INTEGRATED HEAT PUMP SYSTEM TECHNOLOGY DEVELOPMENT FOR NET ZERO ENERGY HOME (ZEH) APPLICATIONS

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Introduction and Summary Results

The US Department of Energy's (DOE's) strategic goal in the buildings technology area is to develop ZEH or net ZEH technology by 2020. A net ZEH is defined as "a home with greatly reduced needs for energy through efficiency gains (60% to 70% less than conventional practice), with the balance of energy needs supplied by renewable technologies." To achieve this goal will require energy service equipment that can meet the space heating and cooling (SH and SC), ventilation (V), water heating (WH), dehumidification (DH), and humidification (H) needs while using 50% less energy than current equipment. One promising approach to meeting this requirement is through an integrated appliance or "integrated heat pump" (IHP). The energy benefits of an IHP stem from the ability to utilize otherwise wasted energy (e.g., heat rejected by the space cooling operation can be used for water heating) and from the ability to justify the cost of more expensive, more energy efficient components because they serve multiple functions (e.g., a variable speed compressor is used to both provide space conditioning and water heating). An integrated heat pump can be designed to be air-coupled or ground-coupled. Based on a scoping study of a wide variety of possible approaches to meet the energy service needs for a ZEH, DOE selected the IHP concept as the most promising, and is supporting the development of both air-source and ground-source versions (Baxter 2005). This paper summarizes the development of IHP technology aimed primarily at future ZEH applications.

A laboratory prototype was developed and tested over a range of operating modes and conditions. Test data was used to validate a detailed heat pump system model - the DOE/ORNL Mark VI Heat Pump Design Model (HPDM). HPDM was then linked to TRNSYS, a time-series-dependent simulation model. The experimentally validated analytical tool was used to calculate the yearly performance of IHP system

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designs optimized for R-410A in five major cities, representing the main climate zones within the United States: Atlanta (mixed-humid), Houston (hot-humid), Phoenix (hot-dry), San Francisco (marine) and Chicago (cold). The calculations extended for a full year using 3-minute time steps. For the air-source IHP version, the simulation results showed ~46-67% energy savings depending upon location. For the ground-source IHP version, the simulation showed over 50% savings in all locations - ~52-65% range. Initial cost analyses (based on 2006 equipment costs and electricity prices) yielded estimated simple paybacks of the IHP systems vs. a baseline HVAC/WH/DH/H system *in a net ZEH* - about 5 to 10 years for the air-source IHP and 6.5 to 14 years for the ground-source IHP (with vertical bore ground HX).

IHP Concept

Net zero energy homes (ZEH) have specific requirements for meeting SC, SH, WH, V, DH, and H loads. First, ZEHs have tighter, less conductive house envelopes resulting in reduced SC and SH demands and, therefore, require smaller equipment capacities than current homes. Second, tighter construction means less natural air infiltration, and forced ventilation is generally necessary to meet accepted residential standards for fresh air. Moreover, bringing moist ventilation air to space neutral conditions increases the need for latent cooling. And third, the water heating load, which depends largely on the number of occupants in the dwelling and their washing requirements, remains essentially unchanged. Consequently, the water heating load tends to become a larger portion of the overall energy service demands of a net ZEH. These requirements suggest that a small capacity integrated load-following system would be an effective way to meet the ZEH energy service needs. Such a system based on the demonstrated high efficiency of vapor compression technology and denoted as the “integrated heat pump” (IHP) here, would provide a single appliance capable of meeting ZEH requirements. As noted the IHP could utilize either outdoor air (air-source) or the earth (ground-source) as the heat source/sink.

