Development of an Air-Source Heat Pump
Integrated with a Water Heating / Dehumidification Module

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Abstract: A residential-sized dual air-source integrated heat pump (AS-IHP) concept is under development in partnership between Oak Ridge National Laboratory (ORNL) and a manufacturer. The concept design consists of a two-stage air-source heat pump (ASHP) coupled on the air distribution side with a separate novel water heating/dehumidification (WH/DH) module. The motivation for this unusual equipment combination is the forecast trend for home sensible loads to be reduced more than latent loads. Integration of water heating with a space dehumidification cycle addresses humidity control while performing double-duty. This approach can be applied to retrofit/upgrade applications as well as new construction. A WH/DH module capable of ~1.47 L/h water removal and ~2 kW water heating capacity was assembled by the manufacturer. A heat pump system model was used to guide the controls design; lab testing was conducted and used to calibrate the models. Performance maps were generated and used in a TRNSYS sub-hourly simulation to predict annual performance in a well-insulated house. Annual HVAC/WH energy savings of ~35% are predicted in cold and hot-humid U.S. climates compared to a minimum efficiency baseline.

Key Words: heat pump, water heating, dehumidification, energy savings

1   INTRODUCTION AND BACKGROUND

Oak Ridge National Laboratory (ORNL) and Lennox Industries, Inc. have been engaged in a collaborative research and development agreement (CRADA) to develop an air-source integrated heat pump (AS-IHP) product for the US market based on an IHP concept developed by ORNL for the U.S. Department of Energy (DOE) (Murphy et al. 2007). Lennox's specific embodiment of the IHP concept (Uselton 2012) is a two-capacity, "two-unit system for possible introduction to the U.S. market as early as 2015-2016. This IHP system concept is a combination of a high-efficiency heat pump already marketed by Lennox to provide HVAC services together with a separate prototype equipment module for water heating (WH) and demand dehumidification (DH) services – a WH-DH module. Figure 1 shows a schematic of the concept. The WH-DH module is integrated with the central heat pump unit by a parallel secondary duct loop around the central air handler, receiving a portion of the central return air when the secondary (WH-DH) blower is operating and returning this air to the supply side. It also has an optional connection to an outdoor air intake to provide a means for conditioning and circulating this air through the central duct system.

The motivation for this unusual equipment combination is the forecast trend for lower residential sensible cooling loads leading to lower sensible heat ratios (SHR). Homes with reduced outdoor air infiltration and more thermal insulation have lower sensible loads but internal moisture loads from showers, laundry and cooking can become a problem. A dedicated space dehumidification cycle addresses humidity control and integration of heat pump water heating is expedient since the small vapour compression components can perform double-duty. The integrated yet independent operation of the WH-DH unit provides dehumidification of the central return and ventilation air as well as a central heat source for the water heating mode. The independent operation is especially useful during the shoulder months when WH loads and, in many cases DH loads, exist but sensible cooling and heating loads are small. Another significant advantage is that this approach can be applied to retrofit/upgrade applications as well as new construction, utilizing standard electric water heaters and a wide range of multi-capacity and variable speed heat pumps.
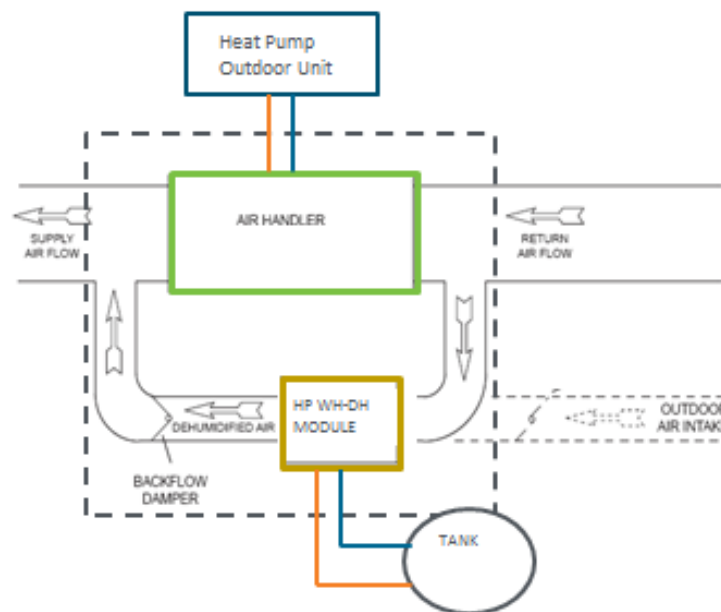

Figure 1 – Two-Unit AS-IHP Concept Schematic

Design performance goals for the WH-DH unit are to meet or exceed Energy Star performance levels for WH and DH modes of operation. For the DH mode, the Energy Factor (EF) requirement for Energy Star rating (Energy Star 2012) is >1.85 L/kWh for units with DH capacity <1.48 L/h (75 pints/day). This capacity was determined adequate for the climates and homes modeled in this paper – see Section 4. For the WH mode, an EF ≥2.0 (W/W) is required for Energy Star designation (Energy Star 2013). The remaining design goal was to provide water heating capacity of ~2 kW, about twice that for standalone residential heat pump water heaters (HPWHs). Initial unpublished analysis indicated that this AS-IHP approach has the potential to exceed 40% annual energy savings for such an integrated system versus a baseline suite of minimum efficiency all electric equipment providing HVAC, DH and WH services.

