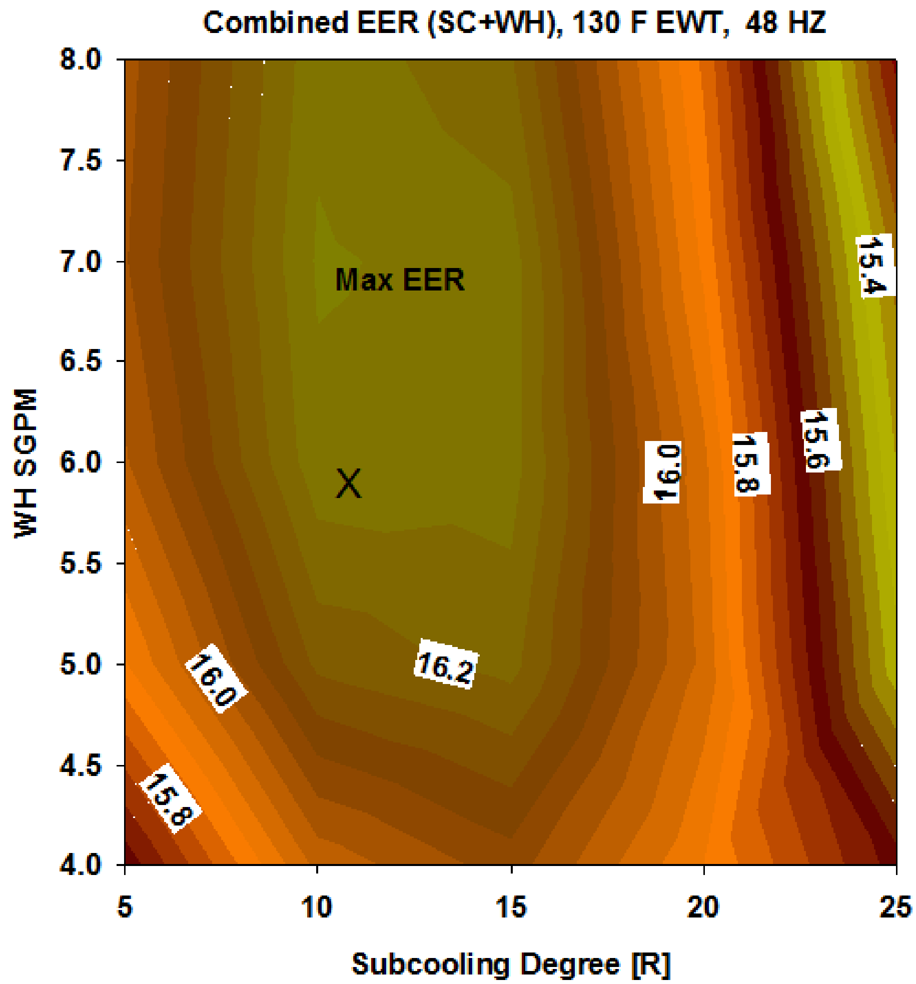


**Fig. 11. Optimal Water Flow vs. Condenser Subcooling for BPM Pump and Brazed Plate HX in Second Prototype Design**

The re-calibrated HPDM was used to generate performance maps for the second prototype AS-IHP unit. These were then used for a second round of annual performance simulations with TRNSYS. As compared to the analyses for prototype 1, thermostat control priority was given in winter operation to SH with WH limited to DS and electric elements until the SH load is satisfied. This approach gave better control of the indoor temperature in the winter season than the previous approach with water heating priority. Dedicated WH (using the outdoor coil as a source) is limited to operation above a specified cutoff ambient, when no space heating call is active, and in shoulder months when the ambient is below a specified cutoff. In SC mode, DS is used first when a WH call is active, until a prescribed water draw is reached, when the unit will switch to combined SC+WH operation.

Results of these annual performance simulations for the five cities are shown in Table 8. The entries in red show the portion of the total energy use for that mode that was from resistance heat. Total HVAC/WH energy savings relative to the all-electric baseline unit again averaged 52%, with a similar range in total savings between the cold and marine climates as before. The predicted average space conditioning savings are 42% with average WH savings of 70%. The increase in predicted WH savings from the first prototype analysis is attributed mainly to the more efficient pump assumed for newer design and a lower allowed ambient limit for dedicated WH operation.

**Table 8. Energy Use and Savings Predictions for AS-IHP Lab Prototype 2 Design**

| Energy Use by Mode, 242 m$^2$ Tight, Well-Insulated House | | | |
|---|---|---|---|
| | Equipment Performance | | |
| | Baseline | Prototype AS-IHP | |
| Operation Mode | Energy Use, kWh ($I^2R$) | Energy Use, kWh ($I^2R$) | Savings from Base (%) |
| **Atlanta** | | | |
| space heating | 2314 | 1359 | 41.2% |
| resistance heat | (42) | (0) | |
| space cooling | 1566 | 905 | 42.2% |
| water heating | 3293 | 987 | 70.0% |
| resistance heat | (3293) | (324) | |
| ventilation fan | 189 | 189 | |
| totals | 7361 | 3440 | 53.3% |
| **Houston** | | | |
| space heating | 1062 | 598 | 43.6% |
| resistance heat | (3) | (0) | |
| space cooling | 2498 | 1480 | 40.7% |
| water heating | 2728 | 664 | 75.7% |
| resistance heat | (2728) | (121) | |
| ventilation fan | 189 | 189 | |
| totals | 6476 | 2931 | 54.7% |
| **Phoenix** | | | |
| space heating | 724 | 398 | 45.0% |
| resistance heat | (1) | (0) | |
| space cooling | 3395 | 2320 | 31.7% |
| water heating | 2392 | 665 | 72.2% |
| resistance heat | (2392) | (117) | |
| ventilation fan | 189 | 189 | |
| totals | 6700 | 3572 | 46.7% |
| **San Francisco** | | | |
| space heating | 1304 | 703 | 46.1% |
| resistance heat | (1) | (0) | |
| space cooling | 21 | 11 | 44.8% |
| water heating | 3676 | 1126 | 69.4% |
| resistance heat | (3676) | (361) | |
| ventilation fan | 189 | 189 | |
| totals | 5189 | 2030 | 60.9% |
| **Chicago** | | | |
| space heating | 6287 | 3974 | 36.8% |
| resistance heat | (1037) | (474) | |
| space cooling | 623 | 340 | 45.5% |
| water heating | 4110 | 1545 | 62.4% |
| resistance heat | (4110) | (691) | |
| ventilation fan | 189 | 189 | |
| totals | 11209 | 6048 | 46.0% |

## 6. Field Test System Performance and Analysis

Based on the favorable performance and projected energy savings of the earlier prototypes NGHVAC proceeded to develop and fabricate a field test prototype in late 2013. The unit was shipped to ORNL in 2014 and installed in a 2,400 $ft^2$ test house (Figure 12) in Knoxville, TN for a one-year field test.  Pictures of the field test system are included in Figure 13 with the field data acquisition system (DAS) shown in Figure 14.

**Fig. 12. Field test site in Yarnell Station neighborhood, Knoxville, TN**

**Fig. 13. Field test prototype installation; l) indoor sections (hot water storage tank, compressor and water heating module, and indoor fan coil), r) outdoor fan coil section**

**Fig. 14. Field DAS**

Before the field testing started, work was done to set up the test house occupancy simulation. The water draw schedule used at the site is based on the latest Building America water draw generator (DOE/BTO Building America Program, 2013). Latent, sensible and other building internal loads are based on the Building America House Simulation Protocols (Hendron and Engebrecht, 2010). Occupancy simulation devices follow a schedule that is input via a database that is read by a programmed controller for operating space heaters (to simulate sensible heat), and humidifiers (to simulate latent heat). Hot water loads (dishwasher, clothes washer, showers, sinks, etc.) are simulated by operating solenoid controlled water valves according to the programmed schedule with an average hot water use of 56.3 gallons/day. Figure 15 shows the hot water valves and controller setup.

**Fig. 15. Hot water use control valves**

The DAS was set up to collect data at 15-sec intervals with 1-min, 15-min, 1-hr, and daily averages. Data was stored on servers located in the new ORNL BTRIC MAXLAB facility (building 4020). A dedicated internet connection was set up that allowed the NGHVAC project team to monitor the data collection in real time.

### 6.1 Cooling Season Field Performance Summary – May - September 2014

Temperature control set points were initially set at 135°F for WH and 76°F for SC and data monitoring began on May 1. Only two minor interruptions occurred during the cooling season.

1. On June 10 the area experienced storms causing a general power outage from 5:30pm until 8:15am, the next morning.
2. On July 27, a severe thunderstorm disrupted the dedicated internet connection so that NGHVAC staff were unable to access the data. However there was no power outage and the AS-IHP operated normally.

A summary of the cooling season performance is given in Figures 16-19. Figure 16 shows the energy consumption by the AS-IHP prototype for space cooling (SC) and water heating (WH) for each operating mode along with monthly totals for May through September. Figure 17 similarly illustrates the SC and WH energy deliveries by mode and totals. The primary operating modes experienced during this period are:

Space cooling only (Ded SC)
Space cooling + desuperheater (DS) water heating (SC+DS)
Space cooling + full condensing water heating (SC+WH)

