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Air-Source Integrated Heat Pump for Net-Zero-Energy Houses: Technology Status Report

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LIST OF ABBREVIATED TERMS

AC	Space cooling (operating mode)
ACEAD	Space cooling with enhanced dehumidification (operating mode)
ACEADWH	Space cooling with enhanced space dehumidification plus water heating (operating mode)
ACWH	Space cooling plus water heating (operating mode)
AD	Dehumidification (operating mode)
ADWH	Dehumidification plus water heating (operating mode)
AH	Space heating (operating mode)
AHWH	Space heating plus water heating (operating mode)
ARI	Air-Conditioning and Refrigeration Institute
AS-IHP	Air-source heat pump
AV	Ventilation (operating mode)
AVVAD	Ventilation with ventilation air dehumidification (operating mode)
BDC	Brushless direct current
C	Compressor
COP	Coefficient of performance
DBT	Dry bulb temperature
DH`	Dehumidifier
DHW	Domestic hot water
DOE	U.S. Department of Energy
EER	Energy efficiency ratio
EES	Engineering Equation Solver
EF	Energy factor
EF_d	Dehumidifier energy factor
EPRI	Electric Power Research Institute
EVI	Expansion valve, indoor
EVO	Expansion valve, outdoor
EWT	Entering water temperature
FI	Variable-speed fan, indoor
FO	Variable-speed fan, outdoor
GS-IHP	Ground-source heat pump
H	Humidification
HP	Heat pump
HPDM	DOE/ORNL Mark VI Heat Pump Design Model
HSPF	Heating season performance factor
HVAC	Heating, ventilating, and air conditioning (system)
HX	Heat exchanger

HXRAI	Refrigerant-to-air heat exchanger, indoor
HXRW	Refrigerant-to-water heat exchanger
HXRWO	Refrigerant-to-water heat exchanger, outdoor
IHP	Integrated heat pump
OCD	Outdoor coil defrosting
P	Pump
PI	Single-speed pump
PID	Proportional/integral/differential
PO	Multi-speed pump
RH	Relative humidity
R&D	Research and development
RV	Reversing valve
SEER	Seasonal energy efficiency ratio
SHR	Sensible heat ratio
UA	Heat loss coefficient
WBT	Web bulb temperature
WH	Water heater/heating
WT	Hot water tank
ZEH	Zero-energy home

1. EXECUTIVE SUMMARY

The energy service needs of a net-zero-energy house (ZEH) include space heating and cooling, water heating, ventilation, dehumidification, and humidification, depending on the requirements of the specific location. These requirements differ in significant ways from those of current housing. For instance, the most recent U.S. Department of Energy (DOE) buildings energy data (DOE/BED 2007) indicate that on average ~43% of residential buildings' primary energy use is for space heating and cooling, vs. ~12% for water heating (about a 3.6:1 ratio). In contrast, for the particular prototype ZEH structures used in the analyses in this report, that ratio ranges from about 0.3:1 to 1.6:1 depending on location. The high-performance envelope of a ZEH results in much lower space heating and cooling loads relative to current housing and also makes the house sufficiently air-tight to require mechanical ventilation for indoor air quality.

These envelope characteristics mean that the space conditioning load will be closer in size to the water heating load, which depends on occupant behavior and thus is not expected to drop by any significant amount because of an improved envelope. In some locations such as the Gulf Coast area, additional dehumidification will almost certainly be required during the shoulder and cooling seasons. In locales with heavy space heating needs, supplemental humidification may be needed because of health concerns or may be desired for improved occupant comfort. DOE has determined that achieving their ZEH goal will require energy service equipment that can meet these needs while using 50% less energy than current equipment.

One promising approach to meeting this requirement is through an integrated heat pump (IHP) – a single system based on heat pumping technology. The energy benefits of an IHP stem from the ability to utilize otherwise wasted energy; for example, heat rejected by the space cooling operation can be used for water heating. With the greater energy savings the cost of the more energy efficient components required for the IHP can be recovered more quickly than if they were applied to individual pieces of equipment to meet each individual energy service need. An IHP can be designed to use either outdoor air or geothermal resources (e.g., ground, ground water, surface water) as the environmental energy source/sink.

Based on a scoping study of a wide variety of possible approaches to meeting the energy service needs for a ZEH, DOE selected the IHP concept as the most promising and has supported the research directed toward development of both air- and ground-source versions. This report describes the air-source IHP (AS-IHP) design and includes the lessons learned and best practices revealed by the research and development (R&D) effort throughout.

Salient features of the AS-IHP include a variable-speed rotary compressor incorporating a brushless direct current permanent magnet motor which provides all refrigerant compression, variable-speed fans for both indoor and outdoor sections, and a multi-speed water pump. The laboratory prototype uses R-22 because of the availability of the needed components that use this refrigerant. It is expected that the HFC R-410A will be used for any products arising from the IHP concept. Laboratory test data was used to validate component models incorporated into the DOE/ORNL Mark VI Heat Pump Design Model (HPDM). HPDM was then linked to TRNSYS, a time-series-dependent simulation model capable of determining the energy use of building cooling and heating equipment as applied to a defined house on a sub-hourly basis. This provided a highly flexible design analysis capability for advanced heat pump equipment; however, the program also took a relatively long time to run. This approach was used with the initial prototype design reported in Murphy et al. (2007a) and in the business case analysis of Baxter (2007). A revised approach was developed for this study based on using HPDM to generate an IHP performance map that is interrogated by TRNSYS using multi-parameter interpolation. A single multi-parameter performance map can be used to analyze IHP performance in different climates, houses, and with various control strategies, though it must be regenerated if significant changes are made to the IHP design. The map-based simulation allows faster run times while maintaining essentially the same accuracy.

The revised simulation approach was used to calculate the yearly performance of an AS-IHP design 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. The results showed 48.4% to 67.2% energy savings for four of the locations. In Chicago the energy savings were somewhat lower — 45.6% — due to more reliance on electric resistance backup energy for space and water heating during periods of extremely low ambient temperatures.

This report includes specifications for a field test prototype and a recommended system control strategy based on the R&D done by ORNL to date. It should be noted that all R&D conducted thus far for the AS-IHP has been aimed at the ZEH. However, modifications to these recommendations will be needed to produce a product optimized to achieve penetration in the current housing market, in which houses differ from the ultimate ZEH goal. These modifications may include eliminating some functions and substituting components to produce a simpler, less-expensive product for initial market penetration. Work in the future with a manufacturing partner (or partners) toward this latter goal is planned.

