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Ground Source Integrated Heat Pump (GS-IHP) Development

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CRADA NFE-07-01000
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Ground Source Integrated Heat Pump (GS-IHP) Development – Final Report

Van Baxter, Keith Rice, Richard Murphy (ret), Jeff Munk, Moonis Ally, Bo Shen
William Craddick

Executive Summary

Between October 2008 and May 2013 ORNL and ClimateMaster, Inc. (CM) engaged in a Cooperative Research and Development Agreement (CRADA) to develop a ground-source integrated heat pump (GS-IHP) system for the US residential market. A initial prototype was designed and fabricated, lab-tested, and modeled in TRNSYS (SOLAR Energy Laboratory, et al, 2010) to predict annual performance relative to 1) a baseline suite of equipment meeting minimum efficiency standards in effect in 2006 (combination of air-source heat pump (ASHP) and resistance water heater) and 2) a state-of-the-art (SOA) two-capacity ground-source heat pump with desuperheater water heater (WH) option (GSHPwDS). Predicted total annual energy savings, while providing space conditioning and water heating for a 2600 ft^2 (242 m^2) house at 5 U.S. locations, ranged from 52 to 59%, averaging 55%, relative to the minimum efficiency suite. Predicted energy use for water heating was reduced 68 to 78% relative to resistance WH. Predicted total annual savings for the GSHPwDS relative to the same baseline averaged 22.6% with water heating energy use reduced by 10 to 30% from desuperheater contributions.

The 1st generation (or alpha) prototype design for the GS-IHP was finalized in 2010 and field test samples were fabricated for testing by CM and by ORNL. Two of the alpha units were installed in 3700 ft^2 (345 m^2) houses at the ZEBRAlliance site in Oak Ridge and field tested during 2011. Based on the steady-state performance demonstrated by the GS-IHPs it was projected that it would achieve >52% energy savings relative to the minimum efficiency suite at this specific site. A number of operational issues with the alpha units were identified indicating design changes needed to the system before market introduction could be accomplished. These were communicated to CM throughout the field test period.

Based on the alpha unit test results and the diagnostic information coming from the field test experience, CM developed a 2nd generation (or beta) prototype in 2012. Field test verification units were fabricated and installed at the ZEBRAlliance site in Oak Ridge in May 2012 and at several sites near CM headquarters in Oklahoma. Field testing of the units continued through February 2013.

Annual performance analyses of the beta unit (prototype 2) with vertical well ground heat exchangers (GHX) in 5 U.S. locations predict annual energy savings of 57% to 61%, averaging 59% relative to the minimum efficiency suite and 38% to 56%, averaging 46% relative to the SOA GSHPwDS. Based on the steady-state performance demonstrated by the test units it was projected that the 2nd generation units would achieve ~58% energy savings relative to the minimum efficiency suite at the Zebra Alliance site with horizontal GHX.

A new product based on the beta unit design was announced by CM in 2012 – the Trilogy 40® Q-mode™ (http://cmdealernet.com/trilogy_40.html). The unit was formally introduced in a March 2012 press release (see Appendix A) and was available for order beginning in December 2012.

Introduction

In FY2008, this Cooperative Research and Development Agreement (CRADA) between UT-Battelle, LLC (ORNL) and ClimateMaster Incorporated (CM) was initiated to conduct the research and development needed to support development of a new residential HVAC & water heating (WH) product - a ground-source integrated heat pump (GS-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. A major need to enable achieving this vision is deep reduction of the energy used by the energy service equipment (equipment providing space heating and cooling, water heating, etc.) - 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. CM expressed interest in the IHP concept and agreed to partner with ORNL in this CRADA, specifically for the GS-IHP which uses the ground as its ultimate heat source and sink.

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

Background – GS-IHP concept development

Full details of the GS-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) uses one variable-speed (VS) modulating compressor, a VS indoor blower, a VS pump for ground heat exchanger (GHX) fluid circulation, and a single-speed pump for hot water circulation. A 50 gallon (~189 l) WH tank is included. The concept development analyses reported in Murphy et al (2007) included a dedicated dehumidification mode and a humidifier option (neither included in CM's initial prototype system described later). In addition those analyses were based on a relatively small (1800 ft^2, 167 m^2) and very well insulated house with nominal cooling design loads of 1-1.5 tons (3.5-5.3 kW) depending upon location. The CM development is a 2-ton (7 kW) nominal size designed for similarly well insulated but larger residences (described in the next section).

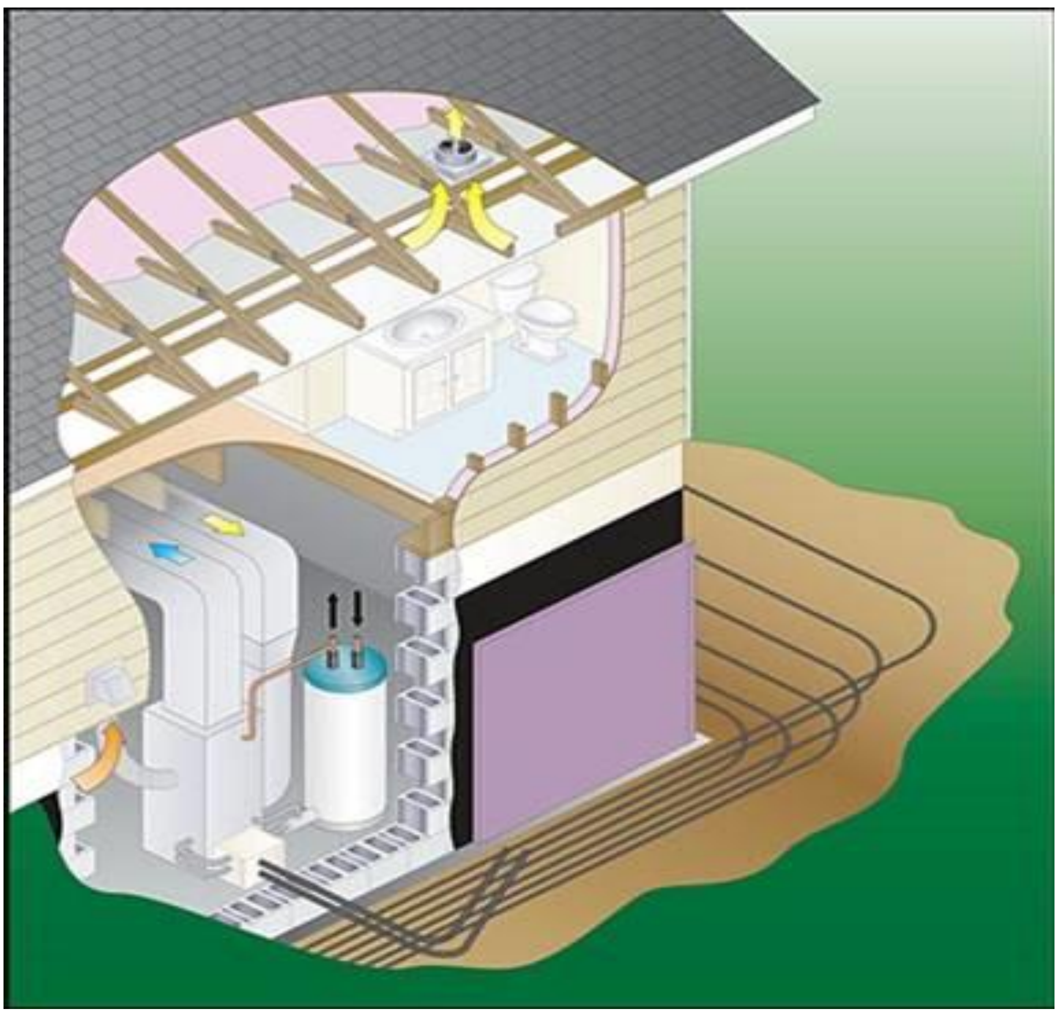

Fig. 1. Conceptual installation of the residential ground-source integrated heat pump.

Annual energy use simulations for a baseline suite of individual systems (13 SEER 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 GS-IHP were performed using the TRNSYS 16 platform (Solar Energy Laboratory, et al. 2010). A vertical ground heat exchanger (GHX) loop consisting of two parallel u-tube pipe loops was assumed for the GS-IHP. Annual, sub hourly simulations were performed for the baseline system and 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 ~65 gallons (~245 l) was assumed on 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 RH ≤60% in summer, fall, and winter; 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 provides summary results of TRNSYS/HPDM sub-hourly simulations for the baseline HVAC system for each of the five locations examined in this study. Table 3 provides results for the GS-IHP including hourly integrated peak kW demand. 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 4. The total energy consumption and consumption by individual modes for the baseline system are from the hourly TRNSYS simulations. For the GS-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 GS-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 2. Annual site HVAC/WH system energy use and peak for 1800-ft^2 well insulated house with Baseline HVAC/WH system

Location	Heat pump cooling capacity (tons)	HVAC site energy use, kWh	HVAC peak integrated hourly kW (W/S)
Atlanta	1.25	7657	5.9/4.4
Houston	1.25	8349	5.9/4.0
Phoenix	1.50	7165	6.2/4.4
San Francisco	1.00	4937	5.6/4.8
Chicago	1.25	10726	9.7/4.8

Table 3. Estimated annual site HVAC/WH system energy use and peak for 1800-ft^2 well-insulated house with GS-IHP system (winter humidification active)

Location	Heat pump cooling capacity (tons)	HVAC site energy use, kWh	HVAC peak integrated hourly kW (W/S)	% energy savings vs. NZEH/Baseline HVAC
Atlanta	1.25	3007	2.0/1.2	60.7
Houston	1.25	3290	1.8/1.1	60.6
Phoenix	1.50	2909	1.7/1.2	59.4
San Francisco	1.00	1699	1.8/1.6	65.6
Chicago	1.25	5126	6.9/1.7	52.2

The results summarized in Tables 3 and 4 show that the GS-IHP achieved >50% savings over the baseline system in the study in all locations. Further, savings approach or exceed 60% in all other cities. Winter peak kW ranged from about 30% to 70% lower for the GS-IHPs than for the baseline. Cooling peaks ranged from about 65% to 73% lower.

Table 4. Detailed GS-IHP performance vs. baseline system

Loads (1800 ft^2 well-insulated house from TRNSYS simulation with Baseline system)		Equipment		
		Baseline	GS-IHP	
Source	kWh	Energy use, kWh (I^2R)	Energy use, kWh (I^2R)	Energy reduction compared to baseline
Atlanta				
Space Heating	4717	1724 (21)	1384	19.7%
Space Cooling	5770	2069	996	51.9%
Water Heating	3032	3402	579 (144)	83.0%
Dedicated DH	208	273	31	88.8%
Ventilation fan	-	189	17	90.9%
Totals	13727	7657	3007	**60.7%**
Humidifier water use	512 kg	512 kg	647 kg	
Houston				
Space Heating	1734	626	493	21.3%
Space Cooling	10093	3652	1936	47.0%
Water Heating	2505	2817	685 (91)	75.7%
Dedicated DH	859	1065	165	84.5%
Ventilation fan	-	189	11	94.3%
Totals	15191	8349	3290	**60.6%**
Humidifier water use	81 kg	81 kg	89 kg	
Phoenix				
Space Heating	1546	515	366	29.0%
Space Cooling	9510	3985	2038	48.9%
Water Heating	2189	2476	473 (19)	80.9%
Dedicated DH	-	-	0	Na
Ventilation fan	-	189	32	83.1%
Totals	13285	7165	2909	**59.4%**
Humidifier water use	167 kg	167 kg	240 kg	
San Francisco				
Space Heating	2839	902	907	-0.6%
Space Cooling	86	32	19	39.5%
Water Heating	3387	3767	742 (100)	80.3%
Dedicated DH	37	47	3	94.4%
Ventilation fan	-	189	28	85.2%
Totals	6349	4937	1699	**65.6%**
Humidifier water use	32 kg	32 kg	29 kg	
Chicago				
Space Heating	11259	5206 (1242)	3901 (293)	25.1%
Space Cooling	2541	908	335	63.1%
Water Heating	3808	4287	846 (327)	80.3%
Dedicated DH	106	136	29	78.7%
Ventilation fan	-	189	15	92.2%
Totals	17714	10726	5126	**52.2%**
Humidifier water use	1387 kg	1387 kg	1721 kg	

First generation prototype development

Development process and projected GS-IHP prototype performance vs. baseline systems in a well-insulated 2600 ft^2 house located in a range of climates. In early 2008 CM and ORNL began a series of GS-IHP system design iterations using results of lab tests performed by CM to calibrate the variable-speed research version of the DOE/ORNL heat pump design model (HPDM) (Rice 1991). The process is documented by Rice, et al (2013) and summarized in this subsection.