2   PROTOTYPE WH-DH MODULE DEVELOPMENT

A prototype design suitable for lab testing was assembled by the manufacturer, starting from a whole house dehumidifier unit. The prototype uses an R-410A rotary compressor rated at about 2 kW (7000 Btu/h) and 2.8 COP (9.5 Btu/Wh EER) under air-conditioning conditions. Separate condensers are used for each operating mode -- a 3.5 kW (1-ton) fluted tube-in-

tube double-walled water-to-refrigerant heat exchanger (HX) and a three-row fin-and-tube air-to-refrigerant HX, in combination with a common two-row fin-and-tube evaporator. Figure 2 shows the tube-in-tube HX unit and a cutaway of the fluted tube design, where water flows through the inner fluted tube and refrigerant flows through the fluted annulus.

Figure 2. Fluted Tube-in-Tube Water-to-Refrigerant HX

This HX was installed around the rotary compressor as can be seen in the CAD drawing of Figure 3.
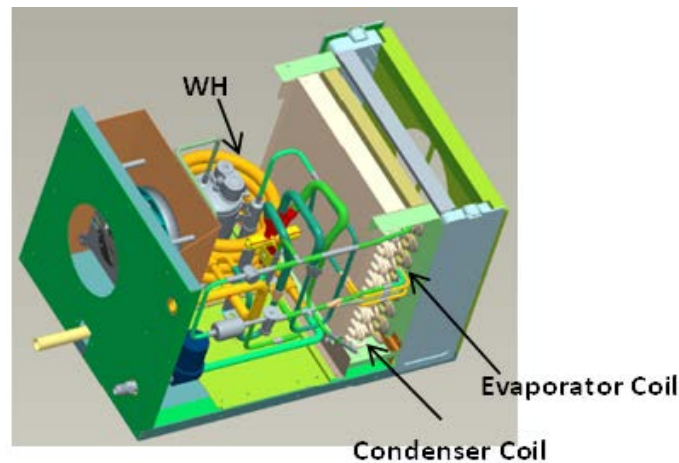

Figure 3. CAD Drawing of Prototype WH-DH Module Layout

Refrigerant-side schematics of the design in the two operating modes are shown in Figure 4. The switch-over valve shown in the drawing is used to switch active condensers between modes; the inactive suction port side is also used as a vent to return refrigerant from the inactive condenser. A thermostatic expansion valve is used to regulate refrigerant flow based upon evaporator superheat. A draw-through backward-curved centrifugal blower and high-efficiency water pump, both with variable-speed brushless permanent magnet (VS-BPM) motors completes the major components list.

A detailed heat pump design model, HPDM (Rice and Jackson 2005) was used in two setup configurations to model the prototype design and determine predicted optimal refrigerant, air, and water control settings for the two operating modes. The fluted-tube water-to-refrigerant model in the HPDM requires internal geometry specifications (obtained by direct measurements of the cutaway section shown in Figure 2). 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. Details of the air-to-refrigerant HXs as well as compressor, blower, and pump performance maps were provided by their respective manufacturers. The manufacturers' data in conjunction with the HPDM were used to predict the optimal refrigerant subcooling levels and air and water flows at the selected design conditions, which were entering air conditions of 26.7°C (80°F)/60%RH for dehumidification and 19.7°C (67.5°F) / 50%RH for the water heating with a 42.2°C (108°F) entering water temperature. The water flow for the WH

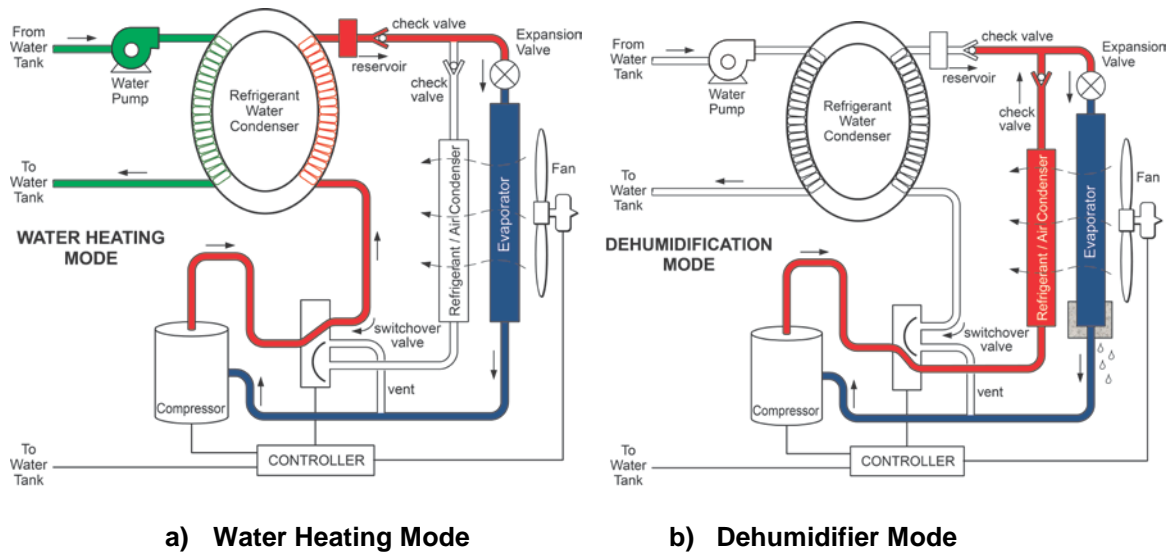mode was set at the maximum available flow of 0.145 L/s (2.3 gpm) for an expected static pressure head curve.