The current ground-source IHP system concept is indicated schematically in Fig 1 and incorporates three separate but interactive loops, one refrigerant, one domestic hot water, and one ground heat exchanger (HX with water or an antifreeze/water mixture for cold climates). Major electrical energy-consuming components are one variable speed compressor (C), one variable speed indoor blower (FI), and two pumps—one single speed pump (PI) for the domestic hot water loop and one multiple-speed pump (PO) for the ground HX loop (GC). Four internal HXs are included to meet the space conditioning and water heating loads: one refrigerant-to-air (fan coil, HXRAI), one water-to-air (tempering, HXWA), and two refrigerant-to-water (domestic hot water interface, HXRWI, and ground coil interface, HXRWO). Remaining major components shown include a reversing valve (RV) and refrigerant expansion valve (EV). Separate indoor and outdoor EVs are shown but a single, bi-directional EV could be used as well. Outdoor ventilation air is drawn in through a duct with flow control damper (not shown), mixed with recirculating indoor air, and distributed to the space via the blower, FI. HXWA uses hot water generated by heat recovery in the SC and DH modes and stored in the hot water tank (WT) to temper the circulating air stream, as needed, to meet space neutral temperature requirements. Modulation of compressor speed and indoor fan speed can be used to control both supply air humidity and temperature as required. With this arrangement, water heating and air tempering can be accomplished simultaneously. The air-source IHP concept is similar with the GC loop and PO and HXRWO items replaced with an outdoor refrigerant-to-air HX and variable speed fan. Murphy, et al (2007a and 2007b) provides a more complete description of the air-source and ground-source IHP concepts.

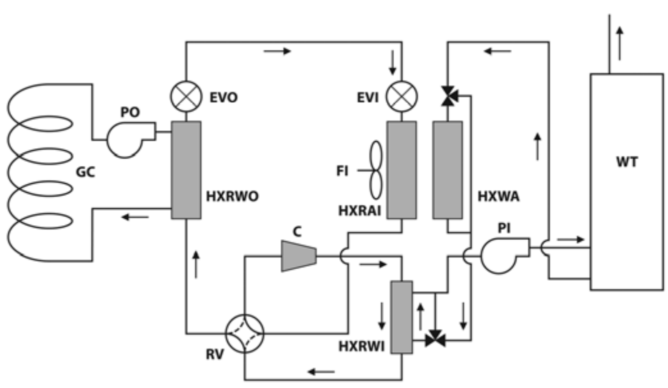

Figure 1 Schematic of ground-source IHP concept (space cooling plus water heating mode shown)

IHP Prototype Testing

A laboratory prototype air-source IHP system was constructed and instrumented as shown in the schematic Figure 2. The prototype was installed in a two-room environmental chamber and tested over a range of operating conditions and modes. Due primarily to availability of suitable components at the required sizes at the time, this prototype system used R-22 as the refrigerant. Murphy et al (2007a) provides a detailed description of the test set up and instrumentation as well as the test results.