A nominal 2-ton (7 kW) design cooling capacity was selected for development leading to the first prototype field testing. The design uses inverter-driven variable-speed brushless permanent magnet (BPM) rotary compressor, blower, and pumps, all with communicating capability. The inverter is suction-line cooled to allow operation in warmer ambient conditions. Dual electronic expansion valves (EEVs) are used to provide a wide range of refrigerant flow control. Single- and double-walled fluted tube-in-tube refrigerant-to-water heat exchangers (HXs) were used for the ground loop and the domestic hot water (DHW) loop, respectively, with a tube-in-fin air-to-refrigerant HX for the indoor coil.

One potential technical challenge in multi-function heat pumps like IHPs is refrigerant charge management. For GS-IHPs, this challenge is less relative to air-source designs because the loop HX has about the same internal refrigerant volume as for the indoor coil. For the prototype design, the maximum difference in HX internal volume between modes was 12% which was accommodated by different levels of condenser subcooling. To deal with the management of refrigerant charge in inactive parallel components, a small capillary tube arrangement and second reversing valve were used to return charge to the suction line.

CM assembled a prototype system and tested it in their laboratory over a wide range of ground-source conditions. ORNL used the detailed lab measurements of refrigerant and source/sink conditions to calibrate the research version of HPDM (Rice et al, 2005) in each of the four operating modes: space heating, space cooling, space cooling and WH, and dedicated WH. The HPDM was linked to a publicly available optimization program GenOpt (Wetter, 2009) 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. Differences between the calibrated model and the lab data in capacity and compressor –only COP for the dedicated WH mode averaged 2.6 and 2.0% with standard deviations of 2.8 and 3.3%. Blower power vs. airflow equations were developed from test data based on an external static pressure of 0.5 inches water column (0.125 kPa) at the design flow rate of 850 scfm (0.40 sm^3/s). Pump power relationships as a function of water or glycol flow rates were developed based on matching manufacturer's performance curves for brushless permanent-magnet (BPM) pumps against manufacturer's system head curves for a reference vertical-bore ground-loop design of 200 ft (61 m) bore depth. A DHW pump power relationship versus flow was also developed for an assumed DWH loop head characteristic.

The HPDM was first used to optimize flow rates for maximum performance over the range of compressor speeds and appropriate associated source/sink temperatures. This information was used by the manufacturer in developing suitable control algorithms and approaches for the four operating modes based on the available system operating temperatures and pressures and component intercommunication capabilities. Examples of these controls are the following: 1) in the space heating mode controlling to a specified supply air temperature within the limits of the minimum and maximum allowed airflow rates, 2) in water heating modes (full condensing), controlling the domestic hot water (DHW) pump flow rate to maintain a near optimal fixed delta-T across the water-to-refrigerant HX, 3) in the space cooling modes, controlling the compressor suction pressure to a specified value, depending on how far the indoor RH was away from the set point, when in active RH control, and 4) in each mode where the outdoor loop was active, controlling the loop flow rate as a prescribed function of the entering water temperature (EWT) to the heat pump from the GHX loop – aka GHX loop EWT. The control capability of the research HPDM was extended to allow each of these equipment internal control approaches to be implemented. In addition, the compressor speed was adjusted with loop EWT in the dedicated WH mode to maintain a WH capacity of 5.3kW or higher – slightly greater than the heat input of typical electric resistance water heaters (4 to 5 kW).

Once the control approaches and calibration equations were complete, we used HPDM to generate performance maps (i.e., tables) for each mode as a function of all relevant independent variables, e.g., compressor speed, indoor DB, indoor RH, GHX loop EWT, and DHW loop EWT. The GHX loop EWT is generally higher than outdoor air temperature in winter and lower in summer as illustrated in Figure 2. (We did not implement active RH control for this analysis, but rather used the default starting suction pressure level for passive RH control for all the space cooling performance maps; this simplified the GSIHP modeling requirements and avoided having to add dehumidifiers to the baseline and GSHPwDS cases.) These performance maps were used as input to TRNSYS (Solar Energy Lab et al, 2010) using a custom interface and thermostat control logic for linkage with house and DHW tank models for annual performance simulations. The selected house for the prototype development analysis was a tight-well insulated 2600 ft^2 (242 m^2) three-bedroom unit with a 2-ton (7 kW) design cooling load and the DHW tank was a nominal 50 gallon (189 l) capacity. Mechanical ventilation per ASHRAE STD 62.2 (2007) was assumed to be provided by continuous operation of a bathroom ventilation fan.

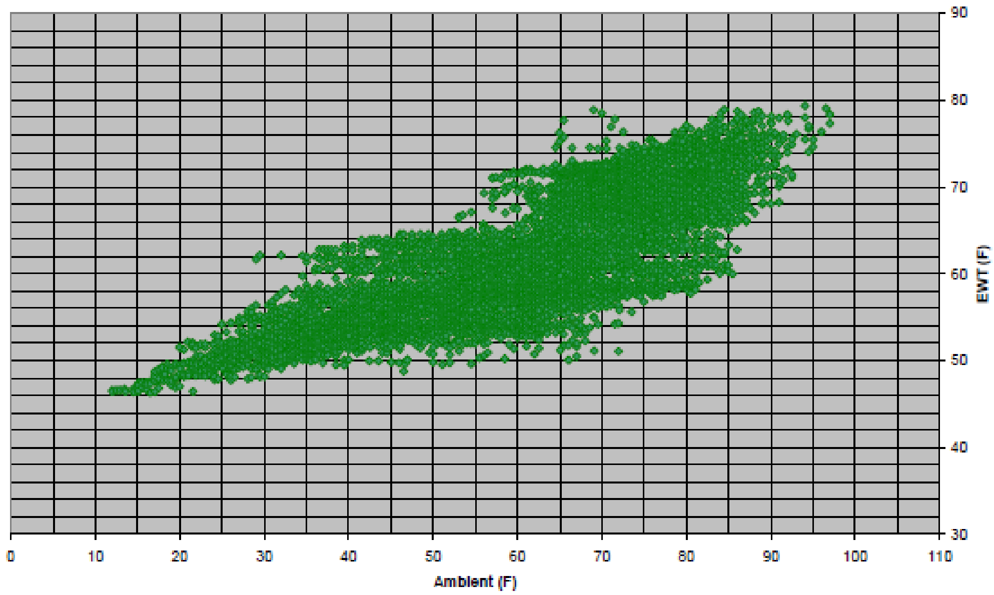

Fig. 2. Representative relationship of GHX loop EWT vs. outdoor ambient air temperature (from TRNSYS simulation for Atlanta location in Murphy et al 2007)

The time steps in TRNSYS for the seasonal performance analysis were 3 minutes between thermostat call priority decisions. Control logic rules were applied to give priority to water heating when both space and water heating calls were active if the indoor DB was within 2°F (1.1°C) of the heating mode set point. DHW controls for heat pump WH operation for the analysis were set to operate until the lower tank temperature was 120°F (49°C) and the upper electric element was set to minimize electric element use while maintaining the upper tank delivery temperature above 105°F (41°C). The assumed daily water use schedule shown in Figure 3 includes discrete tempered and untempered hot water draws totaling 64.3 gal/day (243.4 l/day), consistent with the Department of Energy (DOE 2010) daily hot water draw totals for electric resistance and heat pump water heater (HPWH) Energy Factor (EF) testing.

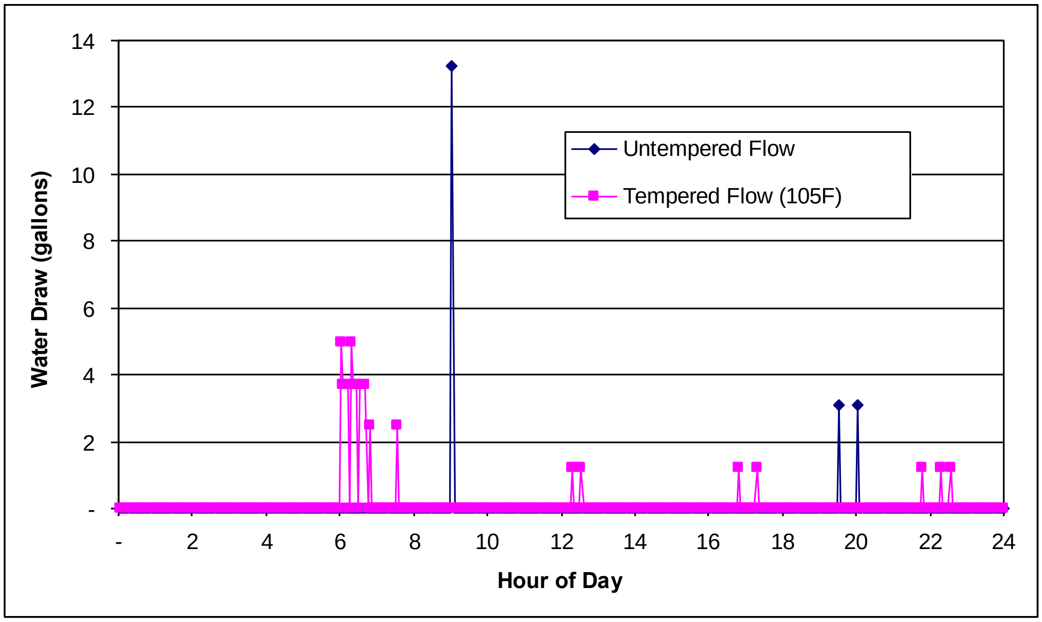

Fig. 3. Assumed daily hot water draw schedule from DHW tank

Baseline equipment modeling. To determine the energy savings potential of the GS-IHP design, two baseline cases were also defined and their annual performance simulated in TRNSYS. First, a minimum efficiency standard, electric-driven equipment set was defined. This included a 2-ton (7 kw) fixed capacity air-source heat pump with a rated SEER of 13 (cooling SPF=3.8) and HSPF of 7.7 (heating SPF=2.3), represented as a function of ambient and indoor conditions based on a manufacturer's published data, and a 0.90 EF electric water heater. This is essentially the same as the baseline system used in the IHP concept development (see Background section) but without the dehumidifier and humidifier units.

Next a high-efficiency commercially available two-capacity 2-ton (7 kW) ground source heat pump with desuperheater (GSHPwDS) was modeled in HPDM, which was calibrated based on manufacturer's lab data as was done for the GS-IHP case. The two-capacity GSHP has a full load rating of 18.5 EER (5.4 COP) cooling and 4.0 COP heating per ISO standard 13256-1 (1998). Part load ratings are 26 EER (7.6 COP) and 4.6 COP. Full- and part-load GSHP cooling capacities are 26.6 and 21.3 MBtu/h (7.80 and 6.25 kW) with full- and part-load heating capacities of 19.8 and 16.5 MBtu/h (5.80 and 4.84 kW). The desuperheater function was modeled in TRNSYS as a fixed HX effectiveness based on the manufacturer's test data, pump operation logic, and recommended control settings for the DHW tank element thermostats for a 120°F (49°C) set point. The ground and DHW loop pumps were typical single-speed induction-motor designs used by the manufacturer.

Ground loop modeling. The ground loop configuration for the GS-IHP was modeled in TRNSYS as two vertical bore wells connected in parallel. Soil properties were assumed or measured for 5 U.S. locations corresponding to Building America climate regions (DOE, 2012) of mixed-humid, hot-humid, hot-dry, marine, and cold. Ten-year sizing runs were made at multiple bore lengths for the GS-IHP and two-capacity GSHPwDS and used to determine the required length to stay within the minimum (winter) and maximum (summer) 10-year design EWTs. (As the minimum and maximum EWTs are approaching asymptotic values at 10 years of operation, 20-year values would be only slightly higher.)

Table 5 shows the assumed soil characteristics and grout types for the 5 U.S. locations, the loop fluid, the min and max design temperatures, and the required bore lengths and specifications.