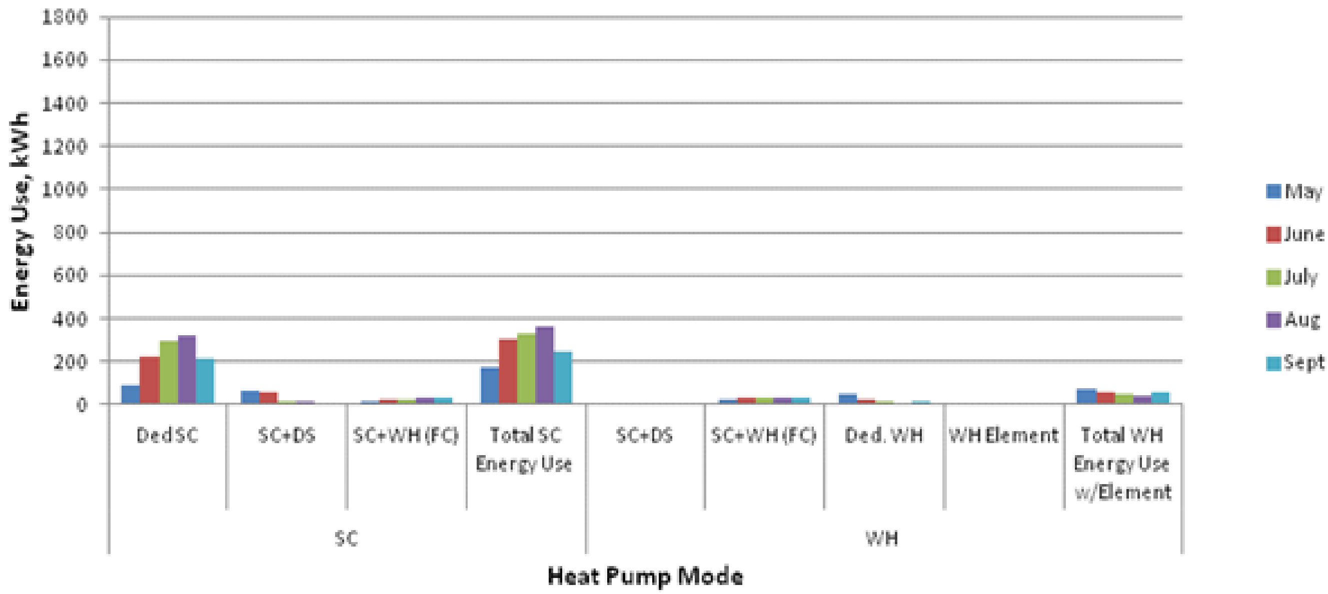


**Fig. 16. Space cooling (SC) and water heating (WH) energy consumption by mode**

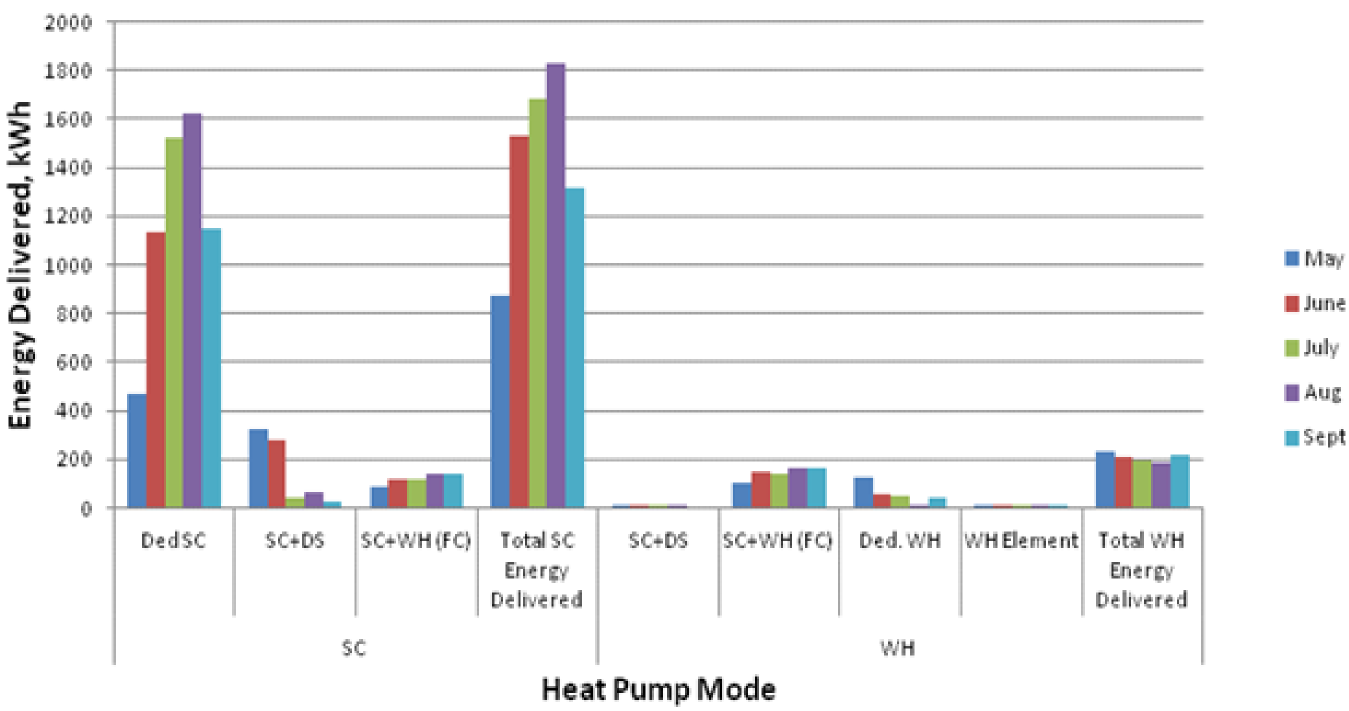


**Fig. 17. Space cooling (SC) and water heating (WH) energy delivered by mode**

Figure 18 illustrates the SC and WH monthly average COPs for each mode and average for the entire month for May through July. The average monthly SC COP ranged from about 5.0 to 5.35 each month while the monthly WH COP ranged from 3.23-4.75 (ignoring electric element power usage). There was a small amount of backup WH electric element energy consumption during the summer but this was due to control system issues (e.g. control computer failing to reboot properly, etc.). No element usage would be expected in the summer period under the hot water use profile in effect at the test house.

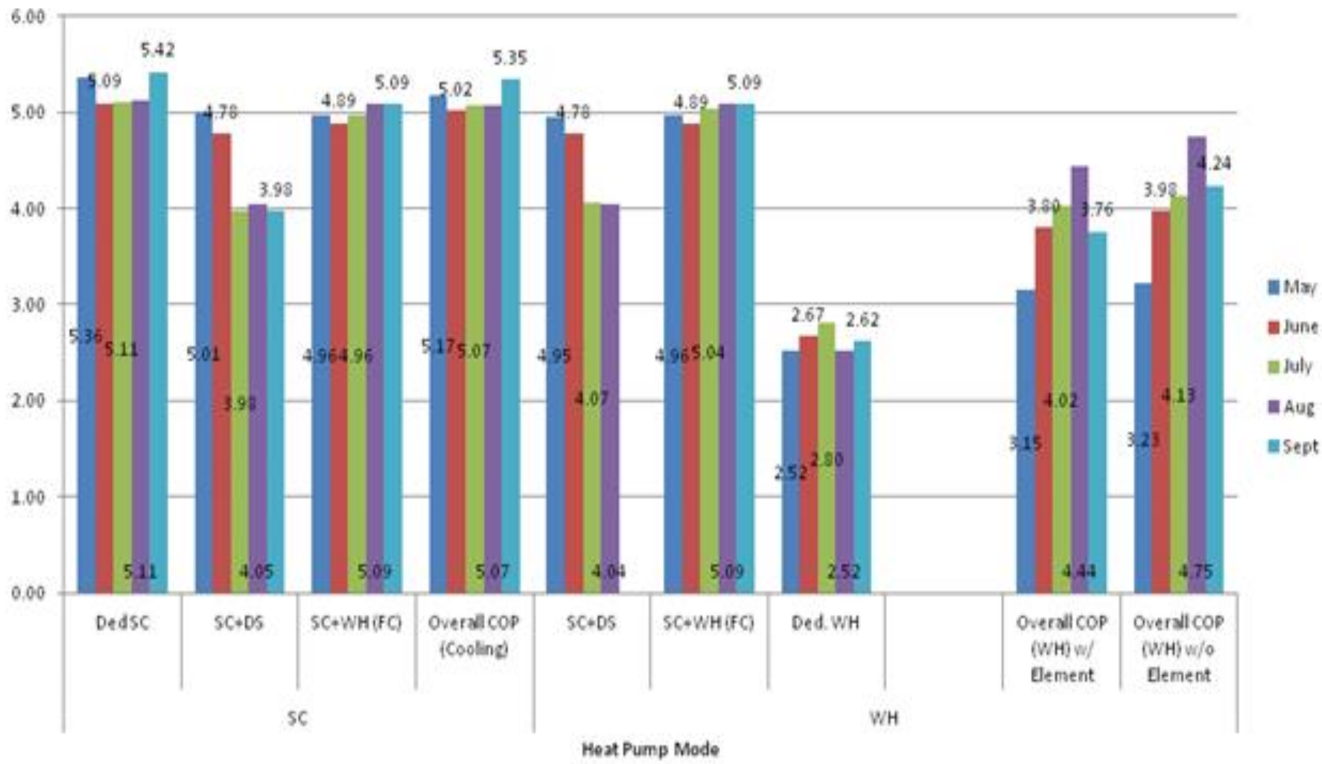


**Fig. 18. Average monthly SC and WH COPs by mode and overall COPs**

The overall average system efficiency for the AS-IHP system from May through September is given in Table 9. Average overall efficiencies by mode (SC and WH) are included as well. Average seasonal efficiencies were 5.14 for SC, 4.00 for WH (neglecting the backup element energy use), and 4.96 for the overall average, including the small amount of SC+WH mode operation in October.

**Table 9. Seasonal average COPs (May-Sept)**

| Mode | Energy delivered kWh | Energy use kWh | Average COP |
|---|---|---|---|
| SC | 7236 (+ ~180 in Oct) | 1413 (+ ~31 in Oct) | 5.12 ~(5.14 incl Oct) (SEER = 17.47 Btu/Wh) |
| WH (no element) | 1013 (+ ~47* in Oct) | 256 (+~9* in Oct) | 3.96 (~4.0 incl. Oct) |
| Total/average | 8249 (~8476 incl. Oct) | 1669 (~1709 incl. Oct) | 4.94 (~4.96 incl. Oct) |

*SC+WH (FC) mode only.

In late June, the system controls were modified to reduce the set point temperature for WH to 120 °F and to limit time spent in the relatively inefficient space cooling plus desuperheating WH (SC+DS) operating mode. For the last three months of the season the average SC COP was up slightly to 5.16. The average WH COP rose ~10% to 4.41 and the overall average COP rose ~2% to 5.07, as shown in Table 10.