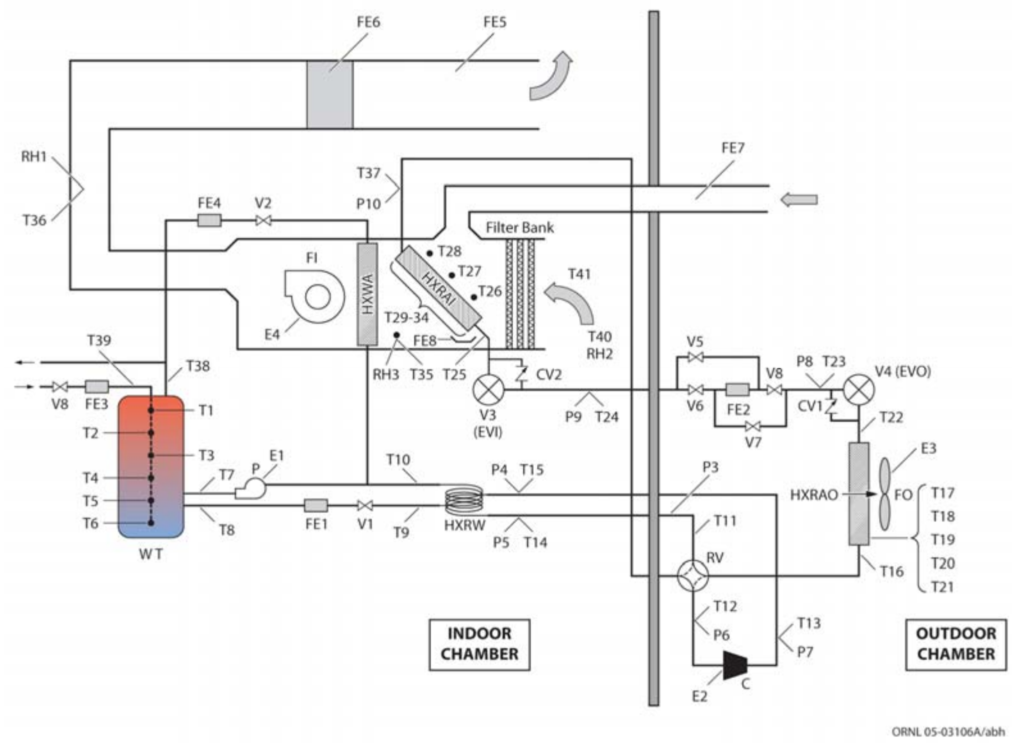

Figure 2 IHP prototype component and instrumentation location diagram – T=temperature; P=pressure; FE=flowmeter; E=Power; RH=humidity

Initial steady-state tests were conducted in cooling mode to determine the most suitable indoor airflow, compressor speed and refrigerant charge at the 35°C ambient design condition. Following these four tests were conducted at the four outdoor dry bulb temperature (DBT) conditions prescribed for rating variable speed cooling systems in the US. An additional test was one run with indoor airflow reduced by 30% to determine the amount of improved dehumidification. The SHR decreased by 10%. Table 1 shows the results from these tests.

Table 1 Steady-state space cooling performance

Outdoor DBT (C)	Outdoor fan airflow (m^3/s)	Indoor DBT/WB (C)	Indoor Airflow (m^3/s)	Compressor speed (Hz)	Cooling capacity (W)	COP or EER (W/W)	SHR
35.0	0.535	26.7/19.4	0.230	79	4363	3.52	0.743
30.6	0.469	26.7/19.4	0.165	58	3147	4.34	0.749
27.8	0.401	26.7/19.4	0.115	36	2115	5.45	0.739
19.4	0.394	26.7/19.4	0.114	36	2185	6.92	0.727
27.8	0.388	26.7/19.4	0.080	36	1961	4.98	0.665

A number of simultaneous space cooling and water heating were conducted as well. For these tests, fixed water temperatures into the tank were maintained. The performance of the IHP in this limited test series illustrates the efficiency advantage of recovering normally rejected heat to provide water heating, with an overall COP (space cooling and water heating) of almost 10. Test results are shown in Table 2.

Table 2 Steady-state space cooling + water heating performance

COP: Space cooling	4.94	5.03	5.05	4.99
COP: Space cooling + water heating	9.45	9.71	9.71	9.62
Water Heating Only COP	4.52	4.68	4.66	4.63
Heat to water using R-W HX (W)	1603	2914	2784	2851
Cooling to space (W)	2071	3124	3013	3072
Sensible heat ratio (SHR)	0.739	0.732	0.775	0.775
Avg. tank temperature (C)	28.8	21.7	21.2	22.0
Avg. compressor power (W)	372.8	569.4	541.3	558.2
Avg. pump power (W)	30.4	28.5	30.6	31.0
Avg. indoor fan power (W)	18.5	26.1	27.3	28.4
outdoor ambient temperature (C)	27.8	27.8	30.6	30.6

Dynamic water heating tests (water heated from starting cold condition to fully heated) were conducted as well. One of the major design considerations with the IHP (and with all water-heating heat pumps) is to accomplish water heating using the compressor without exceeding the compressor discharge pressure maximum imposed by the manufacturer. Example results of this analysis are shown in Figure 3. The envelopes cover acceptable compressor discharge and suction pressures for the R22 variable-speed compressor used in the lab test IHP. The blue envelope covers acceptable conditions for the minimum and maximum compressor speeds - 30 Hz and 100 Hz. The red region contains acceptable compressor conditions in the range 45-90 Hz. It can be seen that compressor discharge and suction pressures remain within the acceptable operating envelope as the tank heats up.