Table 5. TRNSYS 10-year bore sizing results for GSHPwDS and GS-IHP units in reference house in 5 different U.S. locations

	Soil Characteristics, Assumed* or Measured^M		Loop Fluid	Min 10-yr EWT	Max 10-yr EWT	Grout Type	Bore Length / Unit Cap. GSHPwDS	Grout Type	Bore Length / Unit Cap. GSIHP
Location	k Btu/hr-ft-F [W/m-ºC]	diffusivity ft²/day [mm²/s]		ºF [ºC]	ºF [ºC]	GSHP	ft/ton [m/kW]	GSIHP	(ft/ton) [m/kW]
Atlanta	1.2 [2.1]	0.90 [0.97]	Water	42 [5.6]	95 [35]	Std	313 [27.1]	Enh	294 [25.5]
Houston	1.2 [2.1]	0.90 [0.97]	Water	42 [5.6]	95 [35]	Std	294 [25.5]	Enh	220 [19.1]
Phoenix	0.8^M [1.4^M]	1.65^M [1.77^M]	Water	42 [5.6]	95 [35]	Std	572 [49.6]	Enh	449 [38.9]
San Francisco	1.4 [2.4]	1.02 [1.10]	Water	42 [5.6]	95 [35]	Std	268 [23.2]	Enh	310 [26.9]
Chicago	1.4 [2.4]	1.02 [1.10]	20% PG	30 [-1.1]	95 [35]	Std	233 [20.2]	Enh	299 [25.9]
	*per soil property data on GEOKISS site (http://www.geokiss.com/res-design/GSHPDesignRec2.pdf)								

Bore Specifications:
Number of Bores = 2
Bore Diameter = 4.5"[11.4cm], Borehole Separation = 15'[4.57m], Nominal HDPE Pipe Size = 0.75"[1.9cm]
Grout Conductivity Assumptions:
Standard grout, 0.4 Btu/hr-ft-ºF [0.69 W/m-ºC]
Enhanced grout, 0.9 Btu/hr-ft-ºF [1.56 W/m-ºC]

For the primary analysis, standard grout was assumed for the conventional 2-capacity GSHPwDS and enhanced grout for the GS-IHP. Enhanced grout was found to more than pay for the added cost by reducing the required bore length, which was especially beneficial in balanced and cold climates due to the added heat extraction from the ground loop in the winter and shoulder months to meet the DHW load. In Atlanta, the required bore length for the GS-IHP with the enhanced grout was 33% less than had standard grout been used; however, the annual energy use for the GS-IHP was found to be nearly the same regardless of grout used since both cases stayed similarly within the minimum and maximum loop design temperatures. Had standard grout been used for the GS-IHP Atlanta case however, 25% more bore depth was predicted to be required than for the 2-capacity GSHPwDS case.

The relative bore depth requirements between the GS-IHP and two-capacity GSHPwDS given in Table 5 show a 6% shorter bore for the GS-IHP in Atlanta, 22 and 25% less depth needed in Phoenix and Houston, and 16 and 28% longer bores needed in San Francisco and Chicago.

Simulated annual performance results and discussion. Once the vertical bore sizing was completed for the ground-source cases, TRNSYS simulations were performed for comparable system control setups and DHW tanks for the minimum efficiency ASHP/electric resistance WH combination, the two-capacity GSHPwDS, and variable-speed GS-IHP cases. Note that for the ASHP case, only frost/defrost losses from the ASHP ratings test were included so defrost tempering energy use was not included; as such ASHP space heating energy use will be underestimated. Cyclic losses are not included in the two-capacity GSHPwDS or variable-speed GS-IHP cases, but are expected to be small, especially for the variable-speed case. Suitable pump power adjustments were applied in TRNSYS for the actual bore lengths for each location by accounting for the fraction of the pump head attributable to the GLHE in the reference loop design. No fouling effects were assumed for the ground loop or DHW water-to-refrigerant HXs.

In Table 6, energy use for space conditioning, water heating, and ventilation operation is given for the three cases as well as modal and total energy savings percentages relative to the baseline ASHP with electric water heater. Energy use in the combined space cooling and WH mode was apportioned to each function based on the ratio of delivered cooling to total energy delivered. The portion of the space or water heating energy use that was from resistance heating is shown as red in parentheses.

Predicted WH benefits from the two-capacity GSHPwDS cases are shown in Table 7 where the desuperheater provided 11 to 35% of the delivered hot water in San Francisco and Phoenix, respectively, with values ranging from 24 to 32% in the other three locations. Average savings in WH energy use was 21.2%.

Predicted total savings for the alpha prototype GS-IHP design is seen in Table 6 to range from 52.7% to 59.0%. Average savings are 54.9% over the 5 climates. Water heating savings relative to resistance units range from 68 to 78%.

Savings by mode for the GS-IHP are summarized in Table 8 relative to the baseline unit and as a percentage of the total savings in each location The latter depends on the product of the relative modal savings fraction, the normalized baseline modal power per unit load, and the fraction of the total delivered energy in each mode, each of which are given in the table. (The normalization factor in the second term is the total energy savings / total delivered load.). The delivered water heating energy is seen to range from 15% of the total in Phoenix to 44% of the total in San Francisco, ranging from 17 to 20% in the other three locations. As a fraction of the total energy savings over all modes of operation, GS-IHP water heating contributed 47 to 86% of the savings. As houses become tighter and better insulated (and/or smaller) and the sensible space conditioning loads decrease, the fraction of the total delivered energy that is from water heating will increase, which will provide higher total energy savings for GS-IHP equipment.

Table 6. Energy Use and Savings for Prototype 1 Relative to Minimum Efficiency Equipment Suite in Residential 2-ton (7 kW) Cooling Application

	Equipment Options				
	ASHP	2-Capacity GSHP w DS		Variable-Speed GSIHP	
Operation Mode	Energy Use, kWh (I^2R)	Energy Use, kWh (I^2R)	% Savings From Base	Energy Use, kWh (I^2R)	% Savings From Base
Atlanta					
space heating	2388	1660	30.5%	1321	44.7%
resistance heat	(93)	(5)		(5)	
space cooling	1608	1177	26.8%	833	48.2%
water heating	3293	2672	18.8%	872	73.5%
resistance heat	(3293)	(2524)		(1)	
ventilation fan	189	189		189	
totals	7479	5699	23.8%	3215	57.0%
Houston					
space heating	1102	754	31.6%	576	47.7%
resistance heat	(6)	(0)		(1)	
space cooling	2548	2154	15.5%	1680	34.1%
water heating	2813	2030	27.8%	648	77.0%
resistance heat	(2813)	(1876)		(0)	
ventilation fan	189	189		189	
totals	6653	5128	22.9%	3093	53.5%
Phoenix					
space heating	762	542	28.9%	370	51.4%
resistance heat	(0)	(0)		(0)	
space cooling	3450	2756	20.1%	2153	37.6%
water heating	2470	1731	29.9%	536	78.3%
resistance heat	(2470)	(1575)		(0)	
ventilation fan	189	189		189	
totals	6871	5218	24.1%	3248	52.7%
San Francisco					
space heating	1366	1142	16.4%	935	31.6%
resistance heat	(0)	(0)		(0)	
space cooling	23	4	83.9%	12	49.2%
water heating	3766	3405	9.6%	1057	71.9%
resistance heat	(3766)	(3330)		(0)	
ventilation fan	189	189		189	
totals	5344	4741	11.3%	2192	59.0%
Chicago					
space heating	6448	4052	37.2%	3652	43.4%
resistance heat	(1268)	(95)		(39)	
space cooling	651	333	48.8%	277	57.5%
water heating	4140	3309	20.1%	1332	67.8%
resistance heat	(4140)	(3108)		(120)	
ventilation fan	189	189		189	
totals	11429	7884	31.0%	5450	52.3%

Table 7. Predicted Desuperheater Contribution for Two-Capacity GSHP Unit

Location	% DHW Load Supplied By Desuperheater
Atlanta	24.2%
Houston	31.9%
Phoenix	34.8%
SanFrancisco	11.2%
Chicago	26.3%

Table 8. Breakdown of Energy Savings for Prototype 1 in Residential 2-Ton (7 kW) Cooling Application

Variable-Speed GSIHP in 2600 ft^2 (242 m^2) House				
Primary Delivery Function	Fractional Energy Savings from Base	Normalized Base Power Per Unit Load	Fraction of Total Delivered Energy	% of Total Energy Savings
Atlanta				
Space Heat	0.447	1.37	0.410	25.0%
Space Cool	0.482	0.96	0.391	18.2%
Water Heat	0.735	3.88	0.199	56.8%
Houston				
Space Heat	0.477	1.53	0.203	14.8%
Space Cool	0.341	1.14	0.626	24.4%
Water Heat	0.770	4.62	0.171	60.8%
Phoenix				
Space Heat	0.514	1.42	0.148	10.8%
Space Cool	0.376	1.35	0.707	35.8%
Water Heat	0.783	4.70	0.145	53.4%
San Francisco				
Space Heat	0.316	0.80	0.545	13.7%
Space Cool	0.492	0.70	0.011	0.4%
Water Heat	0.719	2.69	0.445	86.0%
Chicago				
Space Heat	0.434	1.57	0.688	46.8%
Space Cool	0.575	0.91	0.120	6.3%
Water Heat	0.678	3.62	0.192	47.0%

Ultimately this iterative collaborative process led to an initial packaged prototype system design by May 2009 (pictured in Figure 4 below). The first of these units were installed in the homes of two CM employees - actually those of the CEO and the lead GS-IHP development engineer. They were tested and evaluated over the next six months (in CM's lab as well as in the field). This testing revealed a few issues requiring some design and controls modifications. By mid-November 2010 development of the 1st generation prototype GS-IHP units was completed.

Summary of 1st generation GS-IHP prototype field performance vs. baseline minimum efficiency system. Two of the units were shipped to ORNL for installation and field evaluation in ZEBRAlliance houses 1 and 2 located in Oak Ridge. Installation was completed in late November and the two units were tested at the site from January 2011 through November 2011 as reported by Munk et al (2011). A significant difference between the Oak Ridge test houses and the house used for annual performance analyses during the prototype system development process is total conditioned floor area. The ZEBRAlliance houses had 3700 ft^2 (345 m^2) vs. 2600 ft^2 (242 m^2) for the analysis house – both had similarly tight, well-insulated thermal envelopes. The field test houses also had horizontal GHX loops (vs. vertical GHXs for the analysis house) and featured automatically zoned air distribution systems with four interior zones (main level master bedroom, main level living/kitchen area, upstairs bedrooms, and conditioned basement).

Fig. 4. CM's 1st generation packaged (field evaluation) prototype GS-IHP unit – l) panels removed with prototype control board pulled out for viewing; r) with all exterior panels in place.

The GS-IHPs were installed in the two houses on November 29-30, 2010. Shakedown testing was conducted during December 2010 to ensure full operational capability before starting data collection on January 1, 2011 and continuing through December 2011. We encountered several technical issues during operation of the 1st generation prototypes that resulted in frequent interruption of GS-IHP operation. While this limited the extent of performance data we were able to collect, the diagnostics provided invaluable information to CM, enabling them to develop a much improved 2nd generation prototype (discussed in more detail in the next section).

Despite the problems noted above, we were able to collect a large body of data during periods when the prototype units were running as designed and extract enough useful data to enable projection of their "as designed" performance potential. The data were used principally to calibrate the ORNL heat pump design model (HPDM) to the field-measured steady state performance of the IHPs. Detailed results of this calibration effort are given in ORNL/TM-2011/527 "*Steady-State Comparison of GS-IHP Field Data to Measured Performance,*" Rice, et al (2012). The calibrated HPDM was then used to develop performance maps and these, in turn, were input to the TRNSYS/HPDM (T/H) annual performance simulator along with the site weather data for the 2010 heating and cooling seasons, the site hot water usage averaging ~51 gal/day, average GHX loop EWTs and water mains temperatures during heat pump operation, and ZEBRAlliance house 2 (aka optimum value framing or OVF house) specifications to estimate annual performance compared to a baseline minimum efficiency equipment suite (the same baseline suite as described in the Background section minus the dehumidifier and humidifier). 2010 site weather data was used for the simulation because it was more complete than that for 2011 and the 2010 winter was more typical (actually colder than normal) than the exceptionally mild 2011/2012 winter for Oak Ridge. The GHX loop EWTs for 2010 are shown in Figure 5.

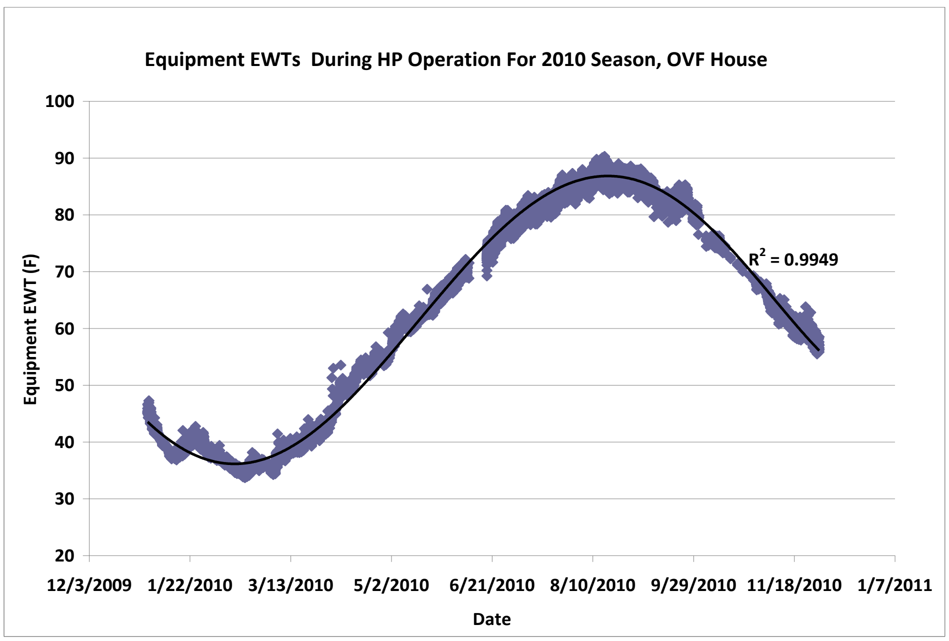

Fig. 5. Average return ground loop temperatures during heat pump operation for 2010 season, ZEBRAlliance house 2 in Oak Ridge, TN

Results of the T/H simulations follow in Tables 9 and 10. Table 9 shows the projected energy savings for the 1st generation prototype and Table 10 shows seasonal COPs (performance factors). Converting the seasonal performance numbers to US SEER and HSPF indices, the GS-IHP had a SEER of 20.8 Btu/Wh and an HSPF of 14.2 Btu/Wh.

