Development of a Residential Ground-Source Integrated Heat Pump

Keith Rice
Member ASHRAE
Tim McDowell
Member ASHRAE

Van Baxter
Fellow ASHRAE
Jeffrey Munk
Member ASHRAE

Shawn Hern
Member ASHRAE
Bo Shen
Member ASHRAE

ABSTRACT

A residential-size ground-source integrated heat pump (GSIHP) system has been developed and is currently being field tested. The system is a nominal 2-ton (7 kW) cooling capacity, variable-speed unit, which is multi-functional, e.g. space cooling, space heating, dedicated water heating, and simultaneous space cooling and water heating. High-efficiency brushless permanent-magnet (BPM) motors are used for the compressor, indoor blower, and pumps to obtain the highest component performance and system control flexibility. Laboratory test data were used to calibrate a vapor-compression simulation model (HPDM) for each of the four primary modes of operation. The model was used to optimize the internal control options and to simulate the selected internal control strategies, such as controlling to a constant air supply temperature in the space heating mode and a fixed water temperature rise in water heating modes.

Equipment performance maps were generated for each operation mode as functions of all independent variables for use in TRNSYS annual energy simulations. These were performed for the GSIHP installed in a well-insulated 2600 ft² (242 m²) house and connected to a vertical ground loop heat exchanger(GLHE). We selected a 13 SEER (3.8 CSPF)/7.7 HSPF (2.3 HSPF, W/W) ASHP unit with 0.90 Energy Factor (EF) resistance water heater as the baseline for energy savings comparisons. The annual energy simulations were conducted over five US climate zones. In addition, appropriate ground loop sizes were determined for each location to meet 10-year minimum and maximum design entering water temperatures (EWTs) to the equipment. The prototype GSIHP system was predicted to use 52 to 59% less energy than the baseline system while meeting total annual space conditioning and water heating loads.

INTRODUCTION

In an assessment of advanced HVAC/WH energy saving options for high-efficiency residences (Baxter 2005), variable-speed integrated heat pumps were identified as a leading candidate. Seeking to develop energy-saving electric-driven equipment designs capable of saving more than 50% energy relative to a baseline suite of minimum efficiency equipment, research was initiated on multifunction heat pump designs for air- and ground-source equipment (Murphy et al, 2007a) and a business case assessment was conducted (Baxter, 2007). Multifunction electric-driven heat pumps have the potential to make fuller use of high-efficiency higher cost variable-speed compressors, blowers, and pumps by meeting not only the sensible space conditioning loads but also the water heating and dehumidification loads. The variable-speed capability allows the larger space conditioning design loads to be met with higher speed operation, while the smaller water heating loads are met at lower speeds. Significant energy savings are possible from the higher efficiency operation of the

Keith Rice, Van Baxter, Jeff Munk and Bo Shen are research engineers in the Building Equipment Research Group, Building Technologies Research and Integration Center, Oak Ridge National Lab, TN, USA. Timothy McDowell is the Vice President of Thermal Energy System Specialists, LLC, WI, USA. Shawn Hern is an engineer in Climate Master, INC.

components, the load matching operation of the equipment (providing heat exchanger unloading benefits), and waste heat recovery in the combined space cooling and water heating mode. While the waste heat recovery is not free due to the elevated condensing temperatures required to meet domestic hot water needs in full condensing WH operation, the combined mode is quite efficient as both space cooling and full condensing water heating are delivered from one compressor power input

GSIHP EQUIPMENT DESIGN AND SIMULATION APPROACH

An investigation of the potential energy savings of a multifunction ground-source heat pump in zero-energy houses was reported by Murphy, et al (2007b). This research led to collaboration with a ground-source heat pump manufacturer to develop a design suitable for existing residential applications. A nominal 2-ton (7 kW) design cooling size was selected for development leading to the first prototype field testing. The design uses inverter-driven variable-speed 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 heat exchangers (HXs) were used for the ground loop and the domestic hot water (DHW) HXs with a tube-in-fin HX for the indoor coil.

One potential technical challenge in multi-function heat pumps is refrigerant charge management. For GSIHP units, this challenge is less relative to air-source designs because the refrigerant-to-water HX connected to the ground loop has about the same internal refrigerant volume as 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.