**Table 10. Seasonal average COPs (July-Sept)**

| Mode | Energy delivered kWh | Energy use kWh | Average COP |
|---|---|---|---|
| SC | 4832 (~5012 incl. Oct) | 940 (~971 incl. Oct) | 5.14 (~5.16 incl. Oct) (SEER = 17.54 Btu/Wh) |
| WH (no element) | 579 (~626 incl. Oct) | 133 (142 incl. Oct) | 4.35 (~4.41 incl. Oct) |
| Total/average | 5411 (~5638 incl. Oct) | 1073 (~1113 incl. Oct) | 5.04 (~5.07 incl. Oct) |

Table 11 provides an estimate of the overall seasonal average COPs for the test system assuming that the WH set point had been at 120°F for the entire May-September summer period. The WH load (energy delivered) was adjusted as follows. First the average daily WH energy delivered was estimated for July-September by 579 kWh / 91 days = 6.363 kWh/d. Then the WH delivered energy for the entire May-September test period was estimated by 6.363 kWh/d × 152 days = 967 kWh. This value was used with the July-September WH COP in Table 9 to estimate total WH energy consumption for May-September as 222 kWh.

**Table 11. Seasonal average COPs (May-Sept); for WH set point = 120 °F**

| Mode | Energy delivered kWh | Energy use kWh | Average COP |
|---|---|---|---|
| SC | 7236 (~7416 incl. Oct) | 1413 (~1444 incl. Oct) | 5.12 (~5.14 incl. Oct, or SEER = 17.54 Btu/Wh) |
| WH (no element) | 967 (~1014 incl. Oct) | 222 (~231 incl. Oct) | 4.35 (~4.39 incl. Oct) |
| Total/average | 8203 (~8430 incl. Oct) | 1635 (~1675 incl. Oct) | 5.02 (~5.03 incl. Oct) |

Figure 19 gives the actual monthly electricity costs for SC and for WH during the summer test period (note that the costs include the impact of the backup WH element usage) calculated at \$0.11/kWh, the prevailing national average rate for residential electricity. The total monthly cost peaked at ~\$49 in August as expected (hottest month with greatest cooling demand during the test period). Monthly WH costs trended lower from May-August but increased again in September. The primary reasons are: (1) backup element usage was 8.4 kW during September, higher than the other months; and (2) use of the dedicated WH mode increased in September. WH COP in the SC+WH mode is much higher than for dedicated WH mode.

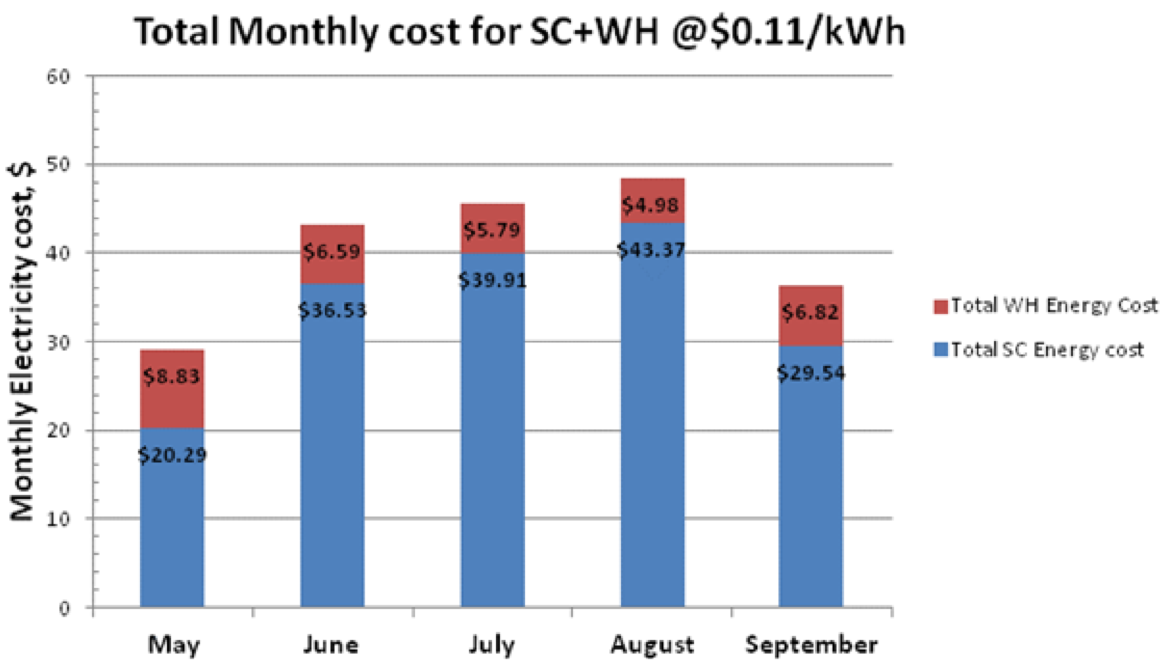


**Fig. 19. Total monthly electricity costs for SC and WH**

Monthly data tables are provided in Appendix A.

### 6.2 Heating Season Field Performance Summary – October 2014 – April 2015

Temperature control set points of 120°F for WH and 71°F for SH were implemented in the system controls prior to starting data monitoring in October. Operational interruptions and adjustments made during the heating season are as follows:

1. In early November, 2014, adjustments were made to correct the compressor speed during WH cycles, and subcooling temperature entering the compressor.
2. Loss of refrigerant charge occurred on December 19, 2014, within a 3-4 hour period, due to leakage at the header tube of the OD microchannel coil as shown below in Figure 20. A replacement microchannel coil was shipped by NGHVAC and installed by ORNL on 12/23/2014. In the meantime, the thermostat was switched to auxiliary heat to maintain space heating set point of 71°F.
3. Compressor underperformance during January was suspected due to degradation of oil caused by operating the compressor above its discharge temperature limit during the loss of charge event and in WH operating modes after potentially overcharging the unit after the coil replacement. This resulted in lower WH COPs than normal for January. A new compressor (same make and model) was installed on January 16, 2015.
4. Excessive frosting in the 3rd week in January caused underperformance of the AS-IHP (see Figure 21, top, taken with visible and infra-red light cameras). Defrosting the coil got rid of most of the frost (Figure 21, bottom) that was adversely affecting performance. While the defrosting valve appeared to work adequately the defrost cycle was unable to completely clear the coil of frost during this high humidity period. This left the lower section colder than the upper section as can be seen in the

infrared image at bottom right of Figure 21. The air flow is also compromised in the lower section. Frost built up rapidly on the OD coil during this period (within 30-45 minutes) requiring frequent defrost cycles. The defrost timings on the unit were adjusted to help alleviate this issue, and subsequent cold days with high humidity did not result in as much frost buildup before defrosting and defrost cycles were completely clearing the coil.  NOTE:  While this issue did not recur during the 2015 winter test season, the root cause remains to be fully determined.  It is quite possible that proper frost melt drainage from the microchannel outdoor coil may be the primary issue. Absent adequate and reliable defrost water drainage this issue could recur given the right combination of outdoor temperature and humidity conditions.

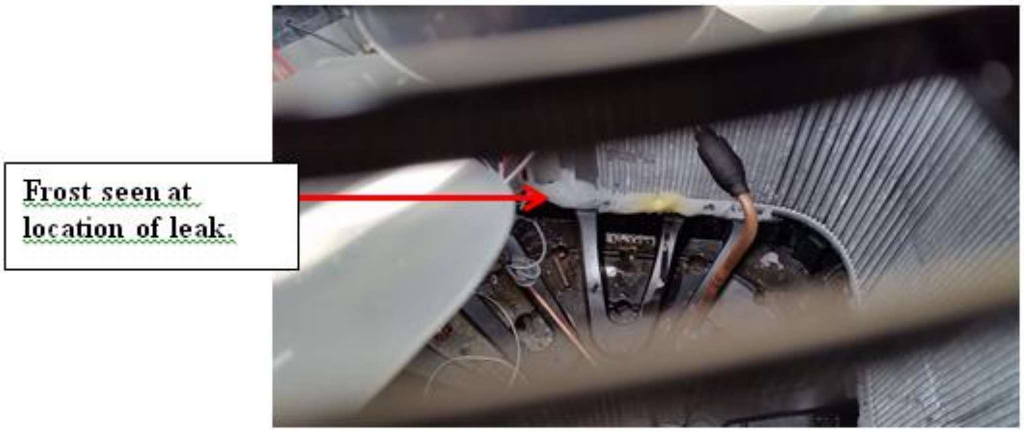


**Fig. 20.  Location of refrigerant leak in the header section of the microchannel OD coil**


**Fig. 21.  Frosted coil (top) and after defrosting was completed (bottom).**

A summary of the field unit performance is given in Figures 22 to 28. Figure 22 shows the energy consumption by the AS-IHP prototype for space heating (SH) and water heating (WH) for each operating mode along with monthly totals for October 2014 through April 2015. Figure 23 similarly illustrates the SH and WH energy deliveries by mode and totals. The primary operating modes experienced during this period are:

Space heating only (Ded SH)
Space heating + desuperheater (DS) water heating (SH+DS)
Dedicated Water Heating (WH)