Table 9. Projected 1st generation GS-IHP prototype energy savings vs. baseline systems for House 2 in 2010 Season

Loads and Energy Use by Mode; OVF House, 2010 Season				
Loads from Base Simulation		1-Speed Base	1st Generation GSIHP	
Operation Mode	kWh	Energy Use kWh (I^2R)	Energy Use kWh (I^2R)	Reduction from Base(%)
Oak Ridge, TN				
space heating	12759	5602	3060	45.4%
resistance heat		(921)	(79)	
space cooling	6111	1821	1002	45.0%
water heating	2506	2797	729	73.9%
resistance heat		(2797)	(0)	
ventilation fan		144	144	
totals	21376	10364	4934	52.4%

Table 10. Projected Seasonal COPs for OVF House in 2010 Season

Predicted Seasonal COPs. OVF House, 2010 Season			
	SC COP	SH COP	WH COP
Baseline ASHP	3.36	2.28	0.90
1st Gen. GSIHP	6.10	4.17	3.44

GS-IHP preliminary field data (2011). Preliminary field performance measurements for the 1st generation IHPs (including impacts of the hardware/controls issues described earlier) are given in Tables 11 and 12, below. Note that the water heating operation of the prototypes was significantly impacted due to a reversing valve issue (described more fully in following section).

Table 11. GS-IHP data from SIP House (#1) (Yr. 2011)

	Energy Used (Ded. H)	COPh	Energy Used (Ded. C)	COPc	Energy Used (Ded. WH)	Overall COP (Ded. WH)	Energy Used: Cooling in (SC+WH)	Energy Used: WH in (SC+WH))	Overall COP (SC+WH)	EWT(Ded. heating)	EWT(Ded. cooling)
Yr. 2011	(kWh)		(kWh)		(kWh)					(F)	(F)
Jan	675.31	4.59	0		134.31	3		0		39.5	
Feb	473.5	3.8	0		36.74	2.85		0		41.5	
Mar	411.5	2.8	0					0		47.04	
Apr	35.61	2.27	5.79	6.6	7.31	1.77	1.5	10.42	2.78	55.4	57.21
May	20.5	2.83	90.39	5.12	61.24	1.72	13.6	57.15	3.23	59.4	64.7
Jun			163.45	4.32	1.8	2.8	11.98	35.98	3.11		72.7
Jul			386.1	4.04	0.19	1.86	0.61	0.24	2.87		77.4
Aug			592.2	3.62				0			87.3
Sep			220.63	4.11				0			82.5
Oct	34.71	5.4	32.5	5.18				0		68.1	74.8
Nov	187.11	4.57	0					0		61.6	
Dec	415.78	4.25	0					0		53.4	
Subtotal (Jan-Nov)	1838.24		1491.06		241.6		27.7	103.8			

Table 12. GS-IHP data from OVF House (#2) (Yr. 2011)

	Energy Used (Ded. H)	COPh	Energy Used (Ded. C)	COPc	Energy Used (Ded. WH)	Overall COP (Ded. WH)	Energy Used: Cooling in (SC+WH)	Energy Used: WH in (SC+WH))	Overall COP (SC+WH)	EWT(Ded. heating)	EWT(Ded. cooling)
Yr. 2011	(kWh)		(kWh)		(kWh)					(F)	(F)
Jan	990.27	3.77	0		160.21	2.7				39.5	
Feb	544.32	3.95	0		109.33	2.6				39.4	
Mar	496.51	3.76	0							47.3	
Apr	82.22	2.97	17.62	7.92	16.95	3.55	3.56	12.34	4.54	55.9	57.1
May	40.96	3.06	87.15	6.11	98.00	1.75	14.07	38.68	4.26	61.4	63.6
Jun			276.26	4.7	9.98	1.28	25.72	76.97	3.36		72.9
Jul			503.61	3.71	5.62	1.4	23.00	109.48	2.29		81.1
Aug			605.91	3.24	3.66	0.96	21.13	121.47	1.96		86.7
Sep			192.05	3.8	47.45	1.07	10.78	90.47	1.81		80.6
Oct	68.40	4.28	29.11	4.16	134.15	1.55	6.22	18.97	3.05	67	73.6
Nov	269.62	4.13	0.00		199.31	1.48	0.00	0.00		60.1	
Dec	226.97	4.08	0.00		137.71	1.4	0.00	0.00		53.6	
Subtotal (Jan-Nov)	2492.30		1711.71		784.66		104.48	468.38			

SC = Space cooling; SC+WH = space cooling + water heating; EWT= entering water temperature; Ded. C = dedicated cooling; Ded. H= dedicated heating; Ded. WH= dedicated water heating.

Based on the 1st generation prototype field data and projected annual performance analyses the following observations are derived:

- The 1st generation GS-IHP prototypes fully provided space conditioning needs for the two test houses but only provided partial water heating needs due to technical issues illuminated during the field test period
- Analytical projections based on ORNL's TRNSYS/HPDM model as calibrated to the measured field performance indicate that the 1st generation units could have achieved >52% savings vs. baseline minimum efficiency HVAC/WH equipment at the ZEBRAlliance site (had they operated without the performance issues described above and in the next section)
- Our field testing efforts uncovered a number of system hardware and control issues that enabled CM to generate an improved 2nd generation design which they have introduced to the market as of March 2012 – the Trilogy 40™
- The technical issues were communicated to CM (the CRADA partner) and have been implemented into their 2nd generation design (see next section).

Second generation prototype development

As noted briefly in the previous section, we encountered several technical issues (enumerated below) that resulted in frequent interruption of the 1st generation GS-IHP operation at the field test site. This required spending an extensive amount of time during the 2011 test year in collecting diagnostic data on these issues from the control board of the IHPs and sharing them with CM. While this limited the extent of performance data we were able to collect, the diagnostics provided invaluable information to CM, enabling them to develop a much improved 2nd generation prototype GS-IHP design with modifications to correct the issues uncovered with the 1st generation design.

Hardware and control technical issues discovered during 1st generation IHP field testing. Below, we list the design changes that CM implemented in their 2nd generation GS-IHP prototypes (or is considering for future product modifications) based on issues discovered in field tests of the 1st generation units.

1. Change air coil from tube-and-fin to microchannel design.
2. Move hot water condenser into a true parallel circuit and eliminate desuperheating function since it adds little value and creates charge compensation problems.
3. Reduce the water heating reversing valve (RV) size to eliminate switching problems. Problems with this RV resulted in poor performance by the 1st generation units in water heating modes throughout the entire 2011 test year, becoming increasingly worse with time (see RV performance history below).
4. Change refrigerant-to-water heat exchanger (HX) for hot water production to a double walled brazed plate type as it is more compact and efficient.
5. Change refrigerant-to-water HX for ground loop to a brazed plate type – primary reason to reduce unit size to fit into a standard 2-ton cabinet.
6. Change the brine loop pump to a newer, more efficient version.
7. Upgrade to a more robust inverter design.
8. Upgrade the controls to address small zone temperature offset (~1-2 °F) seen at test site – may be pushed to a later date.
9. Other small tweaks to make service easier and provide easier access to some components as we experienced with the inverter in the 1st generation design.
10. Changed check valves to one with a reduced seat leakage. The previous valves were found to be problematic during development of the second version. Using check valves with a consistent low seat leakage was critical to improved performance.

<u>History of water heating RV Performance</u>. The water heating RVs in both 1st generation test units operated correctly for better than 95% of the time during January and February of 2011. In March the RV problem became much worse and the water heating operation modes were disabled. Intensive dialog on the issue began with CM. A temporary fix was implemented and WH operation was re-enabled in April but the RV issue resurfaced and as Figure 6 shows, and worsened with time – essentially no “good” WH operation with the IHP in House 1 after June and none in the House 2 unit after August.

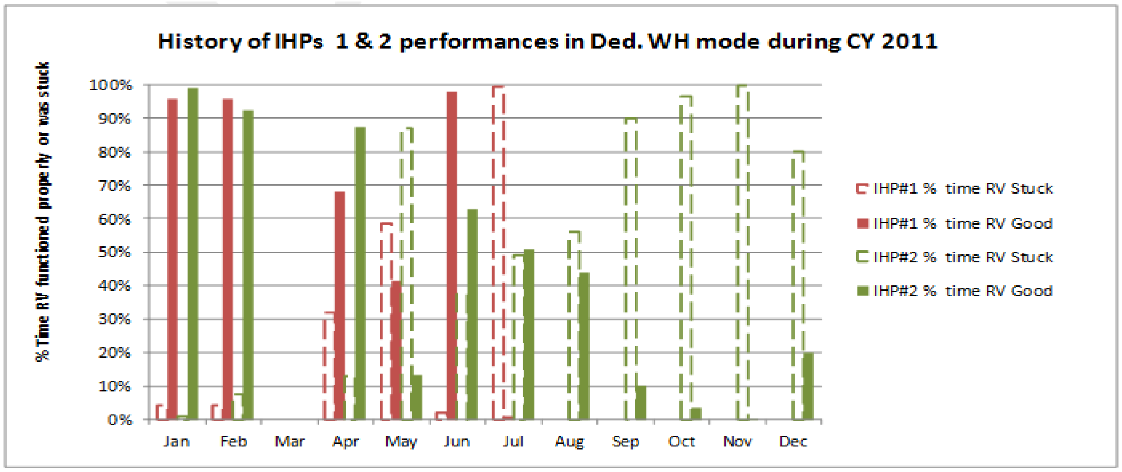

Fig. 6. Performance of water heating RVs in both IHPs

Projected 2nd generation GS-IHP prototype performance vs. baseline systems in a well-insulated 2600 ft^2 house located in a range of climates. As noted, a 2nd generation prototype design has been developed incorporating design changes listed above to resolve the field operational issues experienced by the 1st generation prototypes and to improve unit performance and serviceability. This 2nd generation GS-IHP design is the basis on which CM recently announced limited production of their new IHP product, Trilogy® 40 Q-mode™ (see press release in the Appendix). The system became available for order in December 2012. Samples of the new design were installed in ZEBRAlliance houses 1 and 2, replacing the 1st generation units and field tested from June 2012 through February 2013.

The fluted tube-in-tube and tube-and-fin HXs in Prototype 1 for the water- and air-to-refrigerant components, respectively, were replaced with single- and double-walled brazed plate and microchannel HXs to improve performance and reduce refrigerant charge inventory, weight, and space requirements. CM assembled a prototype 2 system and tested it in their laboratory over a range of ground-source conditions. ORNL used the detailed lab measurements of refrigerant and source/sink conditions to calibrate a newly developed flexible platform version of the HPDM (HPDM-flex) capable for modeling brazed plate and microchannel HXs. This was done in each of the four operating modes: space heating, space cooling, space cooling and WH, and dedicated WH. As with prototype 1 calibrations, the HPDM-flex model was linked to a publicly available optimization program GenOpt (Wetter, 2009) 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, where possible, 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.

Table 13 summarizes the difference between the calibrated models and the manufacturer's lab test data in capacity, compressor power, and compressor-only COP.

Table 13. Agreement of Calibrated Models to Prototype 2 GSIHP Lab Tests

Calibrated Model Results for Prototype 2 GSIHP					
Operation Mode	Calibration Statistics	Capacity	Compressor Power	Compr. Only COP
	(%)	(%)	(%)	(%)
Space Cooling	ave diff.	4.8	1.3	3.5
	std. dev.	2.1	1.9	2.6
Space Heating	ave diff.	4.8	1.4	3.4
	std. dev.	1.2	1	1.4
Dedicated WH	ave diff.	-3.9	-0.9	-3
	std. dev.	3	1.5	3.4

Comparisons of the space cooling and heating performance between prototype 1 and 2 are shown in Figures 7 and 8 as a function of EWT returning from the ground loop. The indoor conditions were set at standard ground-source water/brine-to-air heat pump dry and wet bulb temperatures (ISO 13256-1, 1998). The solid lines are for prototype 1 and the dashed lines for Prototype 2. The EERs and COPs include all blower and pump power required to meet assumed ground and air loop head characteristics. (The ground loop pump power is based on a 200 ft bore depth and 20% propylene glycol fluid and the indoor blower power is based on 0.5" water column (IWC) external static pressure drop at nominal airflow.)