This report documents the development of an AS-IHP through the third quarter of FY2007, and along with two other reports (*Integrated Heat Pump HVAC Systems for Near-Zero-Energy Homes – Business Case Assessment*, ORNL/TM-2007/064, Baxter; and *Non-Energy Attributes and the Potential Market Penetration of Energy-Efficient Technologies: The Case of Air Source Integrated Heat Pumps and New Residential Markets*, ORNL/TM-2007/069, Bjornstad et al.) forms the basis for evaluating the AS-IHP against DOE's Technology Development Stage-Gate management criteria for Gate 4, for transition from Stage 3, Advanced Development, to Stage 4, Engineering Development. This report describes the design, analyses, and testing of the AS-IHP and provides performance specifications for a field test prototype and proposed control strategy. The results obtained so far continue to support the AS-IHP being a promising candidate to meet the energy service needs for a ZEH in support of DOE's goal of ZEH-ready residential building designs by the year 2020.

2. INTRODUCTION TO THE INTEGRATED HEAT PUMP

The pursuit of net-zero-energy residences brings new requirements for meeting space cooling, space heating, water heating, ventilation, and humidity loads. First, the tighter, less conductive house envelopes characteristic of ZEH designs result in reduced space cooling and heating demands and, therefore, smaller equipment capacities than are customary in today's homes. Second, as houses become tighter, there is less natural air infiltration, and mechanical ventilation is generally necessary to meet accepted air quality standards for residences. Moreover, bringing moist ventilation air to space neutral conditions increases the need for latent cooling. And third, although the space conditioning loads are smaller with a tighter building envelope, the water heating load, which depends largely on the number of occupants in the dwelling and their life styles, remains essentially unchanged. Consequently, the water heating load will become a larger portion of the overall energy service demands to be met by building equipment in the house.

Mechanical ventilation combined with reduced space conditioning loads and an unchanged hot water demand suggest that an integrated load-following system would be an effective way to meet the energy service needs of a net-ZEH. Such a system based on the demonstrated high efficiency of vapor compression technology, and denoted as the “integrated heat pump” (IHP) here, would provide in a single appliance for the ZEH space conditioning, ventilation, dehumidification, and water heating requirements.

Systems with the ability to follow load and control supply air sensible heat ratio (SHR) typically employ variable-speed components. Such capabilities also suggest long (near continuous) equipment runtimes, reflecting duty cycles that are well suited to conditioning a supply flow of ventilation air (typically small relative to the air circulation rates of conventional non-variable-speed heat pumps) and to the efficient production of domestic hot water using heat pumping. Load following can also reduce on/off cycling and provide more consistent space temperature and humidity control, all leading to improved occupant comfort.

Variable-speed technologies are growing in use and in efficiency. Newsletters (such as International Institute of Refrigeration 2005) indicate that many Japanese heating, ventilation, and air conditioning (HVAC) equipment manufacturers have shifted their attention to variable-refrigerant flow systems based on variable-speed technologies (e.g., brushless direct current motors, etc.) with attendant inverter drive systems. Although still more costly than conventional induction motors typically used in single-speed HVAC systems, variable-speed drives and motors have continued to drop in cost, as have most electronic components for equipment, especially for the production volumes in Asian markets. The remaining cost premium associated with variable-speed technology can be significantly offset through HVAC designs and control strategies that apply variable-speed technologies to perform the additional functions of dehumidification and water heating.

2.1 Prior Experience

At least three prior efforts have been made in the United States (Thorne 1998) to develop and successfully commercialize an air-source heat pump system with both space conditioning and water heating capability: the HydroTech 2000, the Powermiser, and the AquaPlus.

2.1.1 Carrier/EPRI HydroTech 2000

The result of a cooperative effort between the Electric Power Research Institute (EPRI) and the Carrier Corporation, the HydroTech 2000 (38QE/40QE) was a residential system with five primary modes of operation: space cooling, space cooling plus water heating, space heating, space heating with water heating, and water heating only (Dunshee 1995). A novel defrost auxiliary mode used hot water from the storage tank as the heat source to evaporate refrigerant entering the compressor on its way to heat the outdoor coil, thereby removing ice buildup. Other auxiliary modes available for user selection included emergency heat, cooling plus humidity control, and heating plus humidity control. Separate ventilation and dehumidification-only modes were not incorporated into this system.

Based on development work started in 1982, four early prototypes were fabricated for testing in the laboratory and in a Carrier employee’s home in 1985. In 1987-1988 ten improved prototypes were installed as the initial field trial in homes across the continental United States. Commercial production and sales of the systems began in 1989, with two versions offered: 2- and 3-ton nominal cooling capacities. These had Air-Conditioning and Refrigeration Institute (ARI) certified cooling capacities of 24,000 and 36,800 Btu/hr, heating capacities of 25,800 and 35,400 Btu/hr and ratings of 13.35 and 14.05 seasonal energy efficiency ratio (SEER) and 8.75 and 9.05 heating season performance factor (HSPF), respectively. For the field demonstration phase, 31 sites in 16 states were selected in cooperation with participating utilities. Twenty-three of the 3-ton and eight of the 2-ton instrumented production systems were installed at these locations, of which 27 (21 of the 3-ton systems and 6 of the 2-ton systems) produced useful first-year data and 14 (11 of the 3-ton systems and 3 of the 2-ton systems) produced useful second-year data. A separate field monitoring exercise was conducted at a single house in Maryland by the National Institute of Standards and Technology during the same period (Fanney 1993).

Relative to this unit, the EPRI perspective was: "The HydroTech 2000 represents the first fully integrated, variable-speed heat pump space-conditioning and water-heating system. Its field performance was excellent, and all field test participants noted its comfort and energy efficiency. However, the high first-cost of variable-speed equipment resulted in an expensive unit, which, in turn, resulted in low sales volume and finally, removal as a commercial product. Successful future marketing of similar systems will require adequate consumer understanding of the benefits and costs" (Dunshee 1995). Production and distribution of the HydroTech 2000 was terminated in 1992. Total sales over the three-year period were estimated to be a few hundred units.