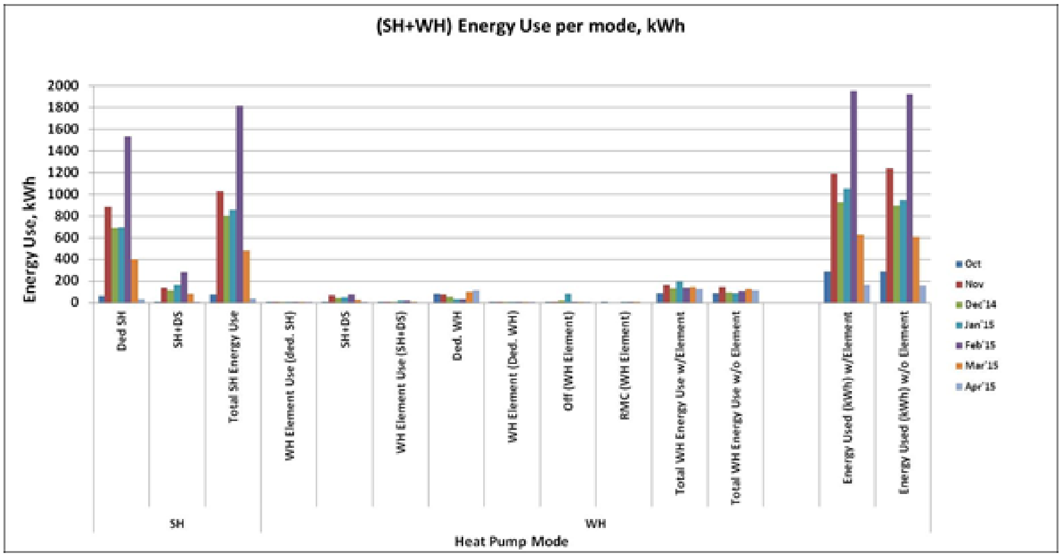


**Fig. 22. Space heating (SH) and water heating (WH) energy consumption by mode.**

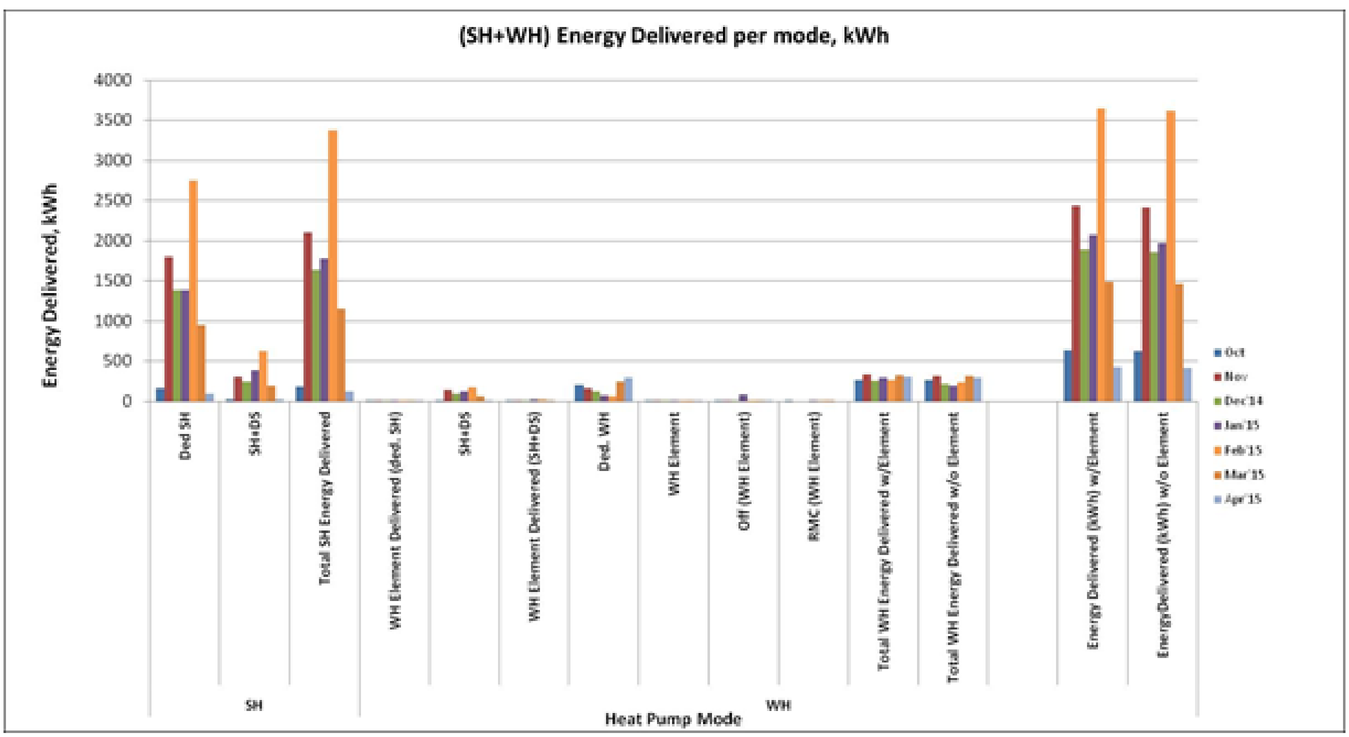


**Fig. 23. Space heating (SH) and water heating (WH) energy delivered by mode.**

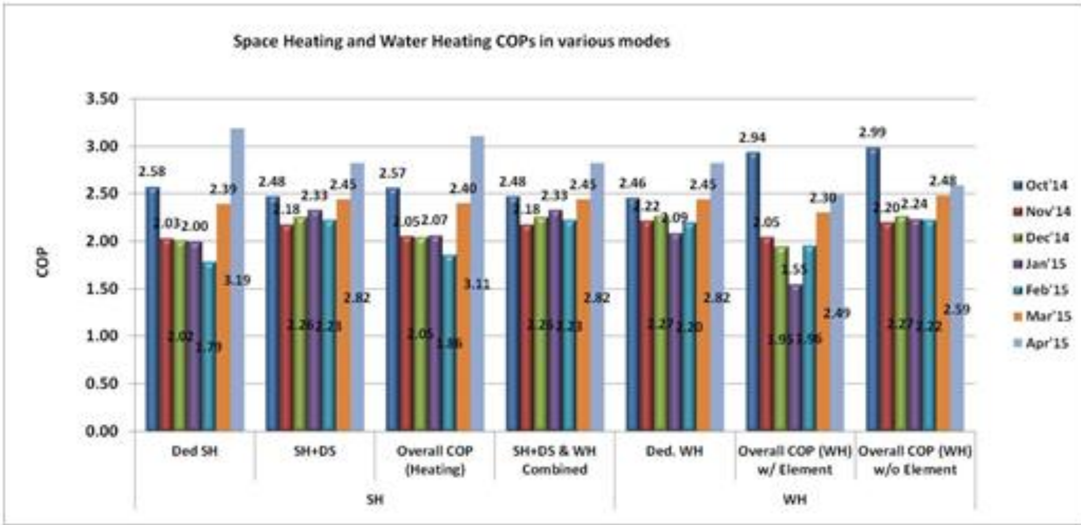


**Fig. 24. Space heating and water heating COP by month.**

Figure 24 illustrates the SH and WH monthly average COPs for each mode and the overall monthly average COP for October 2014 through April 2015. The average monthly overall SH COP (including both dedicated SH and SH+DS modes) ranged from a low of 1.85 (in February during record breaking cold weather) to a high of 3.22 (in April, 2015) while the monthly WH COP ranged from 1.55 (Jan'15) to 2.94 (Oct'14) including electric element power usage. The low WH COP of 1.55 was in January due to the refrigerant loss, compressor replacement and defrosting issues discussed above.

The measured overall heating seasonal system energy deliveries and COPs by mode (SH and WH) for the AS-IHP are given in Table 12. Average seasonal COPs were 2.06 for SH (including backup electric element use for SH and during defrosting), 2.16 for WH (including backup electric elements), and 2.07 for the overall average. Note that the values in Table 12 have been corrected to include estimated energy delivery and usage by the heat pump for the period of 12/19 to 1/16 when the AS-IHP was inoperative due to the coil failure and compressor replacement as noted above.

**Table 12. Seasonal average COPs (Oct'14-April '15)**

| Mode | Energy delivered kWh | Energy use kWh | Average COP |
|---|---|---|---|
| SH | 12125 | 5899 | 2.06 (HSPF= 7.01Btu/Wh) |
| WH (with element) | 2090 | 968 | 2.16 |
| Total/average | 14,215 | 6867 | 2.07 |

Figure 25 gives the actual (uncorrected) monthly electricity costs for SC, SH and for WH during the cooling and heating season (including the impact of the backup SH and WH element usage) for comparative purposes. The total monthly cost peaked at ~$219 in February as expected (coldest month with greatest heating demand during the test period). Monthly WH costs increased gradually from October to December. WH cost increases in January were due to the defrosting and poor compressor performance issues discussed above. The primary reasons for the January WH energy usage spike are: (1) backup element usage was 83.6 kW during January during downtime to replace the compressor, and (2) lower entering water temperatures.

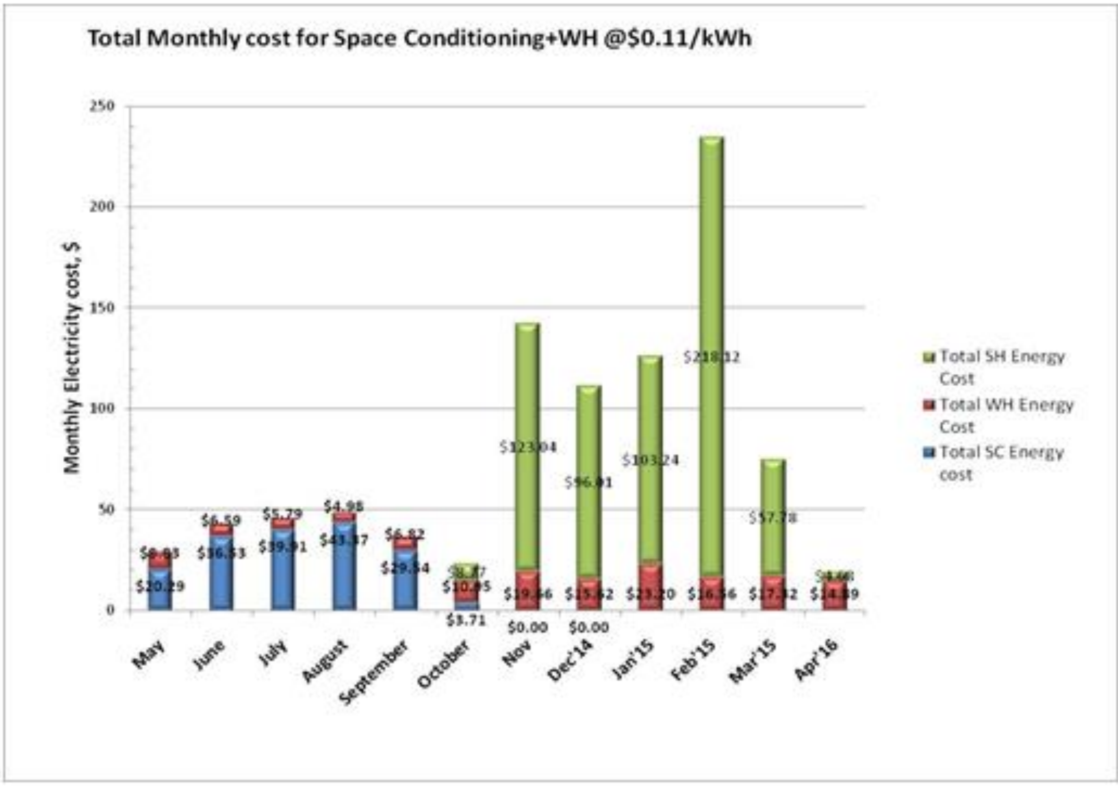


**Fig. 25. Total monthly electricity costs for SC and WH**

Since WH demand is greater in the winter than during summer, it is instructive to report the amount of WH that was done with the heating element against the total water heating energy consumed, as shown in Figure 26. Apart from the anomalous month of January the overwhelming proportion of WH energy throughout the winter test period was supplied by the heat pump.