The comparisons are made at the same low, medium, and high compressor speeds. These are 45, 65, and 85 Hz in space cooling and 35, 82.5, and 130 Hz in space heating. The predicted space cooling EER gains from prototype 2 are 8.5, 10.7, and 14.4% at low, medium, and high speed, averaging 11.2% over all simulated points shown. The space heating COP gains predicted for prototype 2 are -0.8, 10.2, and 5.5% at low, medium, and high speed, averaging 5% over all simulated points shown.(The slight drop off in heating performance at low speed in prototype 2 is due to a lower minimum airflow rate which gives a higher supply air temperature.) The lower COP gains in space heating are in part due to the average heating capacity gain of 11.2% for prototype 2. The average cooling capacity gain was 2.9%.

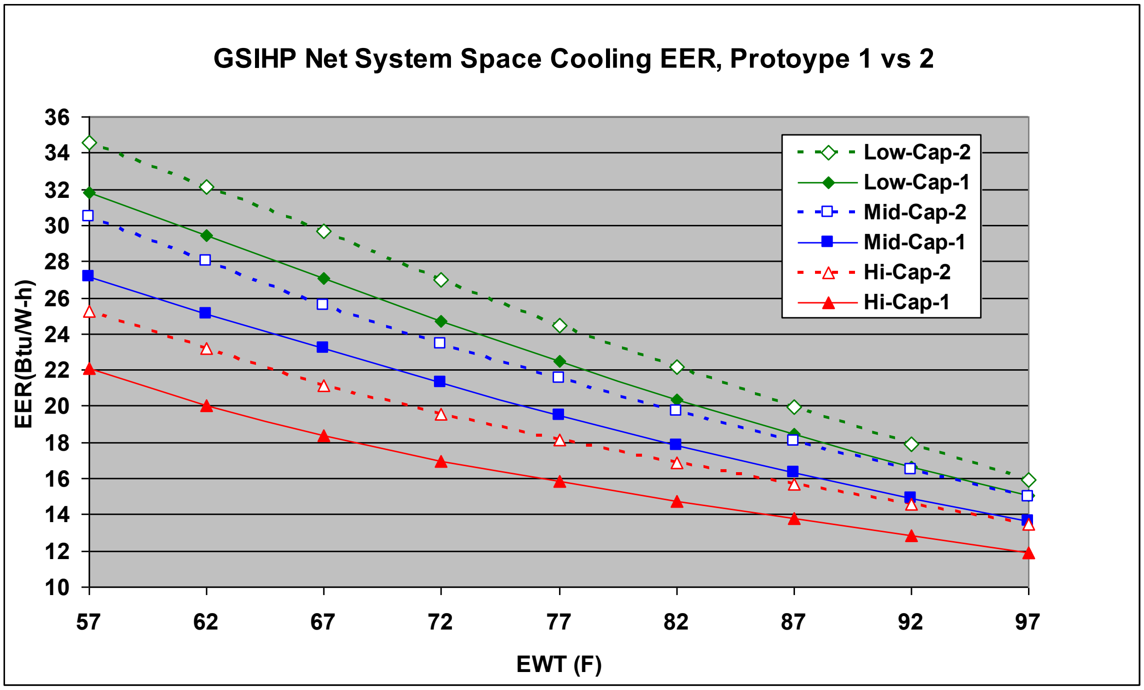

Figure 7. Comparison of space cooling EER versus EWT between GSIHP prototypes 1 and 2 over a range of compressor speeds.

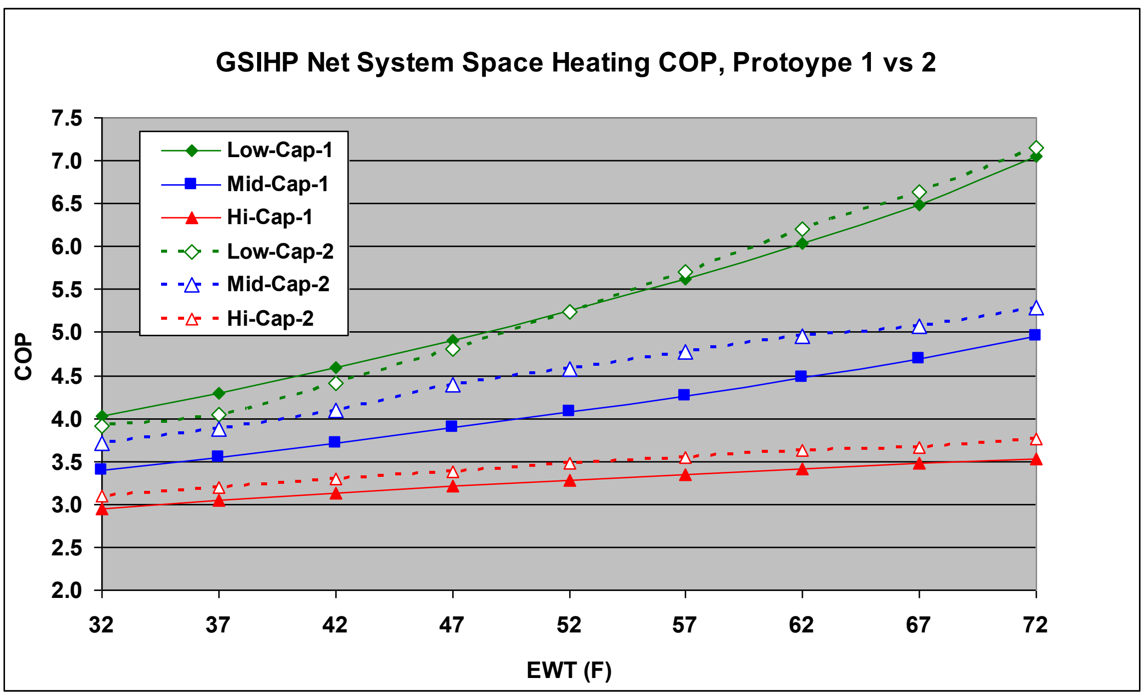

Figure 8. Comparison of space heating COP versus EWT between GSIHP prototypes 1 and 2 over a range of compressor speeds.

The rated performance of the prototype 2 design (per ISO 13256-1 at part load conditions) is over 40 EER. This value is quite a bit larger than the 29.2 EER from Figure

7 at the part load rated EWT of 68F and 45 Hz compressor speed. There are a number of reasons for this. First, the blower and pump power values used in the ISO rating procedure (ISO 13256-1, 1998) assume 30% overall efficiencies and only include the equipment pressure head, not the full loop pressure heads. The model predictions are based on the pump and blower performance maps and the full system loop heads. Second, prototype 2 had a minimum compressor speed of 35 Hz rather than 45 Hz in prototype 1. Last, the rated performance was with the RH control system turned off, which allowed the suction saturation temperature to rise considerably above the fixed 50.3 F value (for passive RH control) used for the simulations, giving a saturation temperature in the upper fifties and a sensible-heat-ratio (SHR) approaching 0.9. As such, this operating point would be seen primarily in dry climate conditions.

In Figure 9, the predicted dedicated water heating COPs are compared between prototype 1 and 2. Here the compressor speed varies from 90 to 50 Hz as a linear function of ground loop EWT between 30 and 80F. As such, we show families of EHWT from 70 to 110F as the second independent parameter. Predicted WH COP changes range from 10.3, 4.3, to -3.3% at 70, 90, and 110F EHWT, respectively, and averaging a 3.8% gain overall.

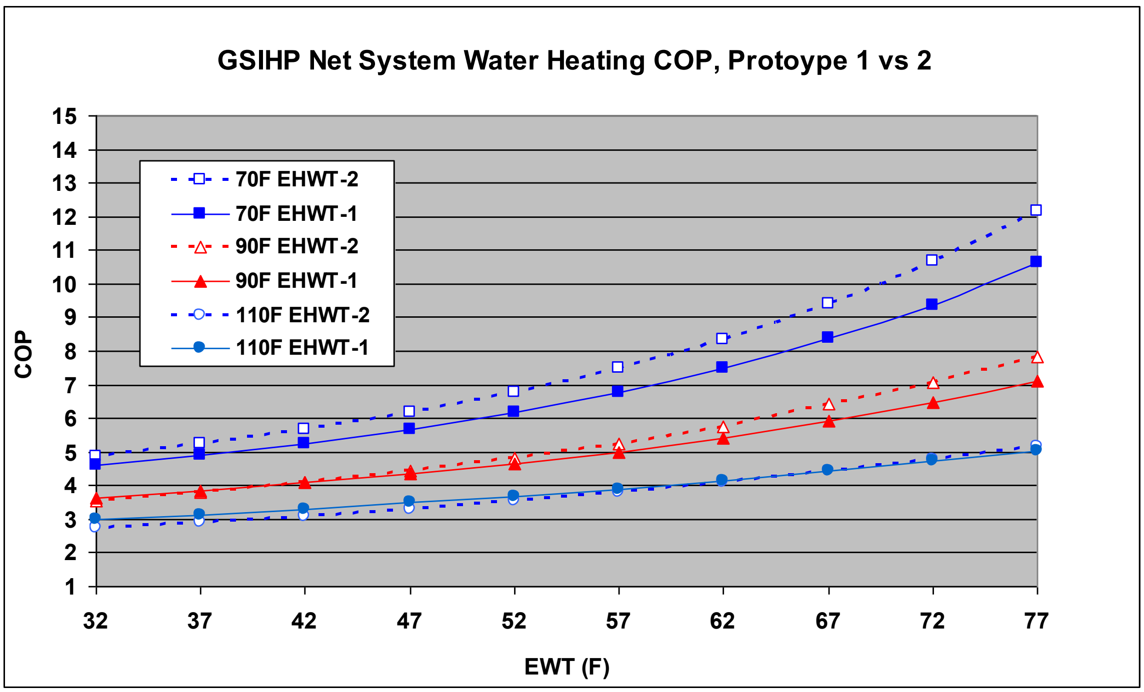

Figure 9. Comparison of dedicated water heating COP versus EWT between GSIHP prototypes 1 and 2 over a range of EHWTs.

Note that the performance increases shown in Figures 7-9 will be dampened somewhat by the longer ground loop lengths used in the TRNSYS analysis (from Table 5), as the ground loop pump power will be a larger fraction of the total power.

In Figure 10, the predicted combined space cooling and water heating COPs are compared between prototype 1 and 2 over a range of EHWTs for low, medium, and high compressor speeds of 50, 70, and 90 Hz. (This is the allowable compressor speed range for high-condensing temperature water heating.). Here the predicted EER gains average 8.7, 14.0, and 5.6% at 50, 70, and 90 Hz, with an overall average gain of 9.4%

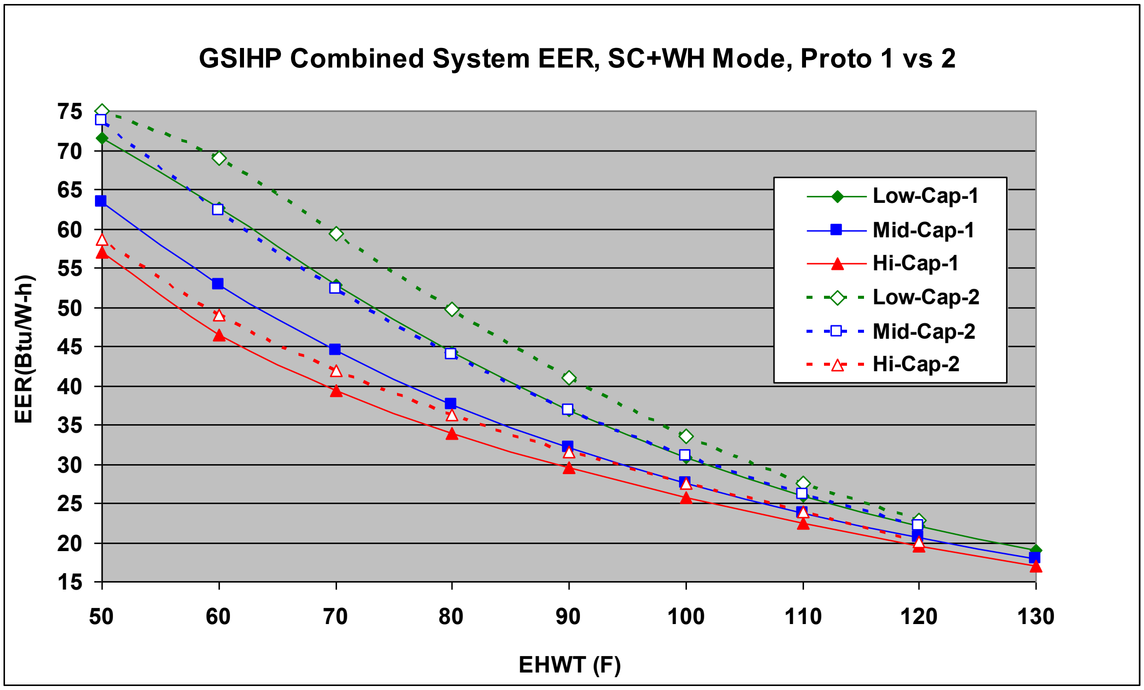

Figure 10. Comparison of combined space cooling and WH EER versus EHWT between GSIHP prototypes 1 and 2 over a range of compressor speeds

Predicted total annual energy savings for the GS-IHP prototype 2 design are shown in Table 14 based on TRNSYS analyses in five Building America locations. The predicted energy savings range from 57.2% to 61%. Average savings are 58.7% over the 5 climates, an average gain of 3.8 percentage points over prototype 1. Of as much or more significance, predicted comfort conditions in space conditioning improved in prototype 2 along with the performance increases. This was due to a combination of closer approach temperatures, lower minimum airflows, and higher heating capacities. The delivered supply temperatures in space heating were 3 to 4ºF higher and the SHR levels in space cooling were reduced from up to 0.85 to a narrower range between 0.76 and 0.79. The number of hours above 60% RH was lower with the GSIHP than with either the GSHPwDS or the baseline ASHP with the assumed passive RH control. The active RH control available in the GSIHP design should reduce these hours significantly further.