This system employed a "triple-split" configuration, chosen to facilitate locating water-containing components indoors to avoid freezing situations. The configuration consisted of three separate parts provided by Carrier: a compressor section, an indoor fan-coil section, and an outdoor fan-coil section. The compressor section was connected, in the standard configuration, to a conventional electric water heater (with resistance elements retained for back up or emergency water-heating) to complete the arrangement required to provide the integrated functions of space conditioning and water heating. The refrigerant employed was R-22 (9.6 and 12.0 lb_m standard charge, respectively).

The compressor section was located indoors and contained the compressor with accumulator, drive, refrigerant control valves, refrigerant-to-water heat exchanger (HX), water pump, control box, and two temperature sensors. The compressor was a two-cylinder, reciprocating type driven by a variable-speed (1800 to 5400 rpm), electronically commutated motor with a permanent magnet rotor and three-phase stator. The refrigerant expansion/metering function was accomplished with a single bi-directional pulsing solenoid valve (with pulse-width modulation to follow system load), which also served to block flow through the refrigerant-to-air HX during defrost cycles and to prevent refrigerant migration during off cycles. Conventional reversing and defrost valves were employed. The copper tube-in-tube refrigerant-to-water HX surrounded the compressor and was sized to handle the full heat rejection of the heat pump. It consisted of four double-walled and vented inner tubes carrying refrigerant and a surrounding water-carrying annular space enclosed by an outer tube. Two of the inner tubes served as a desuperheater/condenser for the water heating modes, and the other two served as an evaporator for the water-source defrost function. The stainless steel water pump was a single-speed centrifugal type sized for 3 gpm. The control box in this section contained a standard outdoor module plus a relay module and related power components. The standard outdoor module served as the microprocessor-based master control for the system, determining the various operating modes, conducting diagnostic functions, and maintaining bus communications with the indoor fan-coil section. The temperature sensors in this section were 10 kΩ thermistors located on the suction side of the accumulator (for freeze protection during water-source defrost) and on the discharge side of the compressor (for high-temperature protection).

The indoor fan-coil section contained a refrigerant-to-air HX, a fan with drive, a control box, and two temperature sensors. The refrigerant-to-air HX (3.16 and 5.00 ft^2 face area, respectively) was constructed using internally enhanced copper tubes and augmented aluminum external fins. Special circuiting was employed to optimize performance and assure proper oil return to the compressor over the full range of variable-load operation. The fan was a direct-drive centrifugal type with a variable-speed (250 to 1500 rpm) integral control motor similar in type to that associated with the compressor. The control box in this section contained a standard indoor module plus power components. The temperature sensors in this section were 10 kΩ thermistors located on the indoor refrigerant liquid tube (for coil freeze protection) and in the return air (for emergency heat over-temperature protection).

The outdoor fan-coil section contained a refrigerant-to-air HX, a fan, and some control elements. The refrigerant-to-air HX (15.0 and 20.5 ft^2 face area, respectively) was constructed using internally enhanced copper tubes and augmented aluminum external fins. As with the indoor HX, it was specially circuited to optimize performance and assure proper compressor oil return over the full range of variable-load operation. The fan was a direct-drive type using a multi-bladed propeller with a single-speed induction motor. The temperature sensors in this section were 10 kΩ thermistors located in the outdoor air (for mode, compressor, and electronic expansion valve control) and on the outdoor refrigerant liquid tube (for defrost control).

The system was controlled by three separate microprocessor-based modules associated with the compressor section, the indoor section, and the thermostat. As described above, six temperature sensors were located in the various sections. One additional temperature sensor (a 10 kΩ thermistor) was located in the bottom fitting of the water tank (for water heating modes control). Together, these seven temperature sensors provided inputs necessary for the control system to determine, at any given time, which of the components (compressor, indoor fan, outdoor fan, reversing valve, expansion valve, water pump, and resistance heating elements) should be operating and at what rate the compressor, indoor fan, expansion valve should be operating.

2.1.2 Nordyne/EPRI Powermiser

EPRI also co-sponsored a more recent effort with Nordyne, Inc., to develop a lower-cost unit with combined space-conditioning and water-heating capabilities. The Powermiser (Nordyne), introduced in 1992, was marketed under the Miller brand name in 2-, 3-, and 4-ton nominal capacities. SEER 10 (HSPF 7.0) versions of these systems had rated cooling capacities of 22,600, 34,600, and 45,000 Btu/hr and heating capacities of 21,000, 34,600, and 45,000 Btu/hr, respectively. SEER 12 (HSPF 7.6) versions employed next-nominal-size indoor coils to boost both efficiency and capacity, having rated cooling capacities of 24,000; 36,000; and 48,000 Btu/hr as well as heating capacities of 24,000; 36,000; and 48,000 Btu/hr, respectively. Production was halted after several years on the market and estimated total sales were a few thousand units.

The Powermiser had many design similarities to the HydroTech: each was a “triple- split” system with three sections, each employed R-22 as the refrigerant (standard charge 9.9, 13.0, and 16.8 lb_m, respectively, for the Powermiser), and each had a similar list of available operating modes. However, major differences in the Powermiser included the use of only single-speed compressor and fan components, the use of electromechanical controls, the use of two fixed orifice (0.071-, 0.082-, and 0.093-in. indoor, respectively, and 0.059-, 0.063-, and 0.061-in.

outdoor, respectively) expansion devices with sliding check valve functions, the use of air-source defrosting, and the implementation of a charge management system.

2.1.3 Lennox AquaPlus

A still more recent product was the Lennox AquaPlus (introduced in mid-1997). Based on concepts initially reported (Gilles 1994) and patented (United States Patent, 1994) by Lennox, this unit was essentially a heat pump water-heating unit that was added to a conventional heat pump system. The main module (Lennox 1997) was based on R-22 as the refrigerant (standard charge 2.4 lb_m) and had a list of components similar in nature to, but different in flexibility from, the HydroTech 2000's compressor section. These encompassed a compressor (single-speed rotary type for the AquaPlus) with accumulator, refrigerant-to-water HX, water pump (single-speed, 3.6 gpm), and controls. However, in the AquaPlus case, this module was solely controlled by demand for hot water. An entirely separate conventional compressor, condenser, expansion valve, and evaporator heat pump system responded to calls for space cooling or heating.