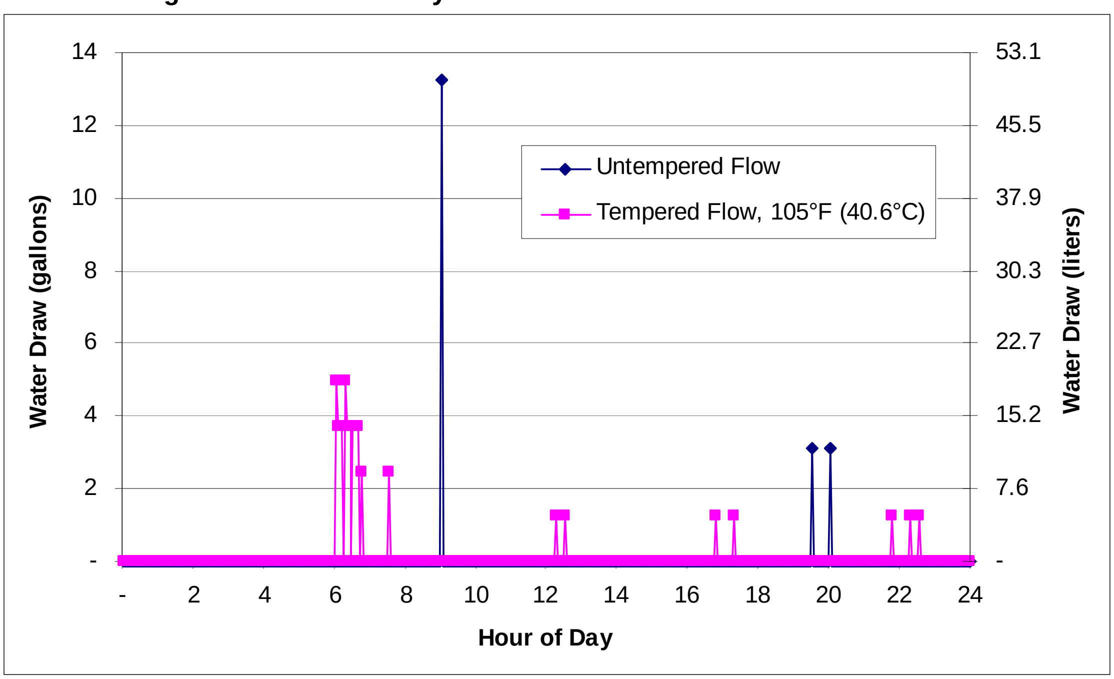
The prototype design was assembled by the manufacturer and tested in their laboratory over a wide range of ground-source conditions. We used the detailed lab measurements of refrigerant and source/sink conditions to calibrate the research version of HPDM (Rice et al, 2005) in each of the four operating 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 of 0.5 IWC (0.125 kPa) at 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 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 relative humidity (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 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 is adjusted with loop EWT in the dedicated WH mode to maintain a WH capacity of 5.3kW or higher which is greater than that of the largest element typically included in residential-sized electric resistance water heaters (4 to 5 kW).

Once the control approaches and calibration equations were complete, we used the HPDM to generate performance maps (i.e., tables) of capacities, powers, and mass flow rates for each mode as a function of all relevant independent variables, e.g., compressor speed, indoor DB, indoor RH, and equipment EWTs from the ground and/or DHW loops,. (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.) These performance maps were used as input to TRNSYS (Solar Energy Lab et al, 2010) using a custom interface and thermostat control logic and linked with standard TRNSYS house and DHW tank models for annual performance simulations. The selected house for the 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. The DHW tank was modeled using a TRNSYS Type 534 module, which models a vertical cylindrical water tank. The tank is divided into 6 isothermal temperature nodes (to model stratification observed in storage tanks) where each constant-volume node is assumed to be isothermal and interacts thermally with the nodes above and below through several mechanisms; heat conduction between nodes and through fluid movement (either forced movement from inlet flow streams or natural destratification mixing due to temperature inversions in the tank). Mechanical ventilation per ASHRAE STD 62.2 (2007) was assumed to be provided by continuous operation of a bathroom ventilation fan.

The time steps in TRNSYS for GSIHP seasonal performance analysis were 3.0 minutes between t-stat 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 1 includes discrete tempered and untempered hot water draws totaling 64.3 gal/day (243.4 l/day), which is consistent with the Department of Energy (DOE 2010) daily hot water draw totals for electric resistance and HPWH Energy Factor testing.

Figure 1. Assumed daily hot water draw schedule from DHW tank



BASELINE EQUIPMENT MODELING

To determine the energy savings potential of the GSIHP design, suitable baseline all-electric 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.

Next a high-efficiency 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 GSIHP 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 GSIHP 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. Far-field earth temperatures were based on average annual air temperatures. Ten-year sizing runs were made at multiple bore lengths for the GSIHP and two-capacity GSHPwDS 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 1 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 1. TRNSYS 10-Year bore sizing results for GSHPwDS and GSIHP 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 ft^2/day [mm^2/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 GSIHP. 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 requirements from the ground loop in the winter and shoulder months to meet the DHW load. In Atlanta, the required bore length for the GSIHP with the enhanced grout was 33% less than had standard grout been used; however, the annual energy use for the GSIHP was found to be nearly the same since both grouting cases stayed similarly within the min and max loop design temperatures. Had standard grout been used for the GSIHP Atlanta case, 25% more bore depth was predicted to be required than for the 2-capacity GSHPwDS case.