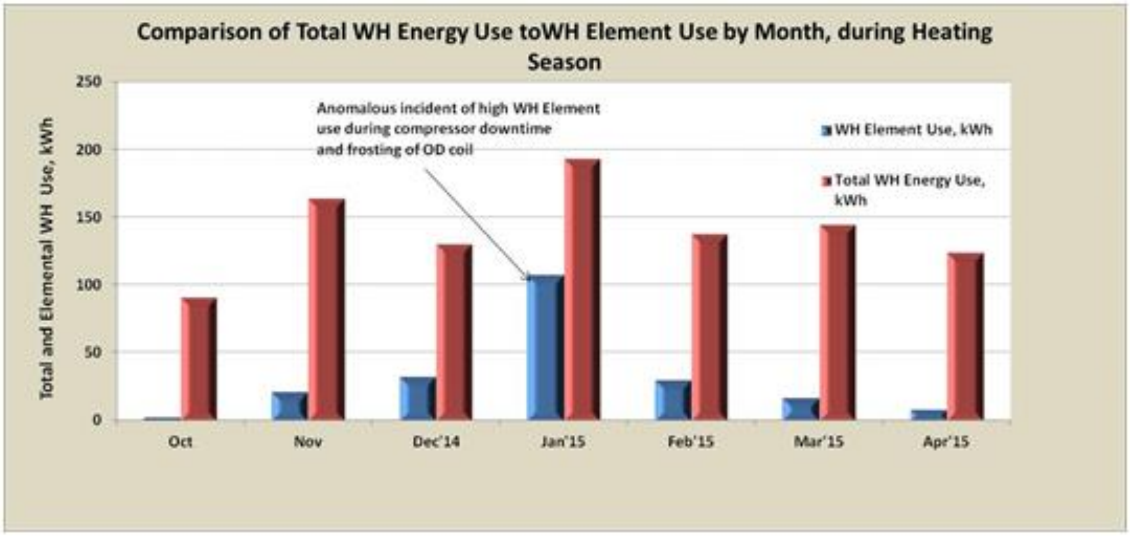


**Fig. 26. Total WH energy use and the portion supplied by back up electric elements.**

The percentage of SH energy supplied by the system's back up electric heater for both the SH and in SH+DS modes is shown in Figure 27. Back up SH energy peaked in the exceptionally cold month of February at 20% and 10.9% in dedicated SH and in SH+DS modes, respectively. But it must be noted that the auxiliary electric heat data in Figure 27 includes both supplemental SH and defrost tempering usage. Defrost tempering usage in February alone is estimated to account for about half of the total auxiliary element energy use. Both auxiliary SH and defrost tempering heat was supplied by a single electric element of 9.46 kW.

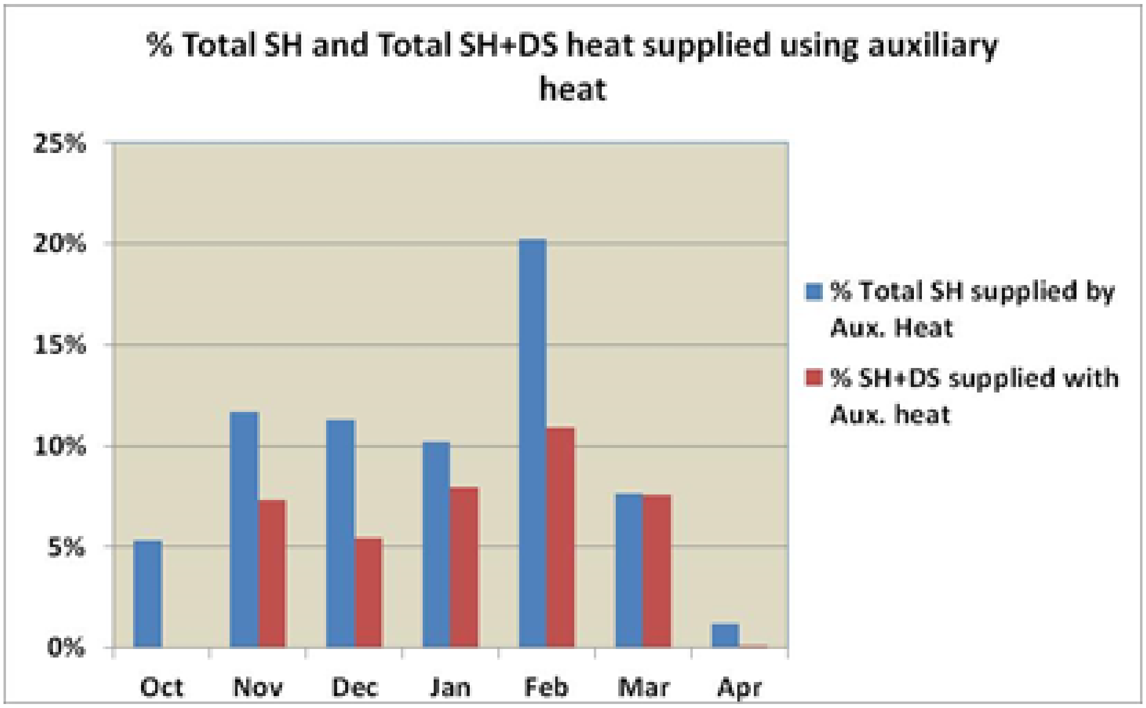


**Fig. 27. Back up electric SH energy use in SH and SH+DS modes.**

During the heating season, most of the AS-IHP operating time occurred in the dedicated SH mode, followed by SH+DS mode. The total hours in dedicated WH mode are much less than the hours spent in SH+DS mode. This trend was the same during the cooling season when the dominant mode is SC followed by SC+WH. The operational hours for each mode from May 2014-April 2015 are shown in Figure 28. It should be noted that the WH capacity in the full condensing modes (dedicated WH and SC+WH) is much larger than that in DS mode and so contributes strongly to the shorter run times and fast WH recovery in these modes.

An interesting feature shown in Figure 28 is that the AS-IHP remained in the off mode for substantial hours each month (except for the exceptionally cold month of February). In the cooling season more time was spent in the off mode than in dedicated SC mode. In the heating season, somewhat less time was spent in the off mode than for dedicated SH. SC+DS is a minor operational mode as far as duration is concerned. Therefore, while relative to the cooling season the AS-IHP may appear to be oversized, this is not necessarily the case relative to the heating season.

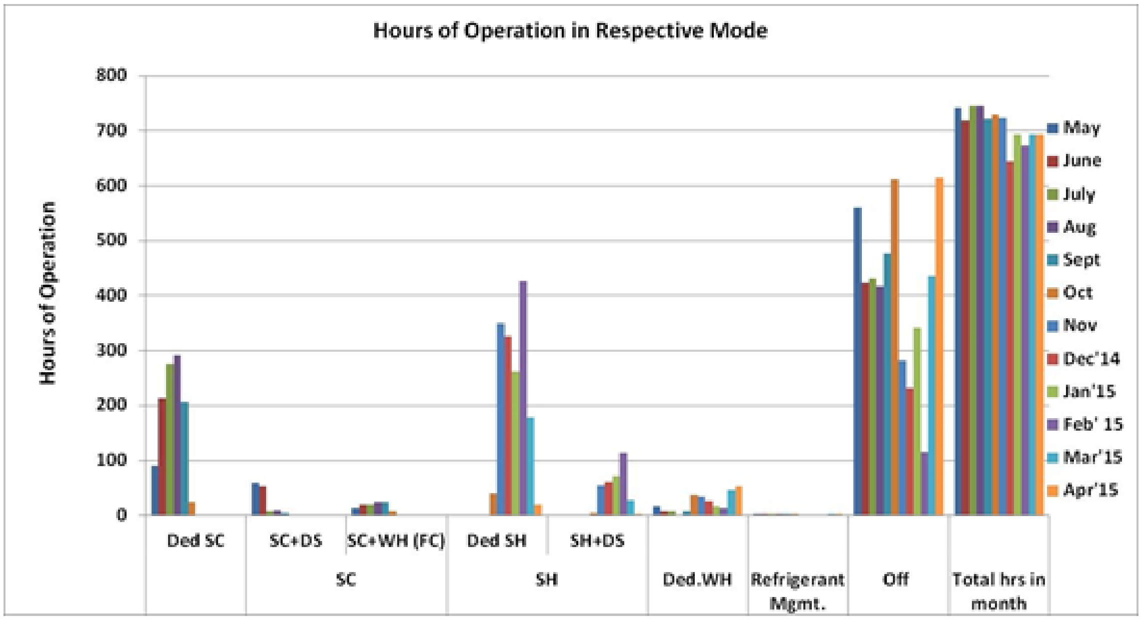


**Fig. 28. Operational hours in each mode of the AS-IHP during cooling and heating seasons**

The monthly averaged outdoor air temperature and relative humidity (%RH) for the heating season are shown in Figure 29. February was an exceptionally cold month with the average monthly temperature just above 30°F. The relative humidity throughout the heating season ranged from 68% (February) to 81% (December).