Electric resistance energy use for space and water heating is predicted to be essentially eliminated in all but the northern climate case, where it was reduced by 97.4%. Water heating savings relative to resistance units range from 68 to 79%.

Table 14. Energy Use and Savings for Prototype 2 Relative to Minimum Efficiency Equipment Suite in Residential 2-ton (7 kW) Cooling Application

	Equipment Options				
	ASHP	2-Capacity GSHP w DS		Variable-Speed GSIHP	
Operation Mode	Energy Use, kWh (I^2R)	Energy Use, kWh (I^2R)	% Savings From Base	Energy Use, kWh (I^2R)	% Savings From Base
Atlanta					
space heating	2388	1660	30.5%	1155	51.6%
resistance heat	(93)	(5)		(6)	
space cooling	1608	1177	26.8%	754	53.1%
water heating	3293	2672	18.8%	848	74.3%
resistance heat	(3293)	(2524)		(3)	
ventilation fan	189	189		189	
totals	7479	5699	23.8%	2946	60.6%
Houston					
space heating	1102	754	31.6%	495	55.1%
resistance heat	(6)	(0)		(1)	
space cooling	2548	2154	15.5%	1542	39.5%
water heating	2813	2030	27.8%	619	78.0%
resistance heat	(2813)	(1876)		(0)	
ventilation fan	189	189		189	
totals	6653	5128	22.9%	2845	57.2%
Phoenix					
space heating	762	542	28.9%	306	59.9%
resistance heat	(0)	(0)		(0)	
space cooling	3450	2756	20.1%	1921	44.3%
water heating	2470	1731	29.9%	510	79.4%
resistance heat	(2470)	(1575)		(0)	
ventilation fan	189	189		189	
totals	6871	5218	24.1%	2926	57.4%
San Francisco					
space heating	1366	1142	16.4%	813	40.5%
resistance heat	(0)	(0)		(0)	
space cooling	23	4	83.9%	10	57.0%
water heating	3766	3405	9.6%	1070	71.6%
resistance heat	(3766)	(3330)		(0)	
ventilation fan	189	189		189	
totals	5344	4741	11.3%	2082	61.0%
Chicago					
space heating	6448	4052	37.2%	3139	51.3%
resistance heat	(1268)	(95)		(41)	
space cooling	651	333	48.8%	251	61.5%
water heating	4140	3309	20.1%	1309	68.4%
resistance heat	(4140)	(3108)		(101)	
ventilation fan	189	189		189	
totals	11429	7884	31.0%	4888	57.2%

Prototype 2 savings by mode for the GSIHP are summarized in Table 15 relative to the baseline unit and as a percentage of the total savings in each location as a fraction of the total energy savings over all modes of operation, GSIHP water heating contributes 43 to 83% of the savings, averaging 57%.

Table 15. Breakdown of Energy Savings for Prototype 2
in Residential 2-Ton (7 kW) Cooling Application

Variable-Speed GSIHP in 2600 ft^2 (242 m^2) House				
Primary Delivery Function	**Fractional Energy Savings from Base**	**Normalized Base Power Per Unit Load**	**Fraction of Total Delivered Energy**	**% of Total Energy Savings**
Atlanta				
Space Heat	0.516	1.29	0.410	27.2%
Space Cool	0.531	0.91	0.391	18.8%
Water Heat	0.743	3.65	0.199	53.9%
Houston				
Space Heat	0.551	1.43	0.203	15.9%
Space Cool	0.395	1.07	0.626	26.4%
Water Heat	0.780	4.32	0.171	57.6%
Phoenix				
Space Heat	0.599	1.31	0.148	11.6%
Space Cool	0.443	1.24	0.707	38.8%
Water Heat	0.794	4.31	0.145	49.7%
San Francisco				
Space Heat	0.405	0.77	0.545	17.0%
Space Cool	0.570	0.67	0.011	0.4%
Water Heat	0.716	2.60	0.445	82.6%
Chicago				
Space Heat	0.513	1.43	0.688	50.6%
Space Cool	0.615	0.83	0.120	6.1%
Water Heat	0.684	3.31	0.192	43.3%

The predicted annual COPs for the 2^{nd} generation prototype GSIHP and the baseline ASHP with electric resistance water heater are given in Tables 16 and 17. The water heating results show delivered COPs ranging from 2.8 to 4.1, as compared to ~0.9 COPs for the baseline electric water heating, an efficiency increase of 310 to 450%, averaging 380%. The space conditioning results predict average performance increases of 210% in space heating and 240% in space cooling.

Table 16, Predicted COPs for Prototype 2 GSIHP in Five U.S. Climates

Predicted Proto 2 GSIHP Performance			
	SC COP	SH COP	WH COP
Atlanta	8.56	5.18	3.42
Houston	6.55	5.96	3.84
Phoenix	5.70	7.18	4.06
SanFrancisco	9.93	5.10	3.03
Chicago	10.65	4.20	2.79

Table 17, Predicted COPs for Baseline ASHP with Resistance WH in Five U.S. Climates

Baseline Performance			
	SC COP	SH COP	WH COP
Atlanta	3.62	2.55	0.90
Houston	3.60	2.69	0.89
Phoenix	3.09	2.93	0.89
SanFrancisco	3.48	3.04	0.90
Chicago	3.61	2.09	0.90

2nd generation GS-IHP field performance observations. Two 2nd generation IHPs (basis of CM's new Trilogy® 40 Q-mode product line) were installed in ZEBRAlliance houses 1 and 2 on May 7, 2012 with the help of CM personnel (replacing the 1st generation units). The only refrigeration hardware issue involved two small refrigerant leaks associated with a pressure transducer installed in the house 1 unit by CM for our field data needs (would not be a part of production units). The first leak was repaired in May and this appeared to resolve the problem. However a second small leak in another solder joint where the transducer fitting joined the main refrigerant line caused lower than expected performance for the unit throughout May. It was repaired in early June and no further leak problems occurred.

The house 1 unit experienced two compressor inverter board failures which required replacements. CM worked closely with the inverter board vendor and determined that the root cause of the failure is related to momentary (few ms) loss of power to the board. When power returns an inrush resistor would become damaged as the compressor was still operating. A temporary software fix was implemented to turn off the compressor anytime a power loss was sensed. A control interface module was also redesigned to hold in relays used to bypass the inrush resistor upon line sync and normal operation.

A few controller operational issues also were identified which limited system operation in both units in May and June. One was related to the zone control system in the houses. The zone controller was interfaced with the CM unit controls but is not itself a CM product. In addition the units experience periodic loss of communication between the house thermostat and the unit control boards. Periodic reset at the main breaker was required to restart the unit after each episode. CM control engineers revised the operating code for the controller and it was uploaded in July. Through August, the two IHPs experienced much more reliable controller operation (apart from the board failures for the house 1 unit noted above).

For the July and August 2012 period, field measurements show that average space cooling only COPs for the beta prototypes ran about 10-15% higher than those achieved by the alpha prototypes in 2011 at similar entering water temperatures (EWT). Similar measurements for the January and February 2013 period show space heating only COPs for the beta units were about 2-5% higher than those of the alpha test units for similar EWT levels.

The GSIHP HPDM for the 2^{nd} generation unit, calibrated against lab data, was again used to develop performance maps and these, in turn, were input to the TRNSYS/HPDM (T/H) annual performance simulator along with the site weather data for the 2012 heating and cooling seasons, the site hot water usage averaging ~49 gal/day, average GHX loop EWTs and water mains temperatures during heat pump operation, and ZEBRAlliance house 2 (aka optimum value framing or OVF house) specifications to estimate annual performance compared to a baseline minimum efficiency equipment suite (the same baseline suite as described in the Background section minus the dehumidifier and humidifier). The GHX loop EWTs and water mains temperatures for 2012 are shown in Figures 11 and 12.

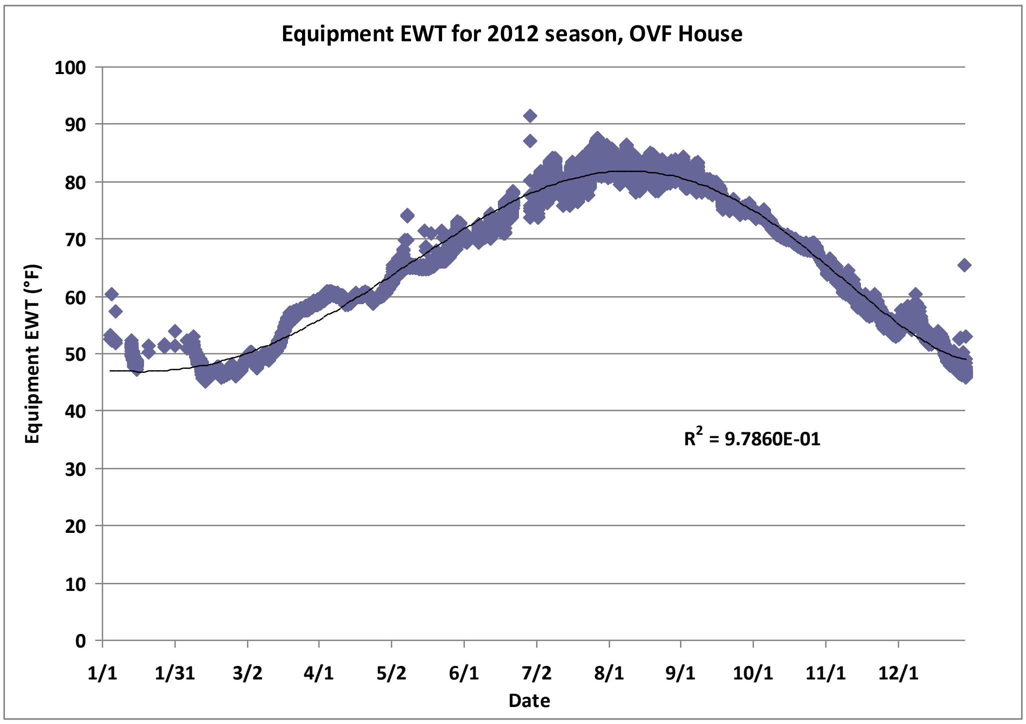

Fig. 11. Average return ground loop temperatures during heat pump operation for 2012 season, ZEBRAlliance house 2 in Oak Ridge, TN

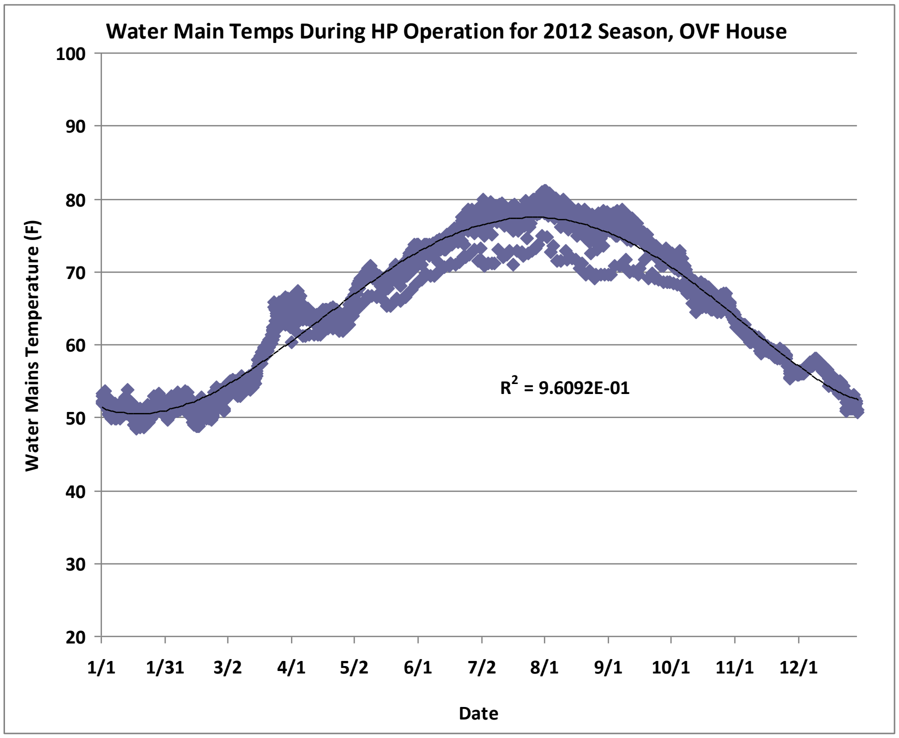

Fig. 12. Average water mains temperatures during heat pump operation for 2012 season, ZEBRAlliance house 2 in Oak Ridge, TN

Results of the T/H simulations follow in Tables 18 and 19. Table 18 shows the projected energy savings for the 2nd generation prototype where predicted total HVAC/WH savings are 57.8%. Space conditioning savings approach 50% while water heating savings exceed 76% compared with the baseline 0.90 EF electric resistance water heater.