a) **Water Heating Mode**   b) **Dehumidifier Mode**

Figure 4. Refrigerant-Side Design in DH and WH Modes

For the WH mode, the optimum COP was centered around 5.6°C (10°F) subcooling and 142 L/s (300 cfm) as shown in Figure 5a by the bold **X**. For the DH mode, the design goal of <1.48 L/h (75 pints/day) and Energy Star efficiency could be achieved with 5.6°C (10°F) subcooling and 142 L/s (300 cfm) air flow as denoted in Figure 5b by the bold **X**. This design subcooling level and related charge quantity (lower than for a typical DH design) was a compromise to be compatible with efficient operation in water heating mode.
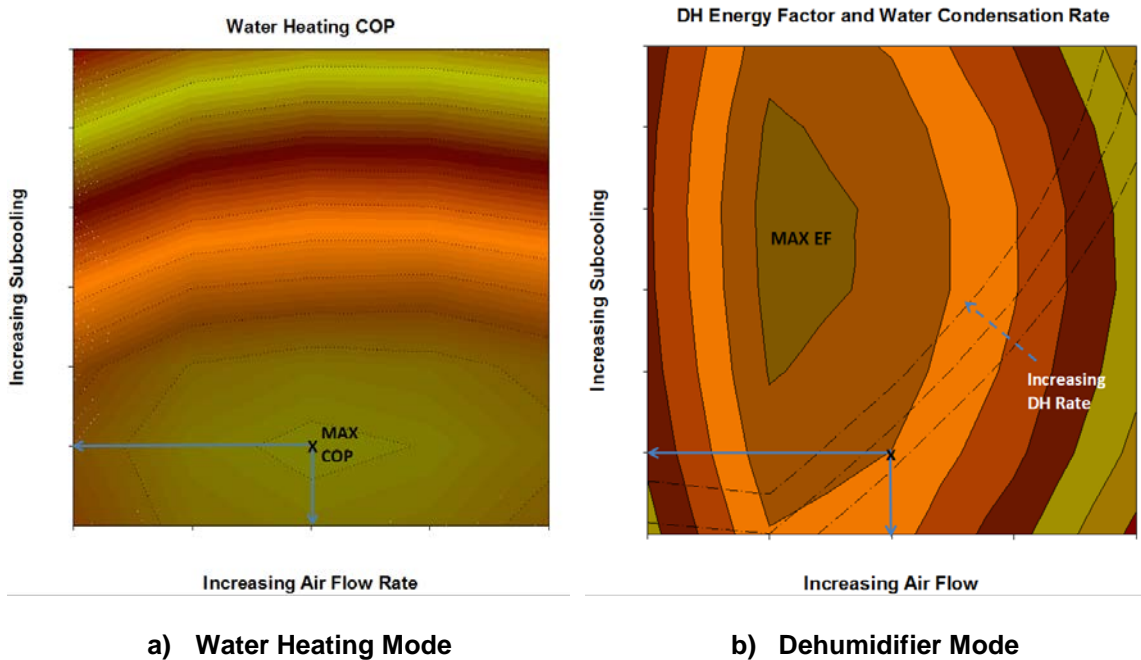

a) **Water Heating Mode**   b) **Dehumidifier Mode**

Figure 5. Performance Parametrics of Indoor Airflow and Condenser Subcooling

3   WH-DH MODULE TESTING

Following in-situ blower and operational unit testing, Lennox shipped the prototype WH-DH module to ORNL for detailed laboratory testing and performance mapping. Figure 6 shows a photo of the unit (with covers removed to show the interior component arrangement).