The AquaPlus employed its own refrigerant evaporator coil (with expansion valve) in the return duct to the conventional indoor fan coil to remove heat from the air stream before it encountered the conventional indoor coil. The indoor fan was controlled so as to operate at a low speed when there was only water-heating demand and a high speed when there was any space-conditioning demand. As in the HydroTech 2000 and Powermiser cases, water was pumped from a conventional electrical resistance water-heating tank through the AquaPlus refrigerant-to-water HX and returned to the tank. The water flowed through spiral double-walled copper inner tubes of the helical coaxial tube-in-tube exchanger, receiving heat from the refrigerant condensing between the inner surface of the surrounding steel outer tube and the exterior of the inner tubes.

When there was demand for hot water, the AquaPlus operated, producing hot water efficiently while removing heat from the return air stream, and thereby reducing by approximately 1 ton the cooling load to be accommodated by the conventional space conditioning heat pump.

2.2 Current Approach

The current IHP approach (Tomlinson et al. 2005, Murphy et al. 2007a) builds on earlier experience garnered from the product development efforts outlined above. At the same time, it recognizes important changes in the residential housing environment that may affect system appeal. As described above, if the marketplace moves toward ZEH-type residences, smaller, more efficient space-conditioning and water-heating systems that can accommodate not only customary loads, but also new active ventilation and dehumidification requirements, will be needed. The relatively large number of current two-story houses with multiple smaller heat pumps might provide a nearer-term market that could induce manufacturers to produce such "futuristic" equipment, especially for "early adopters." If these trends intersect with international component cost reduction trends observed in variable-speed, high-efficiency equipment, and with the increasing cost, capacity, and emissions pressures associated with the world energy production markets, the residential AS-IHP, illustrated conceptually in Fig. 2.1, may fill a substantial and valuable niche in the energy-efficiency arsenal. However, the costs and benefits of the AS-IHP system will be weighed in the marketplace against competing suites of individual components that can meet the same imposed loads.

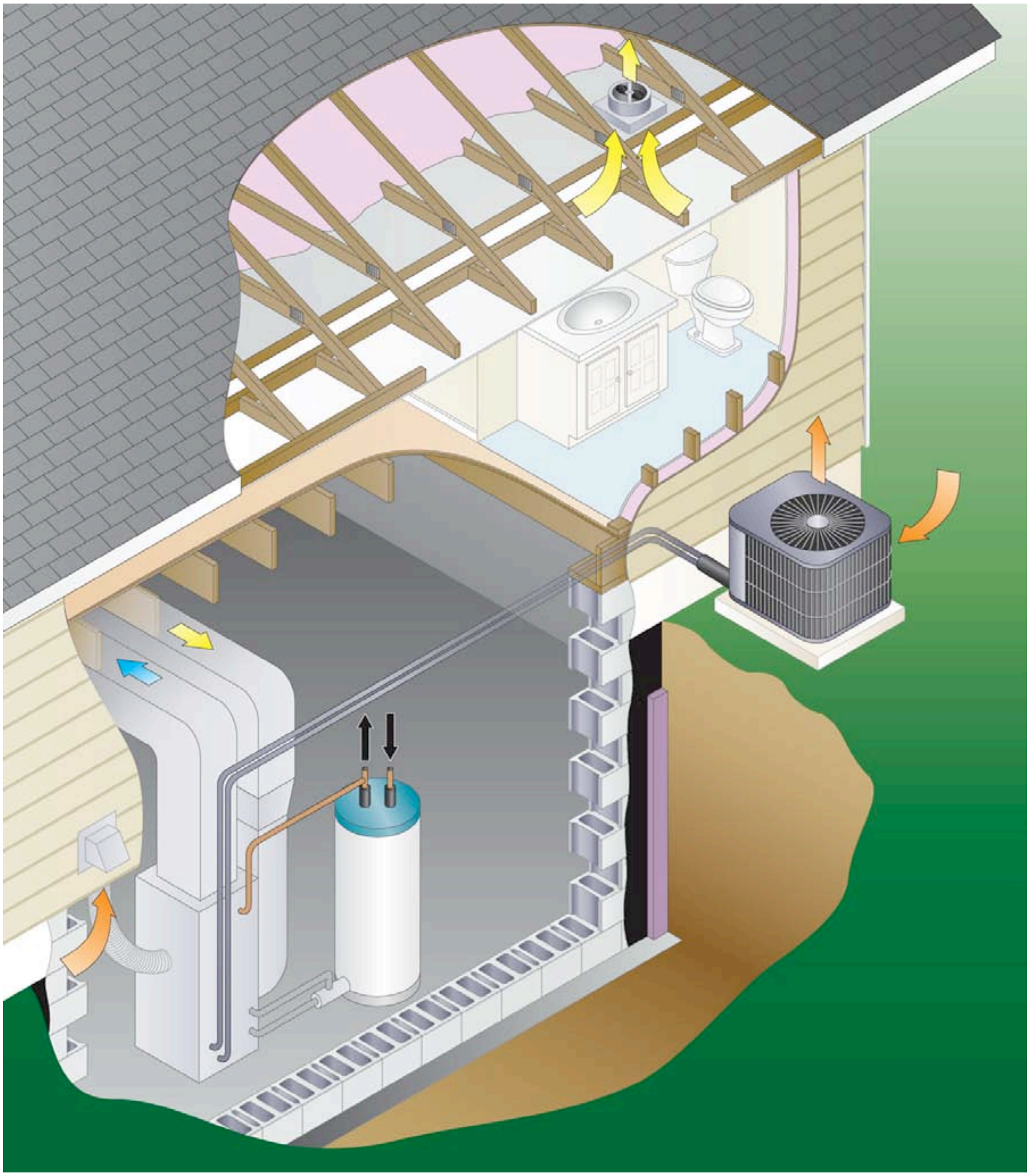

Fig. 2.1. Conceptual installation of the residential air-source integrated heat pump.

3. LABORATORY PROTOTYPE

The laboratory prototype system is based on prior analyses of various IHP concepts. As indicated schematically in Fig. 3.1, the current concept incorporates two separate but interactive loops, one refrigerant and one water, that employ several modulating electrically driven components, including one variable-speed compressor (C), one multi-speed pump (P), and two variable-speed fans [one indoor (FI) and one outdoor (FO)]. The remaining major components include a reversing valve (RV), two expansion valves [one indoor (EVI) and one outdoor (EVO)], and four HXs to meet the space conditioning and water heating loads: two refrigerant-to-air [one indoor (HXRAI) and one outdoor (HXRAO)], one refrigerant-to-water (HXRW), and one water-to-air (HXWA). The water-to-air HX uses excess hot water generated in the cooling and dehumidification modes and stored in the hot water tank (WT) to temper the ventilation air, as needed, to meet space neutral temperature requirements. Modulation of compressor speed, indoor fan speed, and water pump 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. Selection of appropriate specific components and their incorporation in a logical fashion provide flexibility in the laboratory prototype system that allows operation over a wide range of modes and parameter ranges.