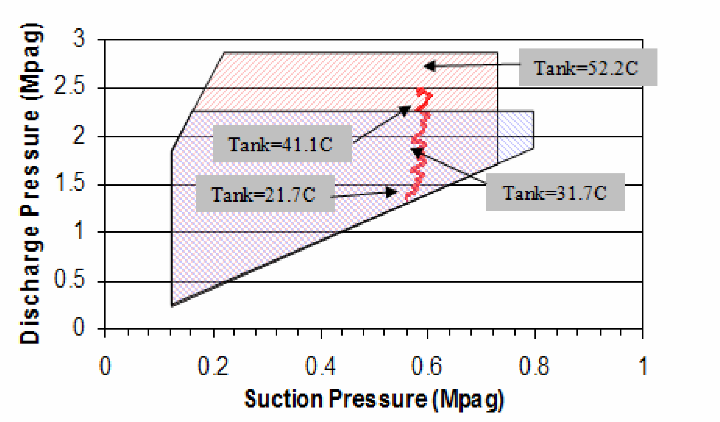

Figure 3 Tank water heat-up process (red line) superimposed on compressor discharge/suction pressure map

Tests were also run to examine the dehumidification performance of the IHP. By design, the IHP dehumidifies the return indoor air then reheats the air by passing hot water from the water tank through the reheat coil. The design point is that with 49 °C inlet water to the tempering coil, there would be sufficient reheating of the air leaving the indoor coil to establish space-neutral conditions.

Performance Analysis

HPDM Calibration

The lab prototype IHP performance data were used to calibrate the predictions of the DOE/ORNL Heat Pump Design Model or HPDM (Rice and Jackson 2002). The measured refrigerant and indoor air flows were used with a data reduction program to calculate the delivered capacities of system HXs, the heat losses and gains and pressure losses in the connecting lines and to deduce the airflows across the outdoor coil at various fan speeds from the condenser energy balance. The performance map for the lab prototype compressor was adjusted for the effects of inverter efficiency, and reduced speeds based on the measured power and mass flow data. This adjusted compressor map was input to the HPDM and initial predictions of the lab tests conducted. HPDM predictions were compared to the actual lab results and, through an iterative process, the HPDM predictions were calibrated to the range of space cooling and water heating tests performed.

Using the calibrated HPDM, IHP design optimization and control assessments were conducted to establish target optimized compressor and fan speed control relationships based on the laboratory R-22 compressor, air-moving, and heat exchanger components. Subsequently a suitable compressor map for a state-of-the-art R-410A variable-speed rotary compressor was obtained and input to the calibrated HPDM. Revised target performance ranges were then established for both the air-source and ground-source IHPs using this preferred HFC refrigerant R-410A. Figures 4 and 5 illustrate a sample of the results for the air-source and ground-source IHPs, respectively.

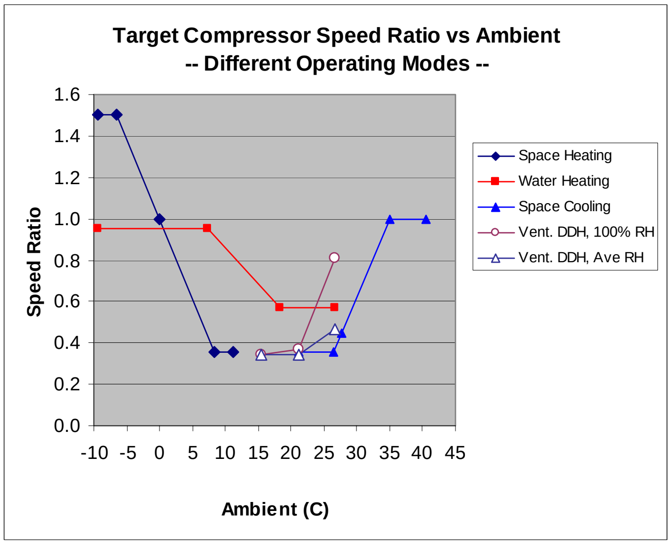

Figure 4 Air-source IHP target compressor speed ratios for various operating modes vs. ambient with R-410A refrigerant (DDH = dedicated dehumidification; RH = relative humidity)