The WH-DH unit was tested in a small environmental chamber along with a nominal 190 L (50 gal) water tank (with actual volume 9% less) connected by insulated water hoses. Unit inlet relative humidity (RH) was measured using a RH transducer with the inlet dry-bulb (DB) temperature obtained from a 4-point averaging Type-T thermocouple (TC) grid. The WH-DH air flow rate was not measured directly; it was controlled to ~142 L/s (300 cfm) under wet coil conditions based on fan external static pressure drop data obtained from in-situ fan tests conducted by Lennox. A short length of round duct with an adjustable damper was installed upstream to regulate the external static pressure head on the WH-DH blower.
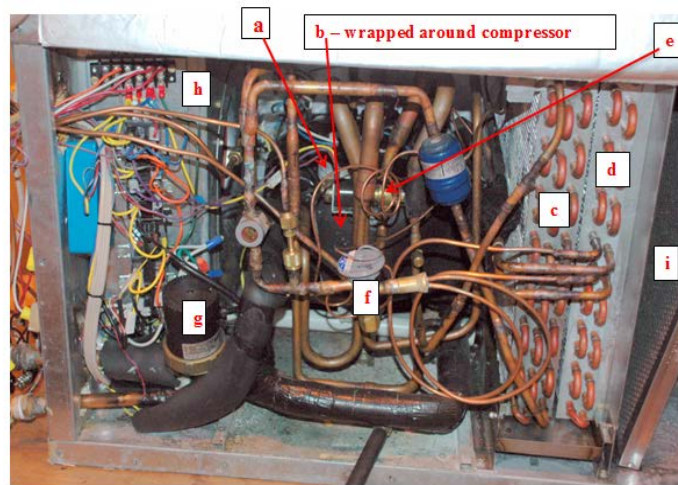

**Figure 6 – WH-DH Module as Received with Cover Removed: A) Compressor;
B) WH Mode Condenser; C) DH Mode Condenser; D) Evaporator;
E) Mode Switch Valve; F) Thermal Expansion Valve & Distributor;
G) Water Pump; H) Blower Housing; and I) Air Filter.**

Note however that the measurements of the WH and DH capacities are not dependent on knowledge of the airflow rate. To determine DH capacity, the condensate collected for the duration of each test period was weighed using an electronic balance scale. Water flow was measured with a turbine flow meter installed in the WH-DH entering water line, with 5 straight pipe diameters upstream and downstream of the meter. Water flow was controlled to ~0.132 L/s (2.1 gpm), the maximum obtainable in the test loop. Refrigerant and water line temperatures were measured with Type T TCs, all surface mounted and insulated except for the tank water temperatures which were obtained with immersion TCs. Refrigerant pressures were measured with transducers on the compressor suction, discharge, and liquid line lines, with ranges of 0-1.72 MPa (0-250 psia) on the low side and 0-5.17 MPa (0-750 psia) on the high side. Refrigerant flow rate was calculated from the manufacturer's compressor map using the measured suction and discharge pressures and the suction superheat.

A steady-state test matrix was developed for WH and DH performance mapping tests over a range in inlet air conditions and water temperatures. As indicated in Table 1, for the DH mode, three ambient temperatures and three RH were used giving nine test points to span the range of expected return air conditions. In the WH mode, twelve test points were run, using the three most likely indoor DB/RH conditions in combinations with four entering water temperatures (EWTs).

The steady-state WH and DH tests were run for a minimum of 30 minutes and 1 hour, respectively. In addition to the performance mapping, tests were conducted to estimate the EF (24 hour duration) and 1st hour ratings for the WH mode and the EF rating (6 hour duration) for the DH mode using the standard rating test procedures for each mode [US CFR 2010 , for WH; AHAM 2008, for DH].

Table 1. Inlet Air and Water Conditions for DH and WH Steady-State Testing of WH-DH Unit

Test Matrix for DH Mode			Test Matrix for WH Mode			
Test #	Test Condition		Test #	Test Condition		
	Inlet DB, °C	RH %		Inlet DB, °C	RH %	EWT °C
1	26.7	60	1	20.0	50	21.1
2	26.7	55	2	20.0	50	32.2
3	26.7	50	3	20.0	50	43.3
4	23.3	55	4	20.0	50	54.4
5	23.3	60	5	23.3	55	21.1
6	23.3	50	6	23.3	55	32.2
7	20.0	60	7	23.3	55	43.3
8	20.0	55	8	23.3	55	54.4
9	20.0	50	9	26.7	60	21.1
			10	26.7	60	32.2
			11	26.7	60	43.3
			12	26.7	60	54.4

Steady-state DH tests were conducted first to establish the required design charge at DH rating conditions and ~10°F condenser subcooling and to determine the dehumidification capacity. After determining the refrigerant charge needed to achieve the desired superheat and subcooling control, a 6 hour DH standard rating test was run. This confirmed that the capacity was just below the 1.48 L/h (75 pints/day) target with an EF above 2, exceeding the 1.85 Energy Star minimum for this size dehumidifier. We recorded the condensate amount at the end of each hour of the test which gave hourly measurements with a maximum deviation of 3.5%. Following this, one hour steady-state tests were run for each of the nine inlet air condition combinations in Table 1.

Initial steady-state water heating tests followed using the same refrigerant charge as for the DH testing. These test results showed somewhat lower WH capacities and COPs than predicted from the simulation. From the refrigerant- and water-side energy flows, we determined that, for the higher EWTs, there was significant heat loss occurring from the refrigerant in the outer annulus of the HX to the cool air stream leaving the evaporator.

Next, a baseline WH mode EF for the "as received" unit was obtained based on the standard 24-h use test procedure [US CFR 2010]; a value of ~1.5 (lower than expected) indicated that the heat losses within the unit were significant. In an attempt to minimize the heat losses and improve the WH performance, further insulation was added to better insulate and isolate the WH condenser and the compressor from the exiting cold air stream leaving the evaporator.