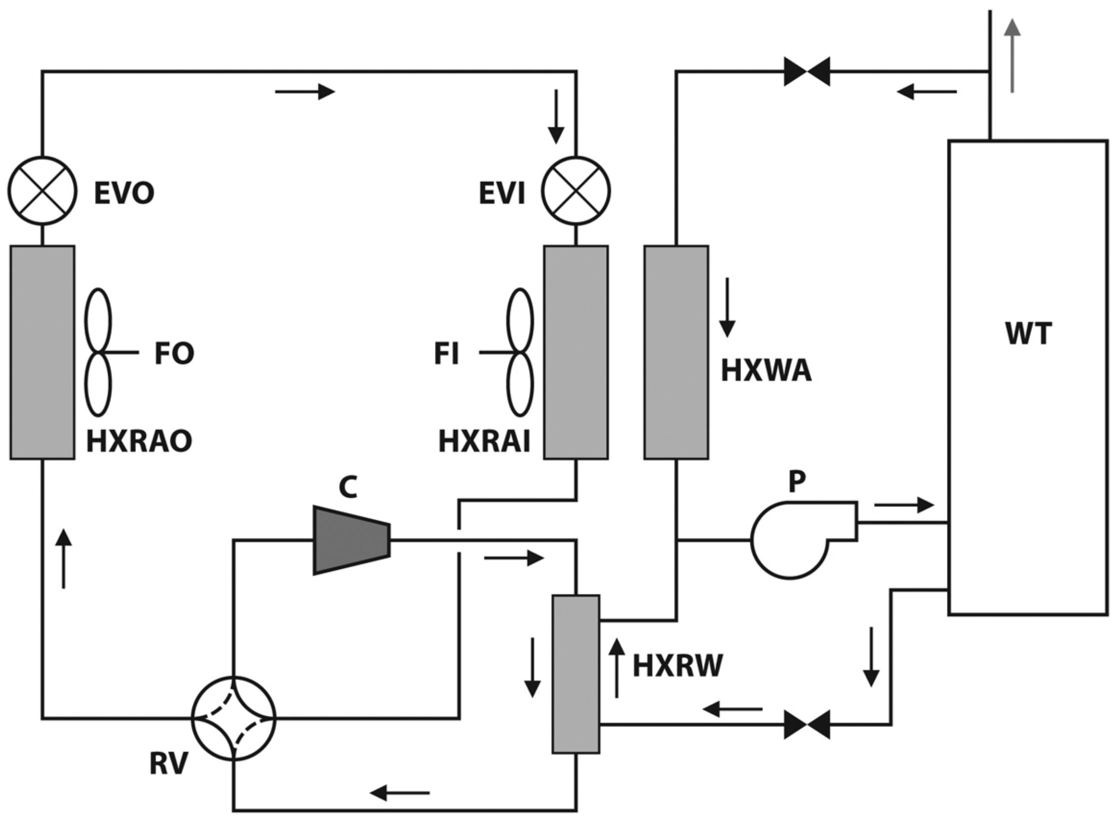

Fig. 3.1. Schematic of integrated heat pump concept.

3.1 Refrigerant Compressor

Several refrigerant compressor options were considered to provide in a single IHP system the capability to meet projected ZEH energy service loads with high efficiency. Preliminary assessments indicated that a compressor speed ratio of 5-to-1 (from 100 to 20% capacity) in cooling mode would provide the capacity range from the minimum ventilation air dehumidification modes to space cooling under design day conditions. Outside of ventilation mode cooling, a modulation range from 45 to 100% for space cooling was needed. Design space conditioning loads computed by NREL for the prototype ZEHs used in this study indicated cooling capacity requirements of 1 – 1.5 tons for the locations of interest — Atlanta, Chicago, Houston, Phoenix, and San Francisco (Christensen 2005).

The same combination of measures was estimated to provide greater design load reductions in heating than in cooling. By using over-speed compressor operation in the heating mode (Rice 1992), the heating balance point can be further reduced to minimize the need for supplemental resistance heat. In space heating, a speed modulation range of 35 to 150% (or higher) of maximum rated operation is desired. Between 35 and 100% of rated speed, a constant voltage-to-frequency ratio is maintained for constant torque capability. Over-speed operation (above 100% of rated speed) with reduced voltage-to-frequency control is possible in space heating because the compressor torque loading is reduced under colder ambient conditions (Rice 1988).

Table 3.1 summarizes the range of compressor options that were considered, the pros and cons of each, and the ratings of each on the basis of estimated compressor efficiency, control simplicity, cost, and availability in the needed capacity. Using these evaluations, we decided on the first option, a single modulating rotary compressor incorporating a brushless direct current permanent magnet (commonly referred to as a brushless direct current, or BDC, or brushless permanent magnet) motor. Importantly, the manufacturer also provided a controller that allows us to operate at fixed speeds, which is essential for conducting steady-state tests. All manufacturers of speed-modulating equipment must provide such controls to have such units tested in the United States in compliance with the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE)/Air-Conditioning and Refrigeration Institute (ARI) rating procedures (Horak 2005).

Of all currently produced hermetic compressors, the ones incorporating BDC drives have the potential to operate most efficiently over a wide modulation range. Consistent relative operating efficiency in all modes is important to achieving the total energy savings goals relative to baseline-13 SEER equipment and resistance water heating. In addition, because such compressors with BPM drives have been mass produced overseas for at least 12 years in the needed capacity range, their incremental costs have been reduced relative to compressor alternatives with single-speed or modulating inverter-driven induction motors.

The standard version of the compressor selected for the IHP laboratory prototype is designed to modulate from 30 to 100 Hz, with a nominal capacity rating of 9500 Btu/hr at 58 Hz (3480 rpm). Over-speed capability extends the range to 100 Hz, where the available capacity at rated conditions reaches approximately 15,000 Btu/hr. It has a compressor-only energy efficiency ratio (EER) rating of 11.5 at 58 Hz with added inverter losses of about 5 to 8% expected.