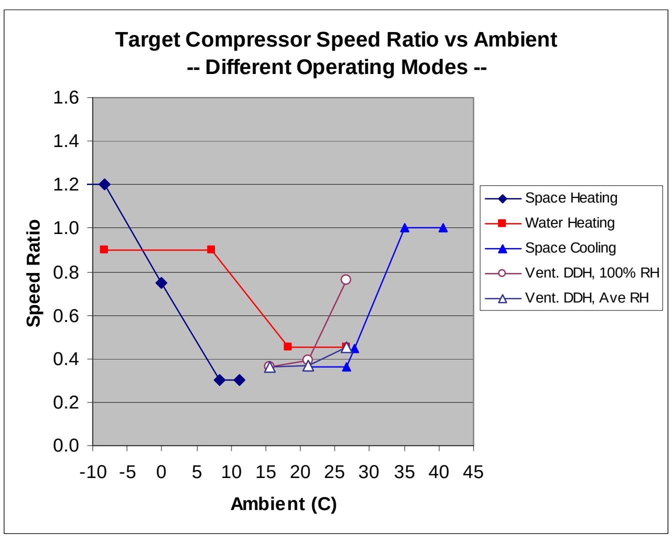

Figure 5 Ground-source target compressor speed ratios for various operating modes vs. ambient with R-410A refrigerant

TRNSYS/HPDM Simulation Approach

Once the HPDM was calibrated the next challenge was to estimate the IHP annual energy use in a net ZEH for a range of climates representative of most US locations. A sub-hourly analysis tool was needed to most accurately account for the competing IHP operating modes, and representative inlet conditions that will be seen simultaneously by the system HXs while heating water. This was accomplished linking the HPDM with TRNSYS (Solar Energy Laboratory, et al 2006). An extensive effort was undertaken to couple the two codes that the outputs of the TRNSYS from modeling the time-dependent ZEH indoor space and water heater conditions would become inputs to the HPDM. In turn, the HPDM output conditions of the indoor air and water leaving the equipment heat exchangers are coupled back to the TRNSYS house and water heater modules to update their operating states. Further details of the house and controls modeling and the HPDM/TRNSYS linkage approach are described by Murphy et al (2007a and 2007b).

Annual Performance

The TRNSYS/HPDM tool was used to simulate the annual performance of air-source and ground-source R-410A compressor based IHP systems for a 167 m^2 net ZEH in five climates. TRNSYS was also used to simulate a suite of baseline equipment for use in determining the potential energy savings of the IHP systems in each location as described below.

Baseline HVAC/WH System

A standard split-system air-to-air heat pump with USDOE-minimum required efficiency (SEER 13 and HSPF 7.7) provides space heating and cooling under control of a central thermostat that senses indoor space temperature. It also provides dehumidification (DH) when operating in space cooling mode but does not separately control space humidity. A standard electric storage water heater (WH) with USDOE minimum mandated energy factor (EF=0.90) provides domestic hot water needs. Ventilation meeting the requirements of ASHRAE Standard 62.2-2004 (ASHRAE 2004) is provided using a central exhaust fan. A separate stand-alone dehumidifier (DH) is used to meet house dehumidification needs during times when the central heat pump is not running to provide space cooling. A DH efficiency or energy factor (EF_d) of 1.4 L/kWh (0.0014 m^3/kWh) was used based on the USDOE proposed minimum requirement for 2012. A whole-house humidifier (H) accessory was included with the heat pump to maintain a minimum 30% relative humidity (RH) during the winter. Hot water from the WH tank was used for the humidifier supply based on manufacturer specifications for application with heat pump systems. The type humidifier adopted for the analyses reported herein consumes no power other than a negligibly small amount needed to operate the water flow control solenoid valve.

Baseline system control set points used in the TRNSYS simulation were as follows – 21.7°C ±1.4°C and 24.4°C ±1.4°C for first stage space heating and cooling, respectively; 18.9°C ±1.1°C for second stage space heating (electric back up heater); 48.9°CF ±2.8°C for WH; 55% RH ±4% for DH; and 34% RH ±4% for H.