After these changes, a new set of steady-state WH data was taken and compared to the earlier results as shown in Figure 7. This shows capacity as a function of EWT for three different inlet air conditions. From the capacity plot, it is clear that isolation of the WH condenser and compressor from the exit air stream significantly boosted the delivered capacity at the higher EWTs as compared to the initial tests (marked "Non" in the legends).

WH Capacity versus EWT, Isolated and Non-Isolated HX

Legend:
- ■ 26.7 C, 60% RH
- □ 26.7 C, 60%RH, Non
- ● 23.3 C, 55% RH
- ○ 23.3 C, 55% RH, Non
- ◆ 20 C, 50% RH
- △ 20 C, 50% RH, Non

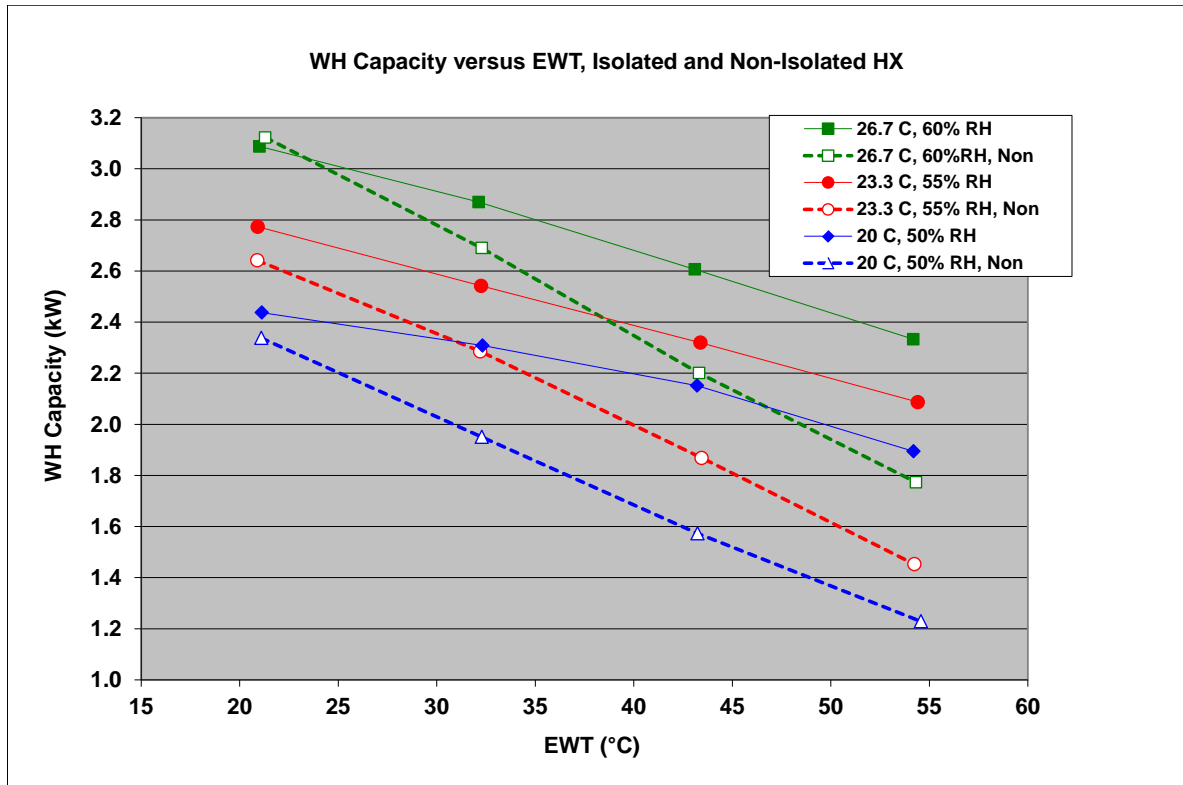Y-axis: WH Capacity (kW)
X-axis: EWT (°C)

Figure 7 – Steady-State WH Capacity Test Results – Showing Performance Improvement After Isolating Condenser with Insulation

The WH mode EF test was then repeated, this time with a lower WH thermostat setting to keep the maximum tank water temperature from exceeding 56.7°C (135°F) as was the case in the first tests. The new EF results improved to ~1.65, but still well below the 2.0 target. Additional insulation was applied to the condenser and compressor along with extra insulation on the refrigerant and water lines inside the unit. A new 24-h use (EF) test followed with a resultant EF of 1.78, an increase of 7.8% from the previous test, but still short of the target level.

Next, from close examination of the tank inlet and exit temperatures during the tank cool-down period, we determined that there were three apparent thermosiphon events in which water that had cooled in the WH-mode condenser flowed back into the bottom of the tank, displacing warmer water in the top of the tank into the WH-DH unit . Note in Figure 8 the 3-4 events where the WH water inlet temperature jumped sharply during the standby period after the 6th water draw of the 24-h test.