Table 3.1: Compressor modulation option assessment (5 = best; 1 = worst)

Modulation Options	Modulation Levels	Pros	Cons	Efficiency (1 – 5)	Simplicity (1 – 5)	Low Cost (1 – 5)	Availability in needed size
1) Brushless permanent magnet rotary compressor (as in newer high-efficiency Asian multi-splits)	Continuous, 4- or 5-to-1 speed ratio	Single compressor, high drive efficiency, wide modulation range, simplest control logic, in mass production	More expensive, needs inverter and permanent magnet motor, requires special controller for fixed frequency testing, limited sizes	5	5	1 – 2	5
2) One single-speed rotary compressor and one equal size brushless permanent magnet rotary compressor in parallel (also in newer Asian multi-splits)	Near continuous	Less expensive electronics and motor than Option 1, wide modulation range, high drive efficiency	Two compressors, no availability of unit with controller for prototype test, dual control more complex	5	4	2 – 3	1
3) Inverter-driven induction motor rotary compressor (as in older or low-end Asian multi-splits)	Continuous, 3- or 4-to-1 speed ratio	Single compressor, moderate drive efficiency, poorer at lower speeds, lower modulation range, simplest control logic, in mass production	Relatively expensive, requires special controller for fixed-frequency testing, limited availability	3 – 4	5	2	2
4) Dual single-speed rotary compressors of unequal sizes in parallel	Three	Less expensive option, good low-capacity efficiency	Limited ventilation humidity control due to discrete modulation	3 – 4	3	3	4
5) Dual stroke reciprocating compressor	Two	Single compressor, least expensive option, reciprocating compressor better at water heating	Limited ventilation humidity control, most discrete modulation, lower modulated efficiency than Option 4, more capacity drop-off in heating mode	2	4	4	4
6) Dual stroke reciprocating compressor in parallel with smaller single-speed rotary compressor	Five	Widest modulation range, good low-capacity efficiency	Limited ventilation humidity control, most complex control logic, more capacity drop-off in heating mode	3	2	2	4
7) Option 4 or 5 with add-on inverter control of the smaller-capacity unit	Continuous for lower capacities, discrete for higher capacity	Acceptable low-capacity efficiency with modulation	Requires 3-phase motors for best efficiency and low-speed control	2 – 3	3	2	3

Primarily constrained by lubrication concerns at low speeds, the 30-Hz lower limit on the standard version does not allow the unit to operate at the lowest capacity needed for the continuous ventilation-air-cooling mode. To accommodate the potential for lower-speed operation, our supplier provided a version modified to improve low-speed oil supply, permitting operating down to 15 Hz. However, the controller currently available from the supplier for this compressor line does not incorporate the vector control characteristics required to reach frequencies below 30 Hz. An alternate ventilation strategy allows us to replace continuous operation below 30 Hz by intermittent operation at or above 30 Hz.

3.2 Refrigerant-to-Water Heat Exchanger

The refrigerant-to-water HX (shown as HXRW in Fig. 3.1) transfers heat from refrigerant heated in the compression process to potable water circulated from the hot water storage tank. This device is intended to take the full refrigerant condensing load of the system when the condenser fan is off, and to act as a desuperheater when the condenser fan is on. To meet these demands, we employ a counter-flow arrangement of a tube-in-tube helical HX with a 1 1/8-in. nominal outside diameter and a 137-in. nominal coiled length [Packless Industries CDAX-6100-H-9-137 rated at 15,000 Btu/hr heat rejection with 162 lb_m/hr R-22 flow (5°F subcooling) and water flow of 3.0 gal/min (3.0 psi water-side pressure drop) with a temperature difference of 20°F between the condensing temperature and the entering water temperature]. This HX, with an installed overall length, width, and height of 14 7/8 in., 10 1/8 in. and 7 1/8 in., respectively, consists of an inner vented double-wall convoluted copper tube surrounded by an outer smooth steel tube (see Fig. 3.2). Potable water flows inside the inner wall of the double-wall tube while refrigerant passes through the annulus formed between the outside of the outer wall of the double-wall copper tube and the inside of the steel tube. Water and refrigerant connections are 5/8 in. and 3/8 in., respectively. Counter-flow is employed to maximize heat transfer from the refrigerant to the water by providing the greatest effective temperature difference between them. The double wall isolates the two fluids to prevent contamination of the water in case of a refrigerant leak. Double-wall protection is a requirement of the International Association of Plumbing and Mechanical Officials for refrigerant-based water heating systems.

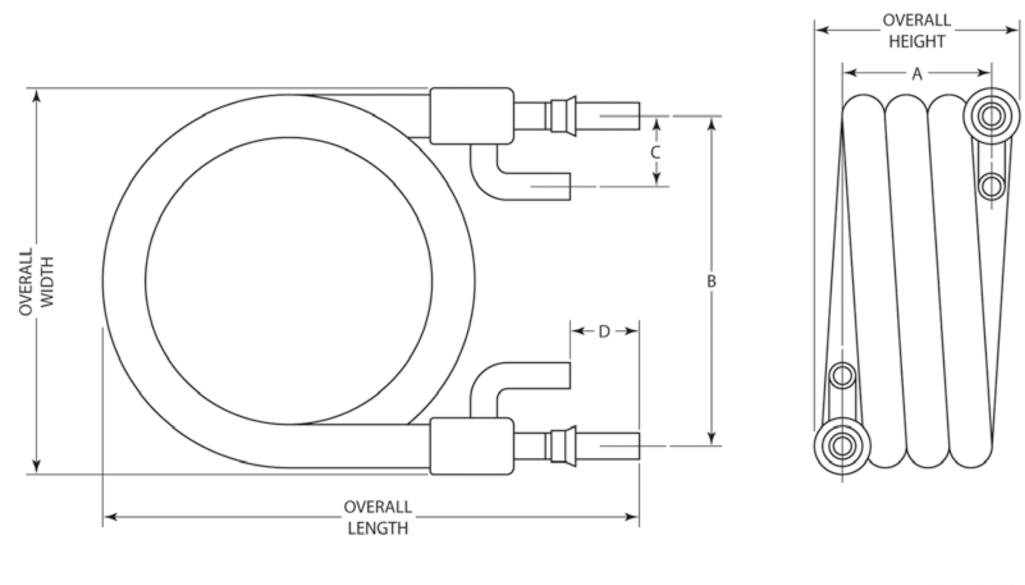

Fig. 3.2. Refrigerant-to-water heat exchanger.