Air-Source and Ground-Source IHPs

The air-source IHP, illustrated in Figure 6, uses one variable-speed (VS) modulating compressor, two VS fans, a single-speed pump, and a total of four HXs (two air-to-refrigerant, one water-to-refrigerant, and one air-to-water) to meet all the HVAC and water heating (WH) loads. A WH tank (same size as for baseline) is included for hot water storage. The same type humidifier as used for the baseline system heat pump was included with the IHP as well. Ventilation (V) air is drawn into the IHP air handler section through a modulating damper as shown.

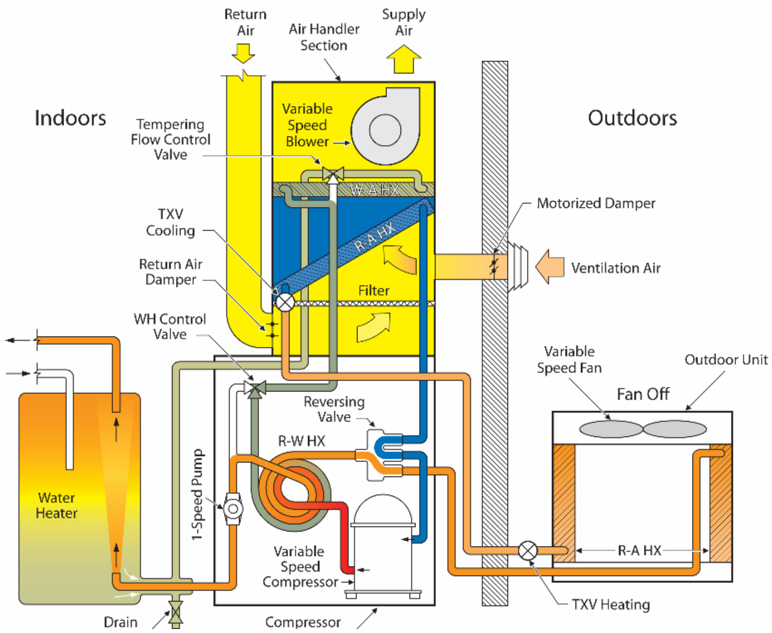

Figure 6 Air-source IHP - dedicated dehumidification and water heating mode shown

The ground-source IHP, illustrated in Figure 7, uses the same set of components as the air-source version with the outdoor section (outdoor air HX and fan) replaced with a multiple-speed pump and ground HX.

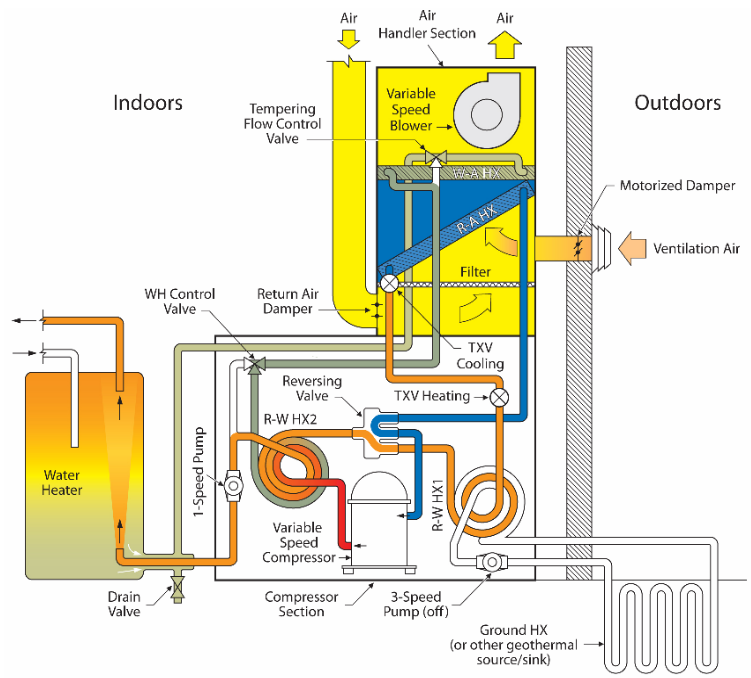

Figure 7 Ground-source IHP - dedicated dehumidification and water heating mode shown