On discovering this, we ran a new cool-down test with the water inlet valve closed to prevent the thermosiphon action. From this we determined that reducing this recurrent heat loss would eliminate the need for any WH-DH unit operation for tank reheat during the standby period of the 24-h EF test and estimated that the resultant EF would have been ~1.9. We repeated the WH EF test after making this adjustment, manually closing the valve in the inlet water line to prevent any thermosiphon incidents. The resultant EF was found to be 1.92, ~8% higher than that from the previous test – much closer to but still short of the target. In summary, the EF was increased from 1.5 to 1.92 over the course of the testing for a 28% increase. WH mode first hour recovery rating tests were also run; achieving an acceptable 223 L (59 gallons) delivered.
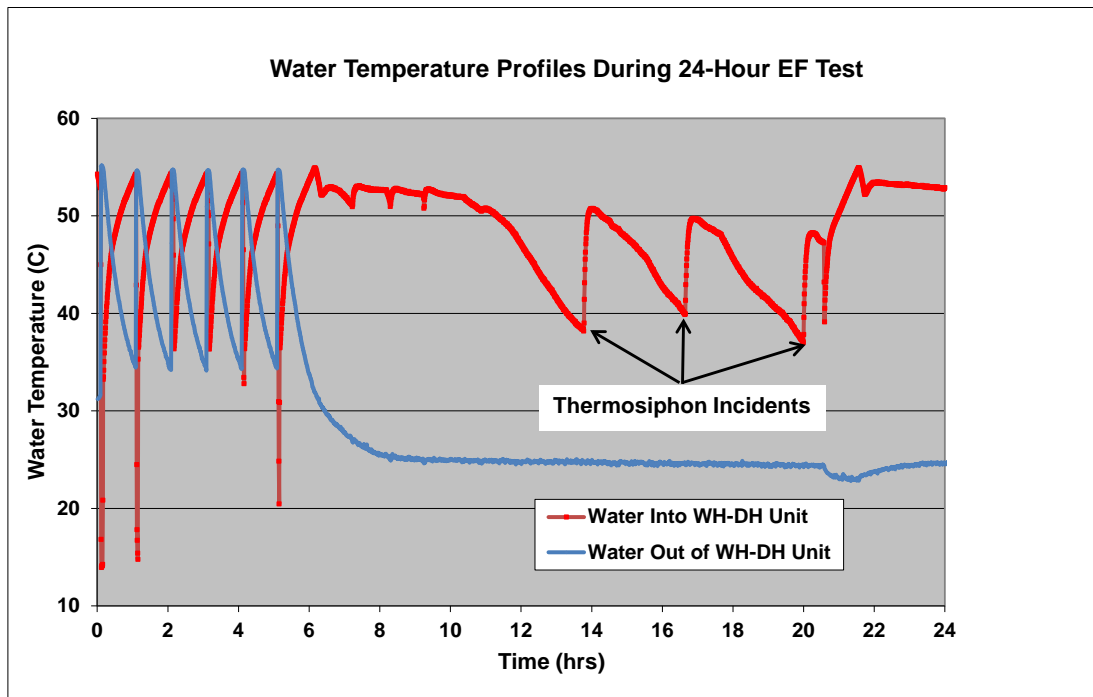
Water Temperature Profiles During 24-Hour EF Test

**Figure 8 – WH-DH Unit Water Inlet and Outlet Temperatures (TC1 and TC2)
During WH mode EF Test**

The final WH COP values versus EWT are shown in Figure 9 for the three indoor conditions. While the WH EF performance was increased substantially from the initial tests, the unit is still losing up to 23% of the available WH energy at 54.4°C (130°F) EWT. It is expected that insulation applied at the factory could reduce this loss by about half and enable achieving the EF target of ≥2.0.
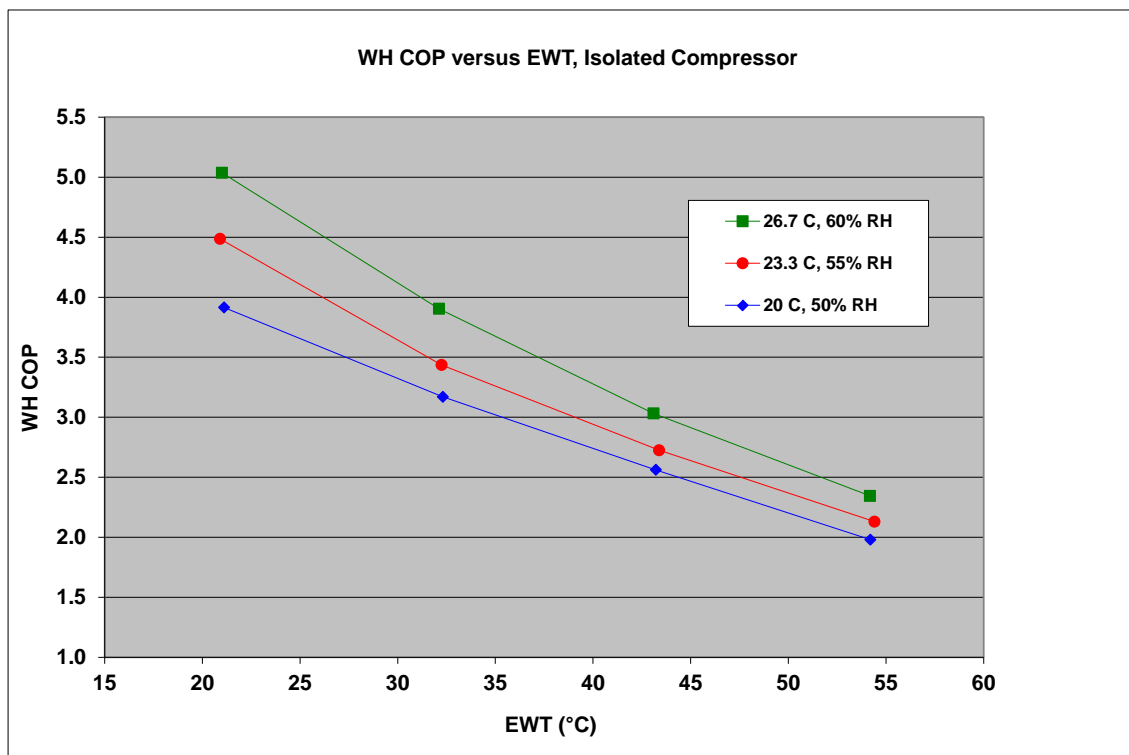
WH COP versus EWT, Isolated Compressor