3.3 Outdoor Section

The requirement for an outdoor unit incorporating a reversing valve and a coupled fan and refrigerant-to-air HX is met with a modified Carrier Model 38YCC018340 outdoor section (Fig. 3.3) of a split-system heat pump. When deployed with its standard supplied compressor (reciprocating) and expansion device (piston) and connected to an approved indoor fan coil, this unit has a nominal capacity of 18,000 Btu/hr cooling load and SEER of 10. However, as employed in the smaller capacity IHP laboratory prototype system, the refrigerant-to-air HX in the 38Y essentially becomes oversized so that the IHP system efficiency achievable well exceeds the 10 SEER standard rating. The wrap-around coil (HXRAO) has a face area of 9.11 ft^2 and uses copper tubes with enhanced aluminum fins (20 per in.). The refrigerant flow configuration involves 2 circuits and 1 row. Nominal airflow induced by the propeller-type, vertical direct-drive multi-bladed propeller fan (FO) in this unit is 1700 cfm at 1100 rpm with 100 W power draw at 60 Hz for the standard 1/12-hp induction motor. Because IHP system operation covers airflows ranging from approximately 30 to 70% of this nominal, an ICM Controls Corporation Comfort Control Center CC750-230 variable-frequency (25-50 Hz) /variable-voltage fan motor speed control (Fig. 3.4) is used as an expedient means to vary the outdoor fan flow rate over this span. The modulating compressor described earlier in this report is installed in place of the original compressor inside the outdoor enclosure. A Swagelok SS-4MG (1/4-in. end connections, 0.03 flow coefficient, 0.056-in. orifice) metering valve (EVO) provides the adjustment needed to meet the expansion/flow metering requirements of the laboratory prototype when the outdoor refrigerant-to-air HX is operating as an evaporator.

Fig. 3.3. Outdoor section.

Fig. 3.4. Fan motor speed controls

3.4 Indoor Fan Coil

A Carrier FK4DNF001 (1/2 hp) fan coil (Fig. 3.5) is used to meet the requirement for an indoor unit incorporating a coupled fan and refrigerant-to-air HX. This is a slope-type coil (HXRAI) using grooved copper tubing with lanced sine wave aluminum fins (14 fins per in., face area of 2.97 ft^2). The refrigerant flow arrangement involves five circuits and three rows. The direct-drive fan (FI) in this unit (Fig. 3.6) has a high-efficiency variable-speed motor — an integral electronically commutated version manufactured by General Electric Company. As supplied in its standard factory configuration, the variable-speed system senses torque and adjusts the fan speed to provide a set flow (nominal 525 cfm), irrespective of static pressure. To allow direct control of fan speed for IHP conditions, we bypass the standard factory mode by using a replacement control head (provided by General Electric Company for their ECM™ 2.3 motor to set it for "V_{spd}" operation and set status flag #7 to "RPM"). An Evolution Controls, Inc., EVO/ECM-VCU-36-mp visual control unit (Fig. 3.4) is employed to provide "GO" and "manually adjustable V_{spd}" signals to the head. The standard supplied expansion device (thermostatic expansion valve) is supplanted by a Swagelok SS-31RS4 (1/4-in. end connections, 0.04 flow coefficient, 0.062-in. orifice) metering valve (upper in Fig. 3.7 and EVI in Fig. 3.1), which provides the adjustment needed to meet the expansion/flow metering requirements of the IHP when the indoor refrigerant-to-air HX is operating as an evaporator.

Fig. 3.5. Indoor fan coil.

Fig. 3.6. Indoor fan with integral variable-speed electronically commutated motor.

Fig. 3.7. Indoor refrigerant expansion valve and check valve arrangement.

3.5 Water-to-Air Heat Exchanger

As shown in Fig. 3.8, the water-to-air heat exchanger, a Heatcraft 2WZ0801L-20.00x14.00, uses 5/16-in.-diameter smooth copper tubing with flat 0.0045-in.-thick aluminum fins (8 fins per in.). It acts essentially as an air reheat coil and, as such, is located within the indoor section enclosure, perpendicular to the air flow and downstream of its refrigerant-to-air HX (Fig. 3.9). The water flow arrangement is 2-circuit with 10 passes. The HX dimensions are approximately 20-in. fin height x 14-in. fin length. Based on entering conditions of 100°F and 1.0 gpm (water) and 55°F and 90 cfm (air), the coil was designed to transfer approximately 2300 Btu/hr of heat from the water stream to the air stream, raising its temperature by 24°F. At these conditions, the water-side and air-side pressure drops incurred in the coil are 3.5-ft water and less than 0.01-in. water, respectively. Water-side connections are 1/2 in.

3.6 Water Pump

To provide required flows within the IHP water system, a three-speed circulator pump, Grundfos UPS 15-42F 230 VAC rated for 15 ft head at 3 gpm for the highest speed, with a two-pole asynchronous squirrel-cage permanent split capacitor motor (1/25-hp) is employed. The multi-speed characteristic gives the laboratory prototype additional flexibility in meeting the recovery load of the water heater. A low pump speed allows the tank to recover slowly when air tempering is being done by the water-to-air HX. If air tempering is not required, a higher pump speed can be used to improve tank heat recovery while reducing the condensing temperature of the heat pump system — a particular advantage at the higher water tank temperatures. The resulting reduction in condensing temperature and pressure increases the system efficiency under these conditions.

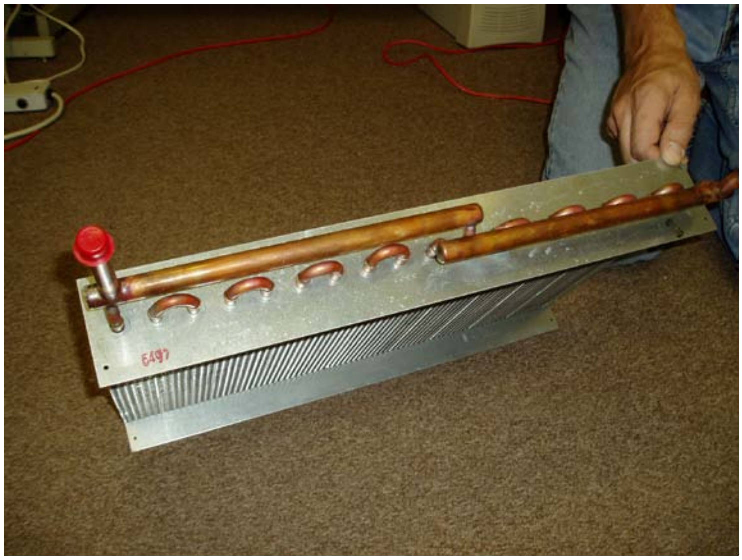

Fig. 3.8. Water-to-air heat exchanger.