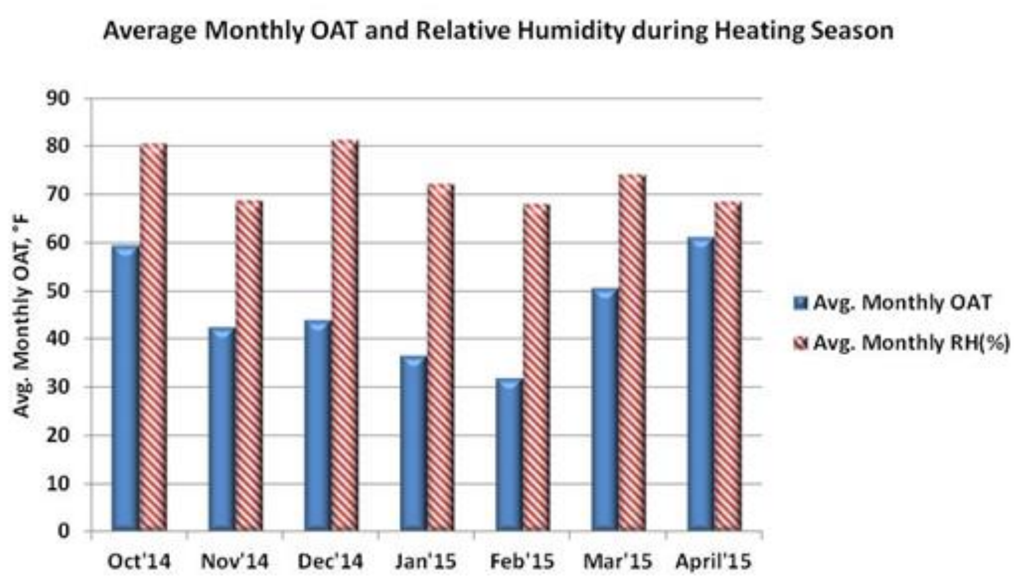


**Fig. 29. Averaged monthly OAT and relative humidity, RH (%) during the heating season.**

Monthly data tables are provided in Appendix A.

### 6.3 Annual Field Performance Summary and Heating Season Field Performance Observations & Discussion

Tables 13 and 14 compare the measured seasonal COPs and delivered loads for the AS-IHP field test to those calculated with the TRNSYS/HPDM (T/H) models. While the field data and the T/H predictions are not directly comparable (different house thermal envelope performance levels, real weather vs. average; different locations; differences between assumed control strategy for T/H simulations and the field test unit control approach, e.g. simple two-stage thermostat for SH and SC, etc.) they do provide at least a qualitative feel for how the field performance compares to predictions. It needs to be noted, however, that the daily DHW use at the field test house was 56.3 gal/d vs. the 64.5 gal/d used for the simulations. This reduces the overall WH load fraction with concomitant negative impact on the overall system annual energy savings potential.

**Table 13. Annual average AS-IHP COPs: T/H calculations vs. field measured**

| Mode | Measured COP (from Tables 10 and 11[2]) | T/H calculated COP[1] | | | | |
|---|---|---|---|---|---|---|
| | | Atlanta | Houston | Phoenix | San Francisco | Chicago |
| SH | 2.06 | 4.38 | 4.77 | 5.32 | 5.63 | 3.36 |
| SC | 5.14 | 6.26 | 6.08 | 4.49 | 6.54 | 6.63 |
| WH[3,4] | 2.68 | 3.34 | 4.11 | 3.60 | 3.26 | 2.66 |

[1]For 2600 $ft^2$ tight, well insulated house (as used for Tables 6 and 8 above)
[2]includes estimated October cooling mode SC and WH energy
[3]For 120°F WH thermostat set point. Includes backup electric element in winter; no element in summer
[4]Based on energy delivery from AS-IHP to WH tank (tank and line losses not included)

**Table 14. Annual delivered loads (kWh): T/H calculations vs. field measured**

| Mode | Measured (from Tables 10 and 11[2]) | T/H calculated[1] | | | | |
|---|---|---|---|---|---|---|
| | | Atlanta | Houston | Phoenix | San Francisco | Chicago |
| SH | 12125 | 5949 | 2851 | 2119 | 3964 | 13341 |
| SC | 7416 | 5670 | 9001 | 10428 | 72 | 2254 |
| WH[3,4] (% of total) | 3104 (14) | 3293 (20) | 2728 (17) | 2392 (15) | 3676 (45) | 4110 (19) |

[1]For 2600 $ft^2$ tight, well insulated house (as used for Tables 6 and 8 above)
[2]includes estimated October cooling mode SC and WH energy
[3]For 120°F WH thermostat set point. Includes backup electric element in winter; no element in summer
[4]Based on energy delivery from AS-IHP to WH tank (tank and line losses not included)

The field measured SC and WH COPs in Table 13 are within the range of the T/H predicted values, albeit toward the lower end. One reason that the field WH COP is lower is that the water pump used in the field test had somewhat higher energy use than that of the pump model assumed in the analyses, both for FC and especially DS WH operation. The field measured SH performance is much lower than that for any of the simulation locations. Both the SC and SH seasonal measured SEER and HSPF were also lower than the estimated AHRI 210/240 (AHRI 2008) rated values for the prototype, as seen in Table 15. The AHRI estimates were computed using both the minimum and maximum house load line assumptions (DHRmin and DHRmax) and for the actual demonstrated test house load lines for 2104/2015 (Figure 30)

**Table 15. Site measured seasonal SH and SC HSPFs (Btu/Wh) vs. estimated AHRI 210/240 ratings for prototype[1] system**

| Mode | Field measured | AHRI 210/240 | % deviation field vs. rated |
|---|---|---|---|
| SH HSPF | 7.01 | For DHRmin load - 10.17<br>For DHRmax load- 8.31<br>For house loads - 9.00[2] | 31<br>16<br>22 |
| SC SEER | 17.54 | For default load and 0.2 Cd - 18.73<br>For house loads - 19.11 | 6<br>8 |

[1]Based on lab measured performance of 2nd prototype
[2]Used AHRI default frost/defrost (F/D) penalties for max speed at 35 °F and the same F/D-to-SS performance multipliers of 0.9 for capacity and 0.985 for power at 35 °F intermediate speed operation

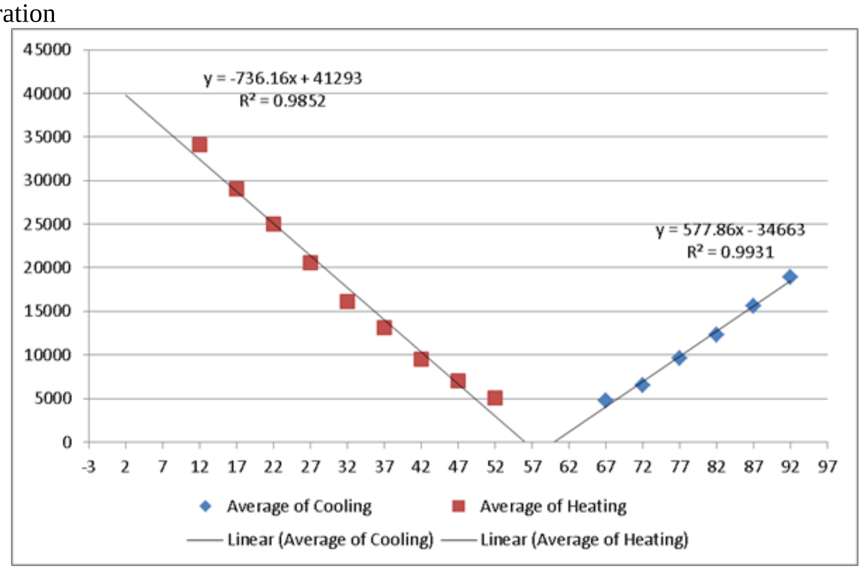


**Fig. 30. Field test house 2014/2015 heating and cooling load lines**

Field measurements on two single-speed (SS) ASHPs were done in the Knoxville area in 2011-2012. Both were tested in a single, two-story house with SS unit 1 conditioning the downstairs and unit 2 conditioning the upstairs. Heating season measurements showed HSPFs of 5.2 and 6.0 for units 1 and 2, respectively (Munk et al 2013). These are 32% lower and 22% lower than the HSPF rating for the units of 7.7 (per AHRI 210/240 based on DHRmin load line). This is similar to the 31% deviation in Table 15 from the estimated AHRI rating for the field prototype (based on DHRmin load line). For cooling operation, however, the two SS ASHPs had field-measured SEERs of 7.1 and 8.4 (45% and 35% deviation from rated) while the field AS-IHP prototype field measured SEER was 17.54 (only 6% deviation from estimated rating value using default load & 0.2 Cd).

There are a number of reasons why the AS-IHP field prototype's measured SH COP is lower than might be expected:

- Blower energy use is higher at the field site than was measured in the lab test phases of the project due to higher duct system external static pressure losses. This is somewhat peculiar to changes made in the test house ducting system to accommodate the AS-IHP. But in general residential duct systems have higher pressure losses than that implicitly assumed in AHRI 210/240 (0.1" water gage). [NOTE – this also negatively impacts the SC seasonal COP.]
- The HSPF procedure does not account for defrost tempering heat usage. This accounted for >10% of the total field system energy use in February alone.
- The indoor temperature during the heating season averaged close to 72°F while lab testing and the HSPF procedure assume 70°F
- The standard house load line used in the HSPF procedure is lower than that experienced at the test house this winter.
- It is also possible that the backup electric heat controls of the field test system may have been driving the indoor space temperature up to the point that the thermostat was satisfied and the system cycled off. If so, controls modification could minimize backup heater usage and avoid system cycling during cold outdoor conditions. The NGHVAC team leader also mentioned that it might be possible to set up the system controls to increase compressor speed to the maximum at some point before the thermostat calls for resistance heat. It is useful at this point to recall that much of the SH seasonal energy use for the field test system was from resistance heat (see Figure 27). Note that in the TRNSYS simulation (Tables 6 and 8), with the exception of Chicago, there was no resistance heat usage estimated for SH or defrost tempering for the AS-IHP and very little for the baseline ASHP. With a significant amount of resistance heat usage (cf. the Chicago case in Tables 6 and 8) the potential annual energy savings for the IHP drops significantly.
- Many of the issues related to the SH control for the field test prototype may also be an unintended consequence of the use of a generic, low-cost 2-stage thermostat to control a variable speed system. Setting up optimal sequence timing for control of the compressor speed based only on a high or low stage thermostat input is a major challenge. It is likely that this approach will not provide good results in all homes due to differences in equipment sizing relative to the actual heating load and the thermal mass of the home.

Table 16 compares average heating and cooling degree-days for Knoxville to those experienced during the 2011-2012 and 2014-2015 test years. The table provides the average degree-days for the five simulation locations as well for comparison. Interestingly, the actual site-measured degree-days for 2014-2015 appear to be a closer overall match to the average year downtown Chicago (Midway airport location) weather than to that of any of the other simulation sites. The 2014-2015 test year weather for Knoxville was somewhat cooler than the long-term averages per ASHRAE (2013) for both heating (~12% colder) and cooling seasons (~8% cooler). It would appear that the high SH and SC loads experienced at the test site are a reflection of poorer thermal envelope (less insulation, less tight) performance of the test house as compared to the house used for the simulations reported in Tables 6 and 8. The 2011-2012 actual weather (when the two SS ASHPs were tested) was a bit warmer than normal; 22% warmer heating season and 14% warmer cooling season.