Table 18. Projected 2nd generation GS-IHP prototype energy savings vs. baseline systems for House 2 in 2012 Season

Predicted Loads and Energy Use by Mode; OVF House, 2012 Season				
Loads from GSIHP Simulation		1-Speed Base	2nd Generation GSIHP	
Operation Mode	kWh	Energy Use kWh	Energy Use	Reduction from
Oak Ridge, TN				
space heating	8765	3265	1690	48.2%
resistance heat		(127)	(29)	
space cooling	5202	1539	768	50.1%
water heating	2313	2605	610	76.6%
resistance heat		(2605)	(0)	
ventilation fan		109	109	
totals	16280	7519	3177	57.8%

Table 19 shows the predicted seasonal COPs (performance factors). Converting the seasonal performance numbers to US SEER and HSPF indices, the GS-IHP had a predicted SEER of 23.1 Btu/Wh and HSPF of 17.7 Btu/Wh. These performance levels are higher than those in Table 10 in part due to the higher efficiency of the 2nd generation design and in part due to the milder winter in 2012 than in 2010.

Table 19. Projected 2nd generation seasonal COPs for OVF house in 2012 season

Predicted Seasonal COPs, OVF House, 2012 Season			
	SC COP	SH COP	WH COP
Baseline ASHP	3.38	2.68	0.89
2nd Gen. GSIHP	6.77	5.19	3.79

Figures 13-14, below illustrate how the space heating and cooling capacity of the 2nd generation prototype varied with outdoor air temperature (OAT) and compressor speed during the 2012-2013 field testing. The percentages inside the squares are the % of total run time in the respective OAT bins (columns) so it can be seen that the units operated at low compressor speeds and capacity most of the time.

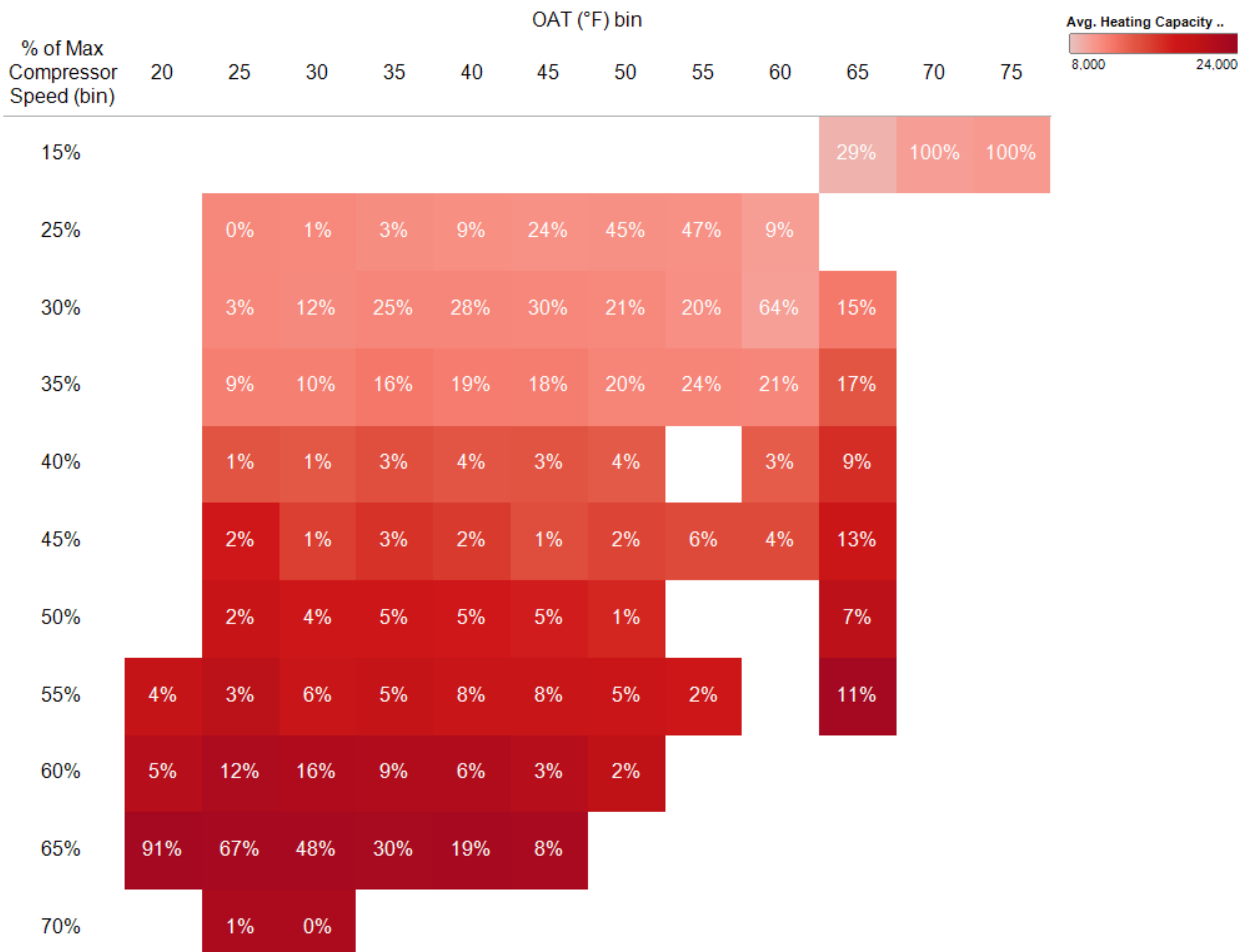

Figure 13. Space heating capacity for the 2nd generation GS-IHP field test prototype vs. outdoor air temperature bins and % of maximum compressor speed

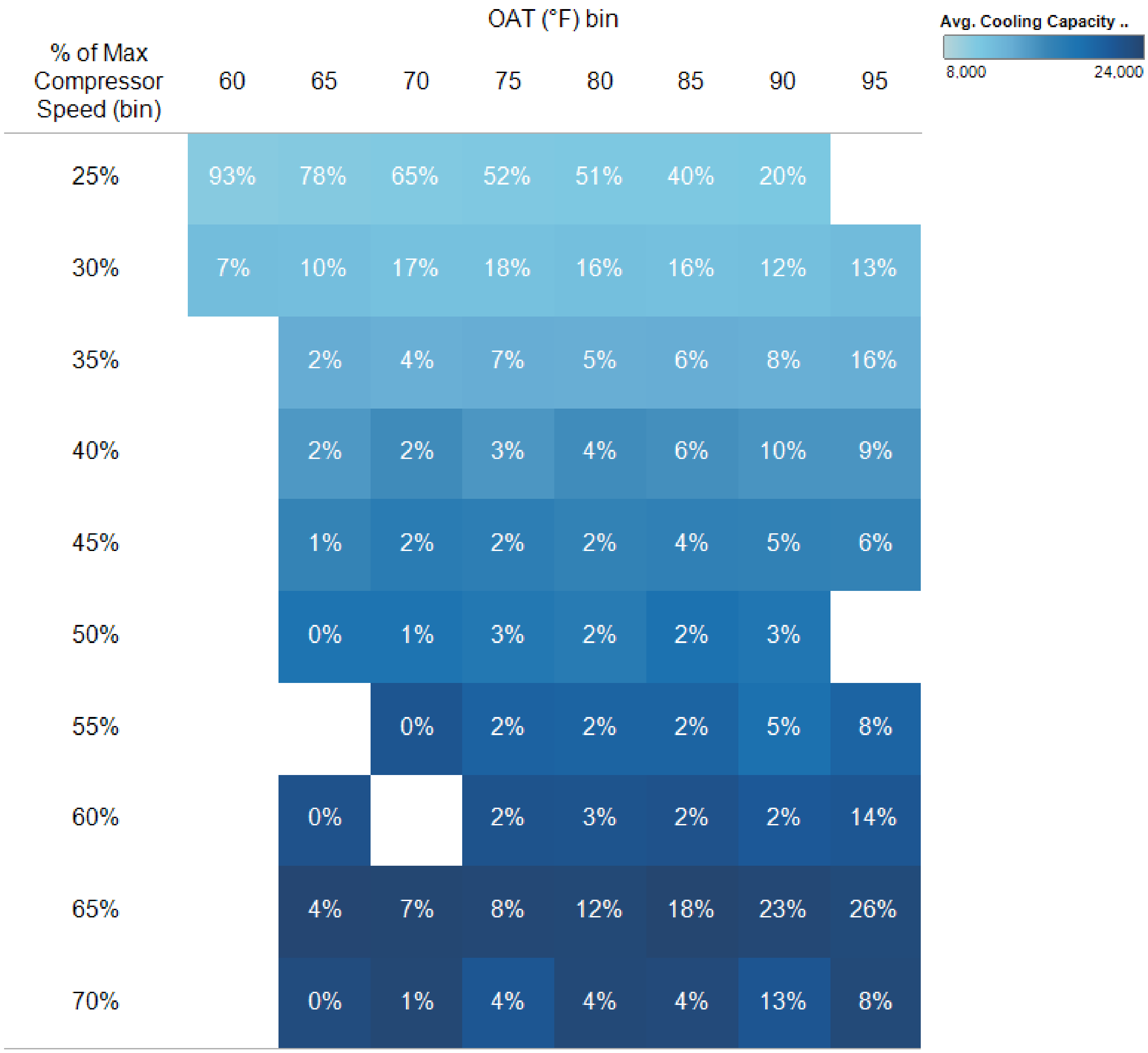

Figure 14. Space cooling capacity for the 2nd generation GS-IHP field test prototype vs. outdoor air temperature bins and % of maximum compressor speed

Figures 15-18 illustrate efficiencies demonstrated by the unit during the field test for each of the principal operating modes (space heating, space cooling, dedicated water heating, and combined space cooling & water heating, respectively). Here again most of the operation is seen to have been at lower compressor speed and/or higher system efficiency ranges.

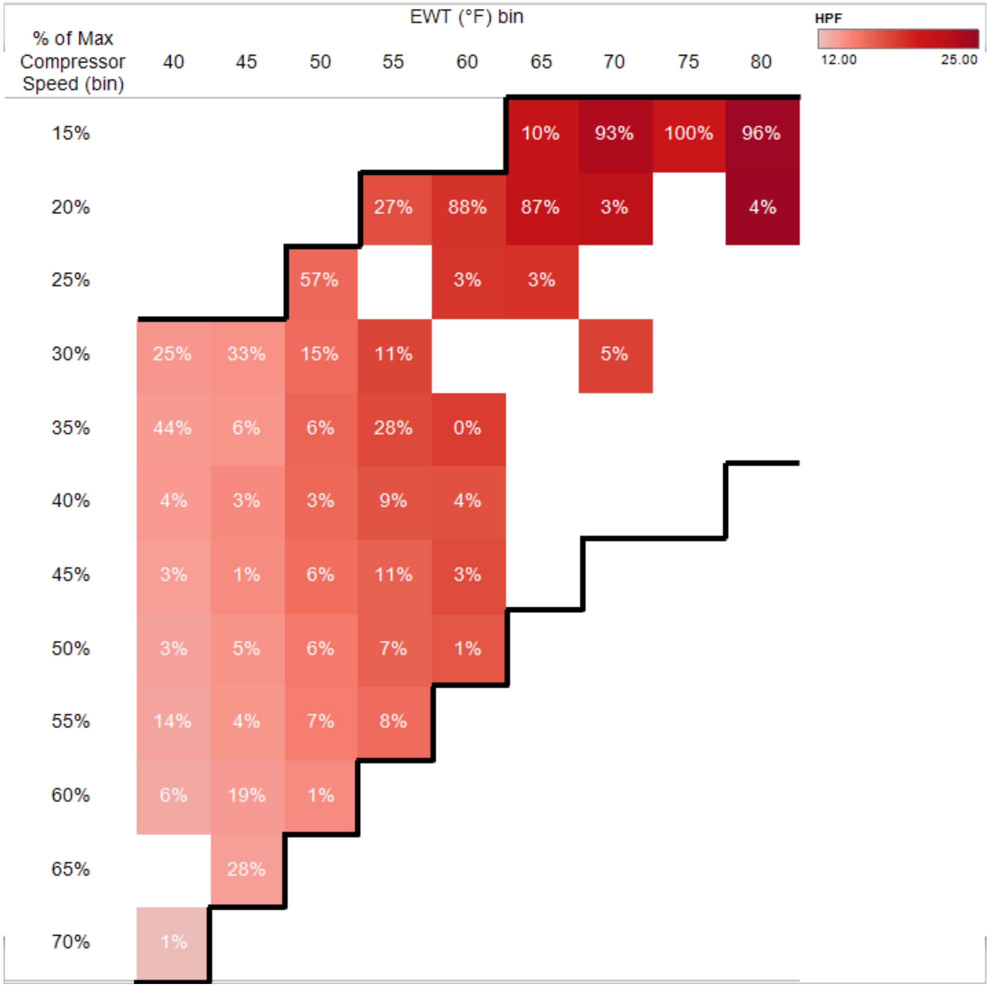

Figure 15. Space heating mode heating performance factor (HPF, Btu/Wh) for 2nd generation GS-IHP field test prototype vs. EWT and % maximum compressor speed