Figure 9 – Final Steady-State WH COP Test Results – Prototype 1

4   ANNUAL ENERGY USE ANALYSIS AND SAVINGS PREDICTIONS

The steady-state DH and WH data were used to calibrate the HPDM for use in generating WH-DH unit performance maps for each mode. These maps were input to a customized project in the TRNSYS (Solar Energy Lab 2010) annual simulation model to estimate the expected energy savings of the Lennox two-unit AS-IHP prototype design. For the two-capacity central heat pump, we used a nominal 7 kW (2-ton) performance map with a brushless permanent magnet (BPM) fan motor air handler (Lennox 2009a, b); the rated seasonal cooling performance factor (CSPF) was 5.4 W/W (SEER of 18.4 Btu/Wh) with a heating season performance factor (HSPF) of 2.67 W/W (HSPF of 9.1 Btu/Wh). The TRNSYS simulations used a 3-minute time step and assumed a 243 L/d (64.3 gal/d) hot water load. Simulations were run for three Building America climate regions (U.S. DOE 2013) of mixed-humid, hot-humid, and cold in a 242 m^2 (2600 ft^2) tight, well insulated two-story house. A minimum efficiency all-electric system was also simulated in TRNSYS for a baseline. This included a 3.8 CSPF (13 SEER), 2.3 HSPF (7.7 HSPF Btu/Wh) heat pump, a 0.90 EF water heater and 1.4 EF dehumidifier. For the baseline system, ventilation air was drawn in from the bathroom fans while for the AS-IHP this air was supplied on the return side of the duct system as shown in Figure 1.

Previous unpublished TRNSYS simulations were performed at the optimal 142 L/s (300 cfm) for the WH-DH unit with operation allowed at all indoor blower speeds. However the Lennox's blower test results for the prototype WH-DH module indicated that it could only achieve 113 L/s (240 cfm) airflow when the central (HVAC) heat pump blower was operating at low speed. At high speed blower operation the static pressure head on the WH-DH unit would be too high to achieve acceptable air flow. Accordingly, the TRNSYS simulations reported below were made with performance maps for 113 L/s airflow and WH-DH operation was not allowed in conjunction with high-speed central heat pump operation. The average reductions in WH capacity and COP at the lower airflow rate were 3.2 and 3.4%, respectively, compared to the same values for 142 L/s air flow. The DH EF and water removal values at the rating point dropped 1.2% and 6.5%, respectively.

Comparisons of predicted energy use and savings between the baseline suite and the reduced flow case are shown in Table 2 for each mode and overall. The total predicted HVAC/WH energy savings for the reduced airflow and operation assumptions range from 33 to 36%. The entries in red show the portion of the total energy use for that mode that was from resistance heat. The net space conditioning savings for the three cases for the AS-IHP combination range from 23% for Chicago to 25% for Houston. The space heating savings are reduced by the cooling effect of the WH-DH unit when in WH mode in the winter months while the space cooling energy savings are enhanced. Predicted water-heating-only energy savings ranged from 50% in Chicago to 59% in Houston for the nominal WH set point of 48.9°C (120°F).

The annual electrical energy that is required to provide space conditioning, active dehumidification, water heating, and ventilation for these energy efficient homes is modest. The web-based program, PVWatts (Dobos 2013) can be used to size a solar photovoltaic array to provide a specified energy requirement at a particular geographic location. A commercially available heat pump (Lennox 2013a, b), compatible with this system, is able to accept solar PV as a second power source. We investigated how many 275 dc watt solar modules would be needed to offset this annual electrical energy requirement for each city. For Atlanta and Houston 13 modules and 15 modules, respectively, should be adequate to supply the annual electric power needs of the AS-IHP system. For Chicago, the maximum of 16 solar modules would still leave a shortfall of 2157 kWh.