The set points for 1st and 2nd stage space heating, space cooling, DH, and H as used for the baseline were also used for the IHPs. For WH, the 1st stage (IHP water heating) set point was 46.1°C ±2.8°C with a 2nd stage set point of 41.9°C ±1.4°C to control an electric resistance back up heating element in the upper portion of the WH tank. The 2nd stage WH set point was intentionally set lower than the 1st stage set point to maximize the amount of water heating supplied by the IHP.

Analysis Results and Preliminary Economics

Table 3 provides summary results of TRNSYS/HPDM sub-hourly simulations for the baseline HVAC system for the net ZEH for each of the five locations examined in this study. Tables 4 and 5 provide results for the air-source and ground-source IHPs, respectively, including hourly integrated peak kW demand for winter and summer (W/S). Maximum peaks occurred in the winter, generally in December or January. Summer peaks are somewhat lower and generally occurred in July or August.

Table 3 Annual site HVAC/WH system energy use and peak for 167-m^2 ZEH house with Baseline system

Location	Heat pump cooling capacity (kW)	HVAC site energy use, kWh	HVAC peak integrated hourly kW*
Atlanta	4.40	7230	8.6/4.6
Houston	4.40	7380	6.1/4.4
Phoenix	5.28	6518	6.1/3.9
San Francisco	3.52	4968	5.7/5.6
Chicago	4.40	10773	9.7/6.1

*the two values for peak in Tables 3-5 are for winter and summer (W/S)

Table 4 Estimated annual site HVAC/WH system energy use and peak for 167-m^2 ZEH with air-source IHP

Location	Heat pump cooling capacity (kW)	HVAC site energy use, kWh	HVAC peak integrated hourly kW*	% energy savings vs. Baseline HVAC
Atlanta	4.40	3349	2.2/1.5	53.7
Houston	4.40	3418	1.9/1.1	53.7
Phoenix	5.28	3361	2.1/1.7	48.4
San Francisco	3.52	1629	1.8/1.6	67.2
Chicago	4.40	5865	7.3/1.6	45.6

Table 5 Estimated annual site HVAC/WH system energy use and peak for 167-m^2 ZEH with ground-source IHP

Location	Heat pump cooling capacity (kW)	HVAC site energy use, kWh	HVAC peak integrated hourly kW*	% energy savings vs. Baseline HVAC
Atlanta	4.40	3007	2.0/1.1	58.4
Houston	4.40	3290	1.8/1.1	55.4
Phoenix	5.28	2909	1.7/1.2	55.4
San Francisco	3.52	1699	1.8/1.6	65.8
Chicago	4.40	5126	6.9/1.7	52.4

Along with the performance analyses above, a preliminary assessment of the system costs and payback for the IHPs vs. the baseline has been completed as well. Murphy , et al (2007b) provides full details of the cost estimation. A summary of the cost study is given below. Table 6 provides the baseline system costs for each of the five locations used in this study. Table 7 provides the estimated cost for the air-source IHP along with its energy cost savings and estimated simple payback vs. the baseline. For the ground-source IHP a vertical bore ground HX configuration was assumed. Installed cost in 2006US$ of the ground HX (including hookup to the IHP package) was estimated at ~$16.40/m ($5/ft) of bore. Table 8 gives the estimated bore lengths for a vertical GHX in each of the five cities as derived from long-term sizing runs using the TRNSYS/HPDM model. Sizing was based on limiting the long-term entering water temperature (EWT) to the IHP from the GHX to a maximum of 35°C during cooling operation in all cities. For heating operation, the long-term minimum EWT criteria was 5.6°C (using water as the GHX fluid) for all cities except Chicago where the minimum EWT criteria was -1.1°C (using a 20% propylene glycol brine solution). Cost estimates for the ground-source IHP in each city are given in Table 9. Energy cost savings and estimated simple paybacks vs. the baseline are included. The energy cost savings for each city were calculated based on 2006 electricity prices - $0.0872/kWh for Atlanta, $0.108/kWh for Houston, $0.0896/kWh for Phoenix, $0.1196/kWh for San Francisco, and $0.0844/kWh for Chicago.