**Table 16. Average vs. 2014/2015 test site heating and cooling degree-days**

| Location | Annual °F-days heating (65°F base) | Annual °F-days cooling (65°F base) |
|---|---|---|
| Atlanta[1] | 2671 | 1893 |
| Houston[1] | 1371 | 3059 |
| Phoenix[1] | 923 | 4626 |
| San Francisco[1] | 2689 | 144 |
| Chicago[1] | | |
| O'Hare | 6209 | 864 |
| Midway | 5872 | 1034 |
| | | |
| Knoxville | | |
| Average[1] | 3594 | 1514 |
| 2011-2012[2] | 2796 | 1725 |
| 2014-2015[2] | 3845 | 1486 |
| 2014-2015[3] | 4020 | 1400 |

[1]1986-2010 averages from ASHRAE (2013).
[2]test year actuals from National Oceanic and Atmospheric Administration (NOAA 2015) for Knoxville McGhee-Tyson airport weather station
[3]For test year – 5/3/2014 to 5/2/2015; site-measured actual

It is not unusual for actual measured heat pump HSPFs to be degraded by 30% or more compared to the HSPF rating (based on the DHRmin load line) due to the reasons cited above plus other miscellaneous effects like colder than normal winters. The higher house load effect alone likely accounts for more than half of the degradation.

***Estimated field prototype AS-IHP energy savings vs. baseline minimum efficiency system at test site.*** Annual energy use of a baseline system [13 SEER & 7.7 HSPF (Region IV) SS ASHP and electric WH] meeting the field test site loads was estimated as described below. Both the HSPF and SEER ratings for the baseline unit were adjusted downward by 27% and 40%, respectively, based the average field measured deviations from rated efficiencies experienced by SS ASHPs previously field tested in the Knoxville area (Munk et al 2013). The results for this comparison are shown in Table 17. Since the tank and hot water distribution line losses from the hot water storage tank were not accounted for in the AS-IHP field performance, they are also omitted from the baseline equipment efficiency (e.g., baseline WH COP = 1.0). The table shows that the largest percentage and absolute savings come from water heating, at 61% and 1905 kWh respectively. SC and SH energy savings are estimated at 1800 kWh (55%) and 1461 kWh (20%), respectively. Estimated total annual savings for the AS-IHP vs. estimated baseline energy use at the Knoxville test site are about 38%, which is a bit lower than the T/H predicted savings (Tables 6 and 8). Heavy reliance on back up electric elements for SH and defrost tempering coupled with higher indoor blower energy usage (vs. lab measured performance) and higher water pump energy usage (vs. that of the pump assumed for the simulations reported in Tables 6 and 8) were likely the major causes of the lower than expected SH performance of the AS-IHP field prototype system. In addition, the higher space conditioning loads relative to the WH load for the field test house reduced the weighting of the WH mode energy savings in the total annual energy savings calculation. A smaller rated capacity IHP in combination with a better-insulated house as simulated in the T/H analyses above (Tables 6 and 8) would be a much closer match to the preferred application and possibly would have yielded total energy savings of ~45% or more at the test site.

**Table 17: AS-IHP 2014-2015 measured performance vs. estimated baseline performance at test site**

| **Mode** | | **AS-IHP** | **Baseline system estimated performance** | **Percent Savings Over Baseline** |
|---|---|---|---|---|
| Space Cooling | COP (SEER) | 5.14 (17.52) | 2.29 (7.80) | |
| | Delivered (kWh) | 7416 | 7416 | |
| | Consumed (kWh) | 1444 | 3244 | 55% |
| Space Heating | COP (HSPF) | 2.06 (7.01) | 1.65 (5.62) | |
| | Delivered (kWh) | 12125 | 12125 | |
| | Consumed (kWh) | 5899 | 7360 | 20% |
| Water Heating | COP | 2.68 | 1 | |
| | Delivered (kWh) | 3104 | 3104 | |
| | Consumed (kWh) | 1199 | 3104 | 61% |
| **Total** | **Consumed (kWh)** | **8542** | **13708** | **38%** |

Application of the AS-IHP system to commercial buildings where the annual loads are dominated by WH and SC needs would also be expected to yield much higher annual energy savings than was demonstrated during this residential field test.

## 7. Future Commercial Product Launch

Nortek Global HVAC LLC is actively pursuing plans to introduce a heat pump product that implements the features of the AS-IHP field test prototype. The major development need remaining is to finalize the controls and control software and convert the field prototype control system to a solid-state hardware design (pc board, etc.) more suitable for production line use. They are assessing AS-IHP product introduction against other new product priorities.

## 8. Conclusions

Between August 2011 and September 2015, Oak Ridge National Laboratory (ORNL) and Nordyne, LLC (now Nortek Global HVAC LLC, NGHVAC) engaged in a Cooperative Research and Development Agreement (CRADA) to develop an air-source integrated heat pump (AS-IHP) system for the US residential market. Over the course of the CRADA three prototype systems, (two lab prototypes and a field test prototype) were designed, fabricated, and tested. The prototypes were modeled in TRNSYS to predict annual performance relative to a baseline suite of equipment meeting minimum efficiency standards in effect in 2006 (combination of air-source heat pump (ASHP) and resistance water heater). For the final prototype design, predicted total annual energy savings, while providing space conditioning and water heating for a tight, well insulated 2600 $ft^2$ (242 $m^2$) house at 5 U.S. locations, ranged from 46% to 61%, averaging 52% (lowest savings at the cold-climate Chicago location). Based on the field performance of the prototype at the Knoxville test house (2400 $ft^2$ and less well-insulated) and estimated performance for a baseline system operating under the same loads and weather conditions it was estimated that the AS-IHP would achieve ~40% energy savings relative to the minimum efficiency suite. The estimated WH mode savings were >60%, and SC mode savings >50%, but estimated SH savings were only about 20%. It is projected that had the test house been better insulated (more like the house used for the savings predictions noted above and in tables 6 and 8) and the IHP system nominal capacity been a bit lower that the energy savings projection would have been closer to 45% or more (similar to that analytically predicted for the cold climate location of Chicago).

The major items impacting the field measured SH energy use for the prototype AS-IHP were 1) heavy reliance on back up electric elements for SH and defrost tempering, and 2) higher indoor blower and water pump energy usage as compared to lab measured blower and simulated pump performance. In addition the daily HW usage for the field test averaged about 56.3 gallons/day – ~13%, lower than that assumed for the analytical performance simulations (64.5 gal/d). The lower HW use coupled with the higher space conditioning loads due to the poorer thermal envelope of the test house cause the WH load during the test year to be a smaller fraction of the total load on the IHP. This reduced the weighting of the WH mode energy savings with concomitant negative impact on the total annual energy savings estimate.

NGHVAC is actively evaluating the AS-IHP system for a possible future new product launch. A specific timeframe has yet to be decided.

The demonstrated cooling season performance results of the residential prototype under this CRADA indicate that application to commercial buildings with thermal loads dominated by WH and SC needs (e.g., restaurants, commercial laundries, health/fitness centers, lodging facilities, etc.) would have higher annual energy savings potential. A follow on CRADA project is planned with a goal to develop and field test a prototype AS-IHP aimed at such commercial building applications. It is planned to start work on this effort in FY2016/17.

## ACKNOWLEDGEMENTS

The authors thank the members of the Nortek Global HVAC, LLC project team for their considerable contributions to the CRADA project and to this report. The authors also thank Mr. Antonio Bouza of the DOE Building Technology Office (DOE/BTO) for supporting ORNL's CRADA efforts under Contract No. DE-AC05-00OR22725 with UT-Battelle, LLC.

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## APPENDIX A - AS-IHP field test system monthly average field performance data

### A.1 Cooling season

**Table A1. Energy delivered to house each month by operating mode**

| | | May | June | July | Aug | Sept |
|---|---|---|---|---|---|---|
| SC | Ded SC | 466 | 1131 | 1524.63 | 1623.1 | 1148.7 |
| | SC+DS | 326 | 276 | 41.66 | 64.22 | 23.9 |
| | SC+WH (FC) | 85 | 120 | 118.96 | 143.47 | 143.5 |
| | **Total SC Energy Delivered** | **877** | **1527** | **1685** | **1831** | **1316.1** |
| WH | SC+DS | 3.22 | 4 | 2.97 | 4.32 | 0.0 |
| | SC+WH (FC) | 99 | 145 | 138 | 165 | 161.5 |
| | Ded. WH | 127 | 56 | 52 | 11 | 43.7 |
| | WH Element | 2.63 | 3.32 | 1.70 | 3.51 | 8.4 |
| | **Total WH Energy Delivered** | **231.67** | **209** | **194** | **184** | **214** |
| | **Total WH Energy Delivered w/o Element** | **229.0** | **205.3** | **192.6** | **180.7** | **205.2** |

**Table A2. Energy consumed each month by operating mode**

| | | May | June | July | Aug | Sept |
|---|---|---|---|---|---|---|
| SC | Ded SC | 87 | 222 | 298 | 317.34 | 211.9 |
| | SC+DS | 65 | 58 | 10 | 16 | 6.0 |
| | SC+WH (FC) | 17 | 25 | 24 | 28 | 28.2 |
| | **Total SC Energy Use** | **169** | **304** | **333** | **361** | **246.1** |
| WH | SC+DS | 0.65 | 0.89 | 0.73 | 1.07 | 0.0 |
| | SC+WH (FC) | 19.90 | 30 | 27 | 32 | 31.7 |
| | Ded. WH | 50.42 | 21 | 19 | 5 | 16.7 |
| | WH Element | 2.63 | 3.32 | 1.70 | 3.51 | 8.4 |
| | **Total WH Energy Use w/Element** | **73.60** | **55** | **48** | **42** | **57** |
| | **Total WH Energy Use w/o Element** | 70.97 | 51.62 | 46.58 | 38.02 | 48.44 |
| | **TOTAL Elec. Used (kWh)** | **243** | **359** | **381** | **403** | **303** |