**Table 2. Energy Use and Savings Predictions for AS-IHP
With Reduced Flow WH-DH Unit Configuration**

Energy Use by Mode; 242 m² Tight, Well-Insulated House			
	1-Speed Base	**2-Speed w WH-DH Unit, 113 L/s**	
Operation Mode	**Energy Use kWh (I^2R)**	**Energy Use kWh (I^2R)**	**Reduction from Base (%)**
Atlanta			
space heating	2311	1965	15.0%
resistance heat	(18)	(31)	
space cooling	1741	1059	39.2%
water heating	3380	1553	54.1%
resistance heat	(3380)	(488)	
dedicated DH	319	299	6.2%
ventilation fan	189	202	-6.9%
totals	7941	5079	36.0%
Houston			
space heating	995	906	9.0%
resistance heat	(0)	(3)	
space cooling	3035	1975	34.9%
water heating	2813	1169	58.5%
resistance heat	(2813)	(246)	
dedicated DH	1154	1035	10.3%
ventilation fan	189	179	5.6%
totals	8187	5264	35.7%
Chicago			
space heating	6214	4915	20.9%
resistance heat	(916)	(669)	
space cooling	740	402	45.6%
water heating	4218	2122	49.7%
resistance heat	(4218)	(906)	
dedicated DH	154	154	0.0%
ventilation fan	189	169	10.5%
totals	11514	7762	32.6%

5   CONCLUSIONS AND RECOMMENDATIONS

In summary, a novel prototype WH-DH unit was developed and tested, providing the basis for calibrated model performance maps. The performance maps were used to predict potential annual energy savings of ~35% for a two-unit AS-IHP system in three U.S. climates, ranging from mixed-humid, hot-humid, to cold. The dehumidifier performance goals were met and the water heating EF goal was approached - falling slightly short due to internal heat losses at high EWTs.

Work is underway on a revised prototype design. Improvements are being made to boost the airflow capability of the unit against expected external static heads. A brazed-plate water-to-refrigerant HX is replacing the tube-in-tube design to provide a lighter weight, more compact and easily insulated design. Modifications also include changes to increase the evaporator surface utilization. A charge reservoir to store excess charge and increase performance in the WH mode is also being considered.

With these improvements in combination with the highest efficiency Lennox multi-speed central heat pump units, we expect to reach or exceed a target average of 40% energy savings in suitable U.S. climates. Field tests of the revised prototype are planned for the 2014-2015 cooling and heating seasons, respectively.

6   REFERENCES

ANSI/AHAM DH-1 2008. *Dehumidifiers*, Association of Home Appliance Manufacturers, Washington, DC.

Dobos, Aron P. 2013. *PVWatts Version 1 Technical Reference*, National Renewable Energy Laboratory, Technical Report NREL/TP-6A20-60272, October. http://rredc.nrel.gov/solar/calculators/PVWATTS/version1

Energy Star 2012. ENERGY STAR® Program Requirements for Dehumidifiers**, U. S. Environmental Protection Agency Energy Star Program, http://www.energystar.gov/products/specs/system/files/ES_Dehumidifiers_Final_V3.0_Program_Requirements.pdf

Energy Star 2013. ENERGY STAR® Program Requirements for Residential Water Heaters**, U. S. Environmental Protection Agency Energy Star Program, https://www.energystar.gov/products/specs/sites/products/files/ENERGY%20STAR%20Water%20Heaters%20V2%200%20Program%20Requirements.pdf

U.S. Department of Energy, 2013. *Building Science-Based Climate Maps.* PNNL-SA-90570, January.http://apps1.eere.energy.gov/buildings/publications/pdfs/building_america/4_3a_ba_innov_buildingscienceclimatemaps_011713.pdf

Lennox Industries Inc. 2013a. SunSource® Home Energy System Product Specifications, Publication #210664, Sept.

Lennox Industries Inc. 2013b. SunSource® Home Energy System Application and Design Guidelines, Publication #CORP1312-L2, July 2013

Lennox Industries Inc. 2009a. "Lennox Engineering Data, Heat Pump Outdoor Units, XP19", Bulletin No. 210415, March.

Lennox Industries Inc. 2009b,."Lennox Engineering Data, Heat Pump Outdoor Units, XP19, Expanded Ratings Tables", Bulletin No. 210415R, April.

Murphy, R. W., C. K. Rice, V. D. Baxter, and W. G. Craddick 2007. *Air Source Integrated Heat Pump for Near-Zero Energy Houses: Technology Status Report*, ORNL/TM-2007/112, Oak Ridge National Laboratory, July.

Rice, C. K. and W. L. Jackson 2005. DOE/ORNL Heat Pump Design Model on the Web, Mark VII Version. http://www.ornl.gov/~wlj/hpdm/MarkVII.shtml

Solar Energy Laboratory (Univ of WI), TRANSSOLAR Energietechnik, CSTB – Centre, Scientifique et Technique du Bâtiment, and TESS – Thermal Energy System Specialists 2010. TRNSYS 16: a TRaNsient SYstem Simulation program, Version 16.01.0000

US Code of Federal Regulations. 2010. "Uniform Test Methods for Measuring the Energy Consumption of Water Heaters," *Code of Federal Regulations*, Title 10, Chapter II, Volume 3, Part 430, Subpart B, Appendix E.

Uselton, Robert 2012. Dedicated Dehumidifier and Water Heater, United States Patent Application Publication, Pub. No. US 2012/0047930 A1, March 1.