Table 6 Estimated installed costs of baseline HVAC/WH system (2006 US dollars)

City	Heat pump cooling capacity (kW)	Heat pump cost	DH cost	WH cost	Vent fan cost	H cost	Total cost
Atlanta	4.40	$3985-4590	$415	$503	$305	$200	$5408-6013
Houston	4.40	$3985-4590	$415	$503	$305	$200	$5408-6013
Phoenix	5.28	$3995-4628	$415	$503	$305	$200	$5418-6051
San Francisco	3.52	$3974-4578	$415	$503	$305	$200	$5397-6001
Chicago	4.40	$3985-4590	$415	$503	$305	$200	$5408-6013

Table 7 Estimated installed costs and payback for air-source IHP (2006 US dollars)

City	Heat pump cooling capacity (kW)	Total cost		Premium over baseline system		Energy cost savings	Simple payback over baseline system, years	
		Low	high	low	high		low	High
Atlanta	4.40	$7,582	$8,786	$2,174	$2,773	$338	6.4	8.2
Houston	4.40	$7,582	$8,786	$2,174	$2,773	$428	5.1	6.5
Phoenix	5.28	$7,596	$8,862	$2,178	$2,811	$283	7.7	9.9
San Francisco	3.52	$7,568	$8,762	$2,171	$2,761	$399	5.4	6.9
Chicago	4.40	$7,582	$8,786	$2,174	$2,773	$414	5.2	6.7

Table 8 Estimated total bore lengths and installed costs for vertical ground HXs (2006 US dollars)

City	Total bore length, m	Installed cost
Atlanta	110	$1800
Houston	110	$1800
Phoenix	154	$2525
San Francisco	110	$1800
Chicago	100	$1640

Table 9 Estimated installed costs and payback for ground-source IHP (2006 US dollars)

City	Heat pump cooling capacity (kW)	Total cost		Premium over baseline system		Energy cost savings	Simple payback over baseline system, years	
		low	High	low	high		Low	High
Atlanta	4.40	$8,671	$9,748	$3,263	$3,735	$368	8.9	10.1
Houston	4.40	$8,671	$9,748	$3,263	$3,735	$442	7.4	8.5
Phoenix	5.28	$9,410	$10,549	$3,992	$4,498	$323	12.3	13.9
San Francisco	3.52	$8,657	$9,724	$3,260	$3,723	$391	8.3	9.5
Chicago	4.40	$8,511	$9,588	$3,103	$3,575	$477	6.5	7.5

Conclusions and Future Direction

The following specific conclusions are highlighted.

1. The air-source IHP system (using R410A) simulation results showed ~46-67% energy savings depending upon location. For the ground-source IHP version, the simulation showed over 50% savings in all locations - ~52-65% range. These predicted savings are obtained with active indoor RH control applied throughout the year.

2. Initial cost analyses (based on 2006 equipment costs and electricity prices) yielded estimated simple paybacks of the IHP systems vs. a baseline HVAC/WH/DH/H system *in a net ZEH* - about 5 to 10 years for the air-source IHP and 6.5 to 14 years for the ground-source IHP (with vertical bore ground HX).

As noted, all R&D conducted thus far for the IHP has been aimed at the net ZEH application. However, to achieve penetration in the current US housing market, with houses significantly different from the ultimate ZEH goal, modifications to these recommendations to produce a product optimized for the current market will be needed. These modifications may include eliminating some functions and substitution of less expensive components to produce a simpler, less-expensive product for initial market penetration. There are, however, certain portions of the current market that might provide a nearer-term market that could induce manufacturers to produce such futuristic equipment, especially for those consumers who desire to be the first to own the latest in energy-efficient, or "green" systems. These include multiple-story houses with independent small heat pumps for each floor, relatively small attached or condominium-style housing units, etc. Working in the future with manufacturing partners toward these market segments is planned.

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