The relative bore depth requirements between the GSIHP and two-capacity GSHPwDS given in Table 1 show a 6% shorter bore for the GSIHP 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 GSIHP 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, especially in the colder climates. Cyclic losses are not included in the two-capacity GSHPwDS or variable-speed GSIHP 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 2, 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 parenthesis. Indoor RH control was comparable between the baseline ASHP, two-capacity GSHP, and GSIHP designs.

Predicted WH benefits from the two-capacity GSHPwDS cases are shown in Table 3 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 22.6%.

Predicted total savings for the GSIHP design is seen in Table 2 to range from 52.3% to 59.0%. 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%. Water heating savings relative to resistance units range from 68 to 78%.

Savings by mode for the GSIHP are summarized in Table 4 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, GSIHP water heating contributed 47 to 86% of the savings. As houses become tighter and better insulated and the sensible space conditioning loads decrease, the fraction of the total delivered energy from water heating increases, which provides higher total energy savings for GSIHP equipment.

Table 2. Energy Use and Savings 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 3. 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 4. Breakdown of Energy Savings 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%

Field testing of prototype 1 was conducted over the 2011 heating and cooling season in Oak Ridge, TN (Munk et al, 2011). A second prototype design has been developed and is being tested over the 2012-2013 cooling and heating seasons. This design includes control hardware and software modifications and uses brazed plate water-to-refrigerant HXs and a microchannel indoor coil which boosts full load cooling performance by over 10% relative to prototype 1 modeled in this work. Product availability to the market is expected by early 2013.

CONCLUSIONS

A prototype residential-sized variable-speed GSIHP has been developed, lab-tested, and modeled in TRNSYS to predict annual performance savings relative to a minimum efficiency combination of ASHP and resistance water heater and a high-efficiency two-capacity GSHPwDS. Predicted total annual energy savings for 5 U.S. locations ranged from 52.3 to 59.0%, averaging 54.9%, 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, for five Building America climate regions of mixed-humid, hot-humid, hot-dry, marine, and cold.

Planned next are performance mapping and TRNSYS predictions of annual energy savings for the higher-efficiency second prototype design based on recent laboratory test data. Work is underway as well to add integrated heat pump simulation capability to EnergyPlus. Activity is ongoing to develop a test standard for integrated heat pumps of both air- and ground-source designs in ASHRAE Standard Project Committee 206. Market introduction of this GSIHP is expected by early 2013.

ACKNOWLEDGMENTS

This work was sponsored by the DOE Building Technologies Program under contract DE-AC05-00OR22725 with UT-Battelle, LLC.

REFERENCES

ASHRAE. 2007. Ventilation and acceptable indoor air quality in low-rise residential buildings. ASHRAE Standard 62.2-2007, Atlanta, GA: American Society of Heating, Refrigerating and Air-Conditioning Engineers

Baxter, V. (2005). HVAC equipment design options for near-zero-energy homes – a stage 2 scoping assessment. ORNL/TM-2005/194. November.

Baxter, V.D. 2007. Integrated heat pump HVAC systems for near-zero-energy homes –business case assessment, ORNL/TM-2007/064, May.

DOE, U.S. Government. 2012. Climate zones. http://www1.eere.energy.gov/buildings/residential/ba_climate_zones.html.

DOE, U.S. Government. 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.

International Organization for Standardization,1998. ISO 13256-1, Water-Source Heat Pumps – Testing and Rating for Performance – Part 1: Water-to-Air and Brine-to-Air Heat Pumps, Case Postale 56, CH-1211, Geneva 21 Switzerland.

Munk, J.D., Ally, M.R., and Baxter, V.D. 2011. Ground-source heat pump field tests in high efficiency residential buildings, presentation at European Heat Pump Summit 2011, Nuremburg, Germany, September.29, http://info.ornl.gov/sites/publications/Files/Pub32040.pdf

Murphy, R. W., Rice, C. K. and Baxter, V. D. 2007a. Integrated Heat Pump (IHP) System Development; Air-Source IHP Control Strategy, and Specifications and Ground-Source IHP Conceptual Design — FY06 Milestone Report. ORNL/TM-2006/143, May.

Murphy, R.W. Baxter, V. D., Rice, C. K., and Craddick, W.G. 2007b. Ground source integrated heat pump for near zero energy houses: technology status report. ORNL/TM-2007/177, December

Rice, C. K. and Jackson, W. L. 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

Wetter, M., "GenOpt® Generic Optimization Program User Manual Version 3.0.0", May 11, 2009, Lawrence Berkeley National Laboratory Technical Report LBNL-2077E