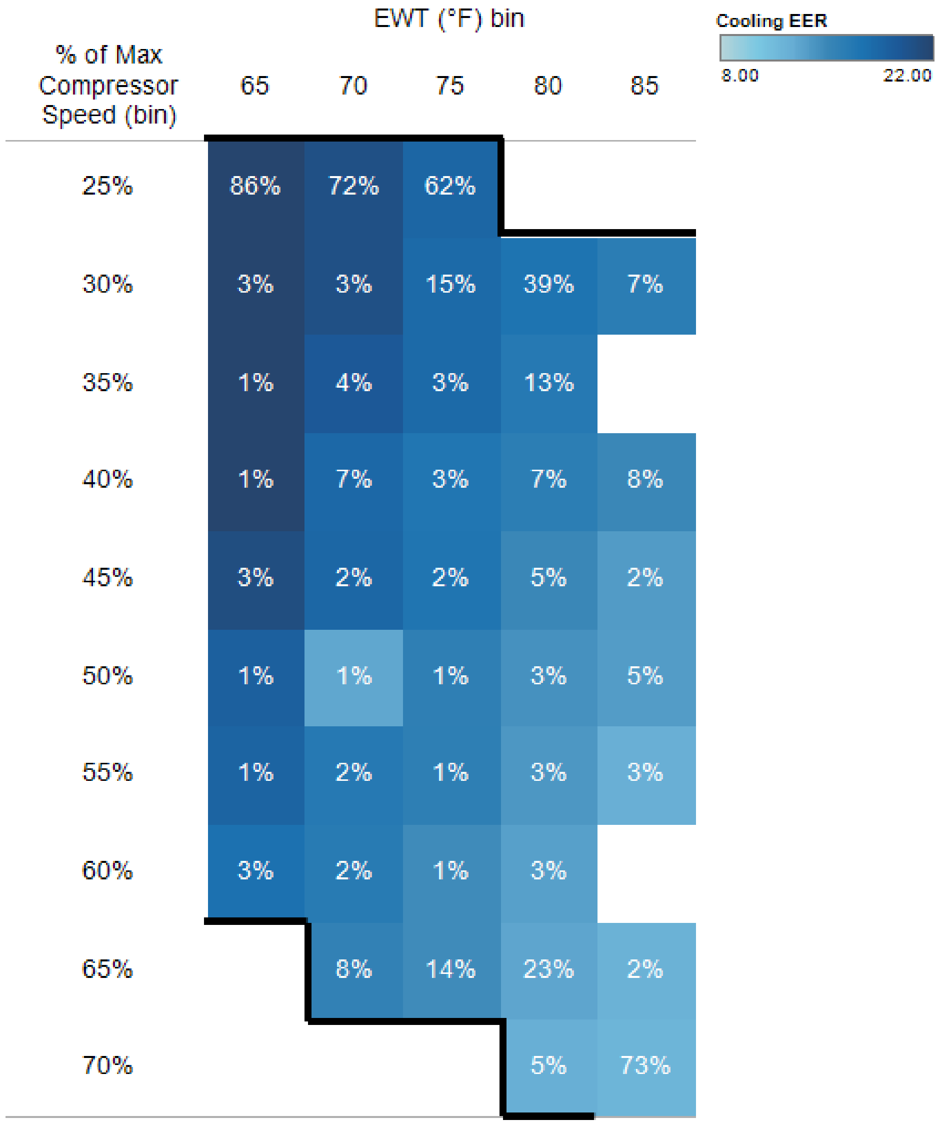

Figure 16. Space cooling mode EER (Btu/Wh) for 2nd generation GS-IHP field test prototype vs. EWT and % maximum compressor speed

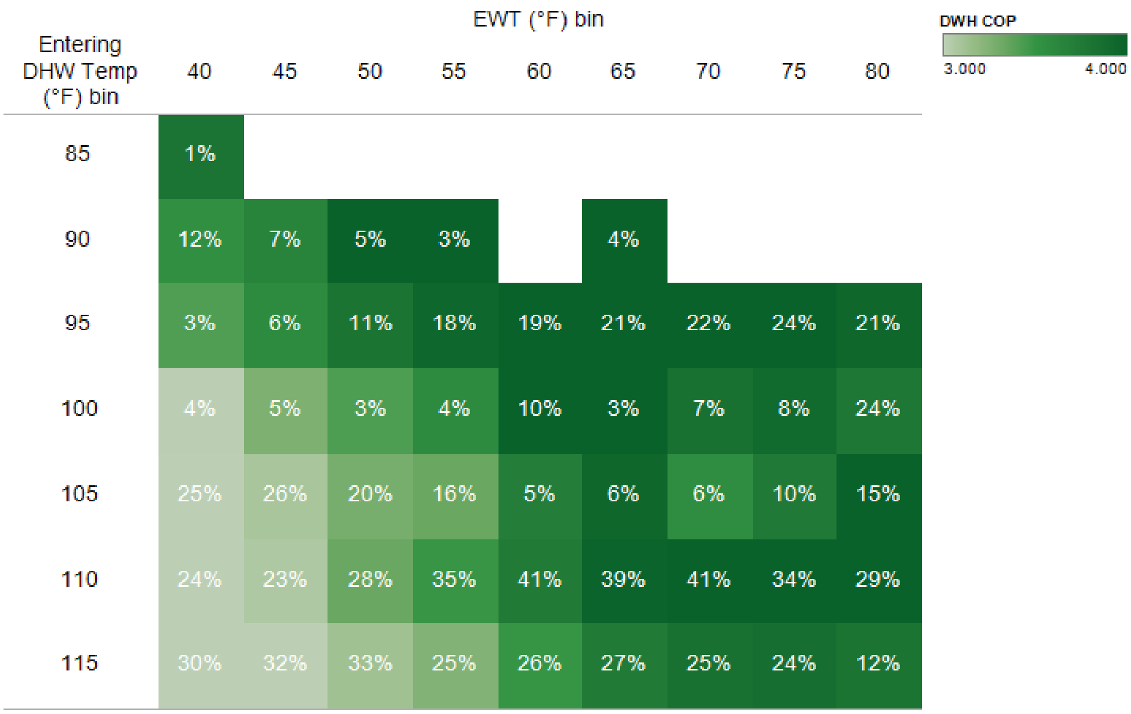

Figure 17. Dedicated WH mode COP vs. EWT and entering DHW temperature

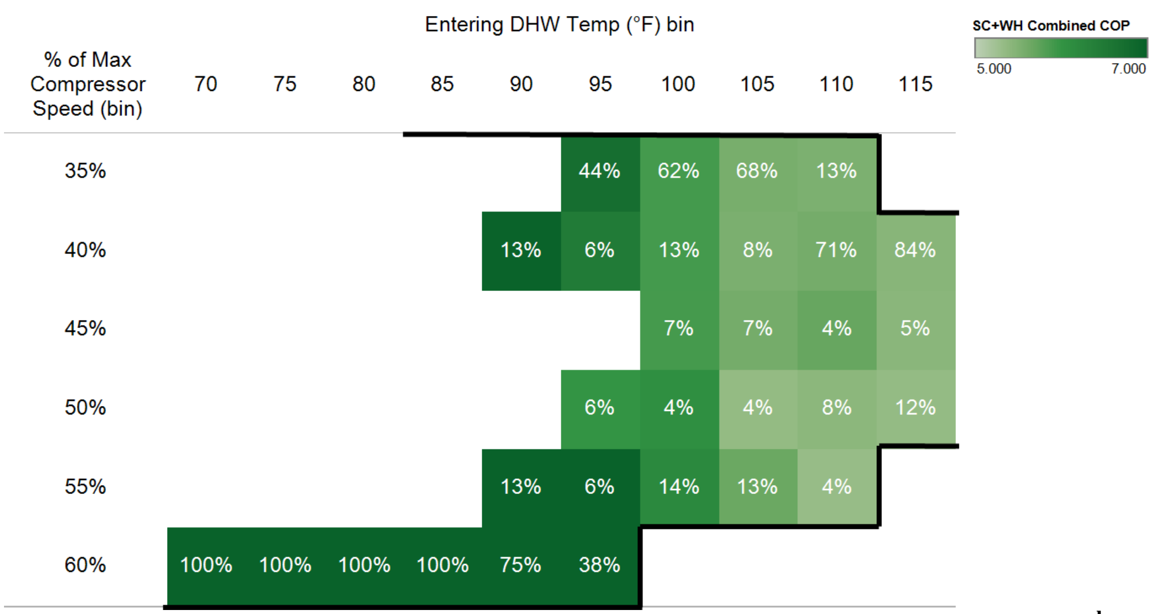

Figure 18. Combined space cooling and water heating mode COP for the 2nd generation GS-IHP field test prototype vs. entering DHW temperature and % of maximum compressor speed – combined EER range is ~17 to 24 Btu/WH

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APPENDIX A – March 2012 CM Press Release

FOR IMMEDIATE RELEASE
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ClimateMaster Breaks the 40 EER Barrier

Oklahoma City, OK March 19, 2012 - ClimateMaster announced an efficiency breakthrough with introduction of the Trilogytm 40 series, the first geothermal heat pumps ever certified by the Air Conditioning, Heating, and Refrigeration Institute (AHRI) to exceed 40 EER at ground-loop (GLHP) conditions.

The revolutionary new Trilogytm 40 utilizes variable speed technology to provide an extremely wide range of heating and cooling capacities, with the ability to perfectly match loads to as low as 30% of maximum. In addition, patent-pending Q-Modetm technology produces year-round domestic hot water on demand, even when space conditioning is not required.

The Trilogy 40 Q-Mode is the outcome of a five year collaboration between ClimateMaster and Oak Ridge National Laboratory (ORNL), which was sponsored by the U.S. Department of Energy (DOE) Building Technologies Program. Based on field tests and analysis by ORNL, the Trilogy 40 Q-Mode can save 55–65% of annual energy use and cost for space conditioning and water heating in residential applications versus new minimum efficiency (SEER 13) conventional systems and 30–35% versus current state-of-the-art two-stage geothermal heat pumps.

"ClimateMaster has a solid track record of leadership and innovation since its founding in 1957," said Daniel Ellis, President. "We are very proud to continue that legacy with the launch of the new Trilogy 40, which is 33% more efficient than any other geothermal heat pump available and the only one with Q-mode technology to provide year-round water heating."

In addition to efficiency, the Trilogy 40 Q-Mode delivers unsurpassed comfort and humidity control by precisely matching its capacity to the heating and cooling load. For installing and servicing contractors, it also offers the latest technology to configure and diagnose the system electronically using communicating controls and sensors that monitor every critical aspect of system operation to ensure peak performance.

"The Trilogy 40 Q-Mode represents a major breakthrough in comfort and efficiency" said John Bailey, Sr. Vice President of Sales and Marketing at ClimateMaster. "With variable speed fan, pump and compressor (Trilogy technology) plus four operating modes (Q-Mode technology), it far exceeds the capabilities of any other HVAC unit on the market today. Plus, it can completely eliminate the use of auxiliary heat even in far Northern climates."

The Trilogy^{tm} 40 series is currently in limited production, with full availability scheduled for late this year.

ClimateMaster, Inc. is the leading manufacturer of geothermal and water-source heat pumps, which are considered to be the most energy efficient and environmentally friendly type of heating and cooling systems available on the market today. Headquartered in Oklahoma City, OK, ClimateMaster, Inc. is a wholly owned subsidiary of LSB Industries, Inc. whose common stock is traded over the New York Stock Exchange under the symbol LXU. For more information, visit www.climatemaster.com.

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 ClimateMaster and ORNL – none

2. Disclosures by ORNL – none

3. Disclosures by ClimateMaster – U. S. patent submission 61/614,070