**Table A3. Monthly average COPs by operating mode**

| | | May | June | July | Aug | Sept |
|---|---|---|---|---|---|---|
| SC | Ded SC | 5.36 | 5.09 | 5.11 | 5.11 | 5.42 |
| | SC+DS | 5.01 | 4.78 | 3.98 | 4.05 | 3.98 |
| | SC+WH (FC) | 4.96 | 4.89 | 4.96 | 5.09 | 5.09 |
| | Overall COP (Cooling) | 5.17 | 5.02 | 5.07 | 5.07 | 5.35 |
| WH | SC+DS | 4.95 | 4.78 | 4.07 | 4.04 | |
| | SC+WH (FC) | 4.96 | 4.89 | 5.04 | 5.09 | 5.09 |
| | Ded. WH | 2.52 | 2.67 | 2.80 | 2.52 | 2.62 |
| | Overall COP (WH) w/ Element | 3.15 | 3.80 | 4.02 | 4.44 | 3.76 |
| | **Overall COP (WH) w/o Element** | 3.23 | 3.98 | 4.13 | 4.75 | 4.24 |

**Table A4. Test month operating hours for each mode**

| | | May | June | July | Aug | Sept |
|---|---|---|---|---|---|---|
| **SC** | **Ded SC** | 90 | 213 | 276 | 291 | 207 |
| | **SC+DS** | 60 | 52 | 7 | 9 | 4 |
| | **SC+WH (FC)** | 14 | 21 | 20 | 24 | 24 |
| **WH** | **Ded.WH** | 16 | 7 | 8 | 2 | 7 |
| | **Refrigerant Mgmt.** | 2.18 | 1.96 | 2.07 | 2 | 3 |
| | **Off** | 560 | 424 | 432 | 416 | 477 |
| | **Total hrs in month** | 742 | 719 | 745 | 745 | 721 |
| | Average % on time (SC only) | 22% | 40% | 41% | 43% | 33% |

## A.2 Heating season

| | | Oct'14 | Nov'14 | Dec'14 | Jan'15 | Feb'15 | Mar'15 | Apr'15 |
|---|---|---|---|---|---|---|---|---|
| SH | Ded SH | 167 | 1,803 | 1,391 | 1,393 | 2,748 | 956 | 96.5 |
| | SH+DS | 21 | 303 | 252 | 385 | 634 | 201 | 25 |
| | **Total SH Energy Delivered** | **188** | **2,106** | **1,643** | **1,778** | **3,382** | **1,156** | **121** |
| SC | **Total SC Energy Delivered** | **182** | **0** | **0** | **0** | **0** | **0** | **0** |
| WH | **WH Element Delivered (ded. SH)** | **0.1** | 7.72 | 6.11 | 3.27474 | 4.95 | 4 | 0.02 |
| | SH+DS | 10 | 149 | 97 | 121 | 176 | 66 | 8.05 |
| | WH Element Delivered (SH+DS) | 0 | 9 | 7 | 20 | 20 | 9 | 0.00 |
| | Ded. WH | 208 | 166 | 124 | 72 | 65 | 249 | 293.33 |
| | WH Element Ded. WH) | 0.18 | 0.31 | 0.14 | 0.05 | 1.47 | 0.67 | 0.03 |
| | Off (WH Element) | 1.88 | 4.02 | 19.47 | 83.64 | 3.55 | 2.99 | 7.77 |
| | RMC (WH Element) | 0 | 0 | 0 | 0 | 0 | 0 | 0.00 |
| | Ded SC & SC+WH | 47 | 0 | 0 | 0 | 0 | 0 | 0 |
| | **Total WH Energy Delivered w/Element** | **268** | **335** | **254** | **300** | **270** | **332** | **309** |
| | **Total WH Energy Delivered w/o Element** | **266** | **314** | **221** | **193** | **240** | **315** | **301** |
| | | | | | | | | |
| | **Energy Delivered (kWh) w/Element** | **637** | **2442** | **1897** | **2078** | **3652** | **1488** | **430** |
| | **EnergyDelivered (kWh) w/o Element** | **635** | **2421** | **1864** | **1971** | **3622** | **1471** | **423** |

**Table A5. Energy delivered to house each month by operating mode**

**Table A6. Energy consumed each month by operating mode**

| | | Oct'14 | Nov'14 | Dec'14 | Jan'15 | Feb'15 | Mar'15 | Apr'15 |
|---|---|---|---|---|---|---|---|---|
| SH | **Ded SH** | 65 | 887 | 689 | 695 | 1533 | 399 | 30.24 |
| | **SH+DS** | 8.3 | 139 | 111 | 165 | 284 | 82 | 8.76 |
| | **Total SH Energy Use** | **73** | **1025** | **800** | **860** | **1818** | **482** | **39** |
| SC | **Total Energy Use** | **31.0** | **0** | **0** | **0** | **0** | **0** | |
| WH | **WH Element Use (ded. SH)** | 0.1 | 7.7 | 6.1 | 3.3 | 4.9 | 4 | 0.02 |
| | **SH+DS** | 4.2 | 68 | 43 | 52 | 79 | 27 | 2.85 |
| | **WH Element Use (SH+DS)** | 0.0 | 9 | 7 | 20 | 20 | 9 | 0.00 |
| | **Ded. WH** | 84.6 | 75 | 55 | 34 | 29 | 100 | 113.42 |
| | **WH Element (Ded. WH)** | 0.2 | 0 | 0 | 0 | 1 | 1 | 0.03 |
| | **Off (WH Element)** | 1.9 | 4 | 19 | 84 | 4 | 3 | 7.77 |
| | **RMC (WH Element)** | 0.2 | 0.0 | 0.0 | 0.0 | 0.1 | 0 | 0.00 |
| | **Total WH Energy Use w/Element** | 91 | 164 | 130 | 193 | 138 | 144 | 124 |
| | **Total WH Energy Use w/o Element** | 89 | 143 | 98 | 86 | 108 | 127 | 116 |
| | | | | | | | | |
| | **Energy Used (kWh) w/Element** | 164 | 1189 | 930 | 1054 | 1956 | 626 | 163 |
| | **Energy Used (kWh) w/o Element** | 162 | 1168 | 898 | 946 | 1926 | 609 | 155 |

**Table A7. Monthly average COPs by operating mode**

| | | Oct'14 | Nov'14 | Dec'14 | Jan'15 | Feb'15 | Mar'15 | Apr'15 |
|---|---|---|---|---|---|---|---|---|
| SH | Ded SH | 2.58 | 2.03 | 2.02 | 2.00 | 1.79 | 2.39 | 3.19 |
| | SH+DS | 2.48 | 2.18 | 2.26 | 2.33 | 2.23 | 2.45 | 2.82 |
| | Overall COP (Heating) | 2.57 | 2.05 | 2.05 | 2.07 | 1.86 | 2.40 | 3.11 |
| | SH+DS & WH Combined | 2.48 | 2.18 | 2.26 | 2.33 | 2.23 | 2.45 | 2.82 |
| WH | Ded. WH | 2.46 | 2.22 | 2.27 | 2.09 | 2.20 | 2.48 | 2.82 |
| | Overall COP (WH) w/ Element | 2.66 | 2.05 | 1.95 | 1.55 | 1.96 | 2.30 | 2.49 |
| | Overall COP (WH) w/o Element | 2.70 | 2.20 | 2.27 | 2.24 | 2.22 | 2.48 | 2.59 |

**Table A8. Test month operating hours for each mode**

| | **DURATION of MODE (hrs)** | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | | **Oct'14** | **Nov'14** | **Dec'14** | **Jan'15** | **Feb'15** | **Mar'15** | **Apr'15** |
| **SH** | **Ded SH** | 41 | 349 | 325 | 261 | 427 | 179 | 20 |
| | **SH+DS** | 4 | 55 | 61 | 72 | 115 | 28 | 3 |
| **Ded.WH** | | 38 | 35 | 26 | 16 | 14 | 47 | 52 |
| **Refrigerant Mgmt.** | | 3 | 2 | 2 | 1 | 1 | 3 | 2 |
| **Off** | | 612 | 282 | 231 | 342 | 115 | 436 | 615 |
| **Total hrs in month** | | 729 | 723 | 645 | 693 | 673 | 693 | 693 |
| **Avg % ontime (SH only)** | | **6%** | **56%** | **60%** | **48%** | **81%** | **30%** | **3%** |

**Table A9. Breakdown of Auxiliary heat use in SH and in SH+DS modes**

| | Energy Delivered (kWh) | | Energy Used (kWh) | | Aux. Energy (kWh) | | Aux. Energy (kWh) | |
|---|---|---|---|---|---|---|---|---|
| | SH | SH+DS | SH | SH+DS | SH | SH+DS | (%) SH | (% SH+DS) |
| **Oct** | 167 | 21 | 65 | 8.3 | 8.91 | 0.011 | 5.33% | 0.05% |
| **Nov** | 1,803 | 303 | 887 | 139 | 211.24 | 22.17 | 11.71% | 7.32% |
| **Dec** | 1,391 | 252 | 689 | 111 | 157.54 | 13.8 | 11.33% | 5.48% |
| **Jan** | 1,393 | 385 | 695 | 165 | 142.38 | 30.65 | 10.22% | 7.96% |
| **Feb** | 2,748 | 634 | 1533 | 284 | 556.3 | 69.16 | 20.24% | 10.91% |
| **Mar** | 956 | 201 | 399 | 82 | 73 | 15 | 7.61% | 7.57% |
| **Apr** | 96.5 | 25 | 30.24 | 8.76 | 1.156 | 0.034 | 1.20% | 0.14% |
| **Total** | 8,554 | 1,820 | 4,298 | 799 | 1,150 | 151 | | |

## APPENDIX B – Invention Disclosures Filed under CRADA Work Program

This appendix lists invention disclosures resulting from work done under this CRADA project.

1. Joint disclosures by Nortek Global HVAC, LLC and ORNL – none

2. Disclosures by ORNL – Invention Disclosure 201303204, DOE S-124,795, "Refrigerant Charge Management in an Integrated Heat Pump"

3. Disclosures by Nortek Global HVAC LLC – Invention Disclosure "Refrigerant Management for Multi-purpose Heat Pump Water Heater (MHPWH)" dated 12/01/2011. As results of this invention disclosure, 1) one patent titled "Refrigerant Charge Management in a heat pump water heater" (patent No.: US 2013/0160985 A1) was awarded by US patent office. 2) One CIP (Continuation in Part) titled: "Refrigerant Charge Management in a heat pump water heater" (Patent application No.: 14/278,982) is pending.