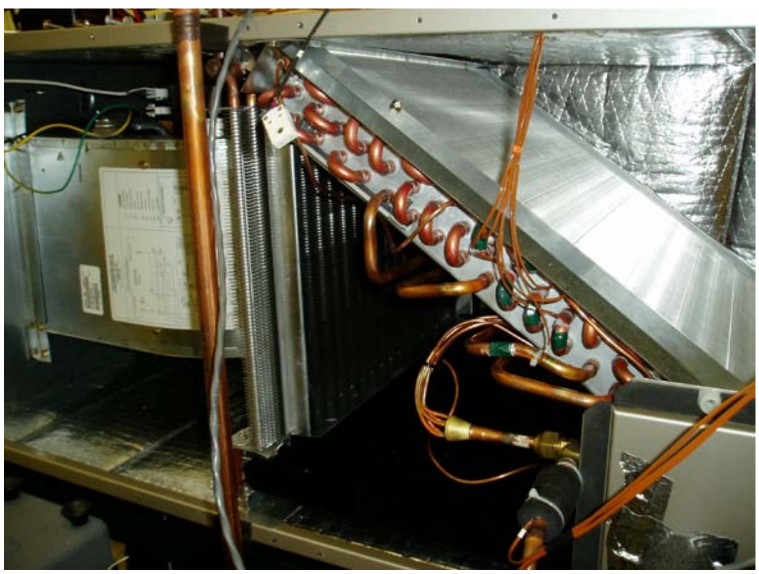

Fig. 3.9. Positions of indoor refrigerant-to-air, water-to-air heat exchanger, and fan.

3.7 Experimental Configuration

The laboratory prototype system is installed inside two environmental chambers that share a common wall to facilitate testing of split heat pump systems. Each chamber has 10 x 10 ft in floor area and constitutes an independently controlled psychrometric room, including an air circulating fan, as well as cooling (water-cooled compressor-driven), heating (electric resistance), humidification (steam injection), and dehumidification (regenerated desiccant wheel) systems. Temperature and humidity conditions are separately achieved and maintained in each chamber using two proportional integral differential controllers (Honeywell UDC 3000 Versa-Pro

DC300E-E-0A0-20-0000-0). With maximum cooling and heating capacities of 7.5 tons and 30 kW, respectively, the chambers provide the capabilities required to test residential-sized heat pump systems over a wide range of outdoor and indoor conditions. For water-heating tests, an auxiliary system near the chambers provides a source of temperature-controlled water. Figure 3.10 shows a diagram of the laboratory prototype IHP components and their locations inside the environmental chambers.

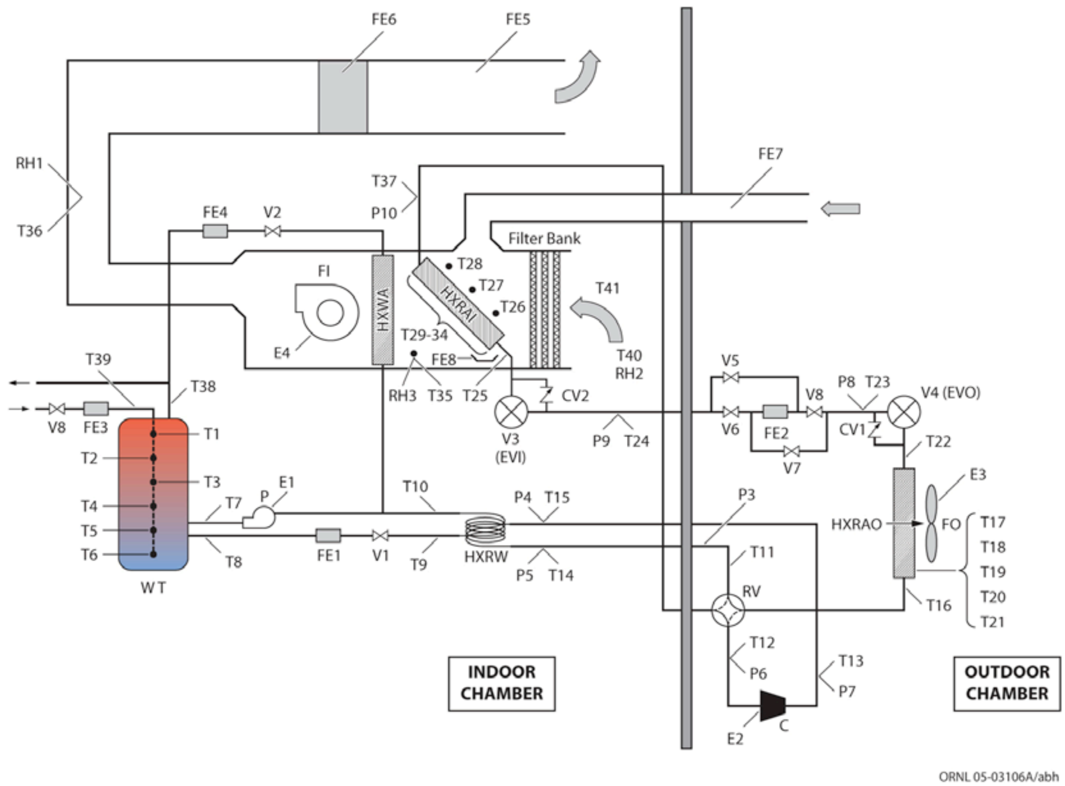

Fig. 3.10. Component and instrumentation location diagram.

The outdoor section is elevated above the floor of its chamber as shown in Fig. 3.3 to facilitate installation of refrigerant piping and instrumentation and to allow capture of condensate when the laboratory prototype is operated in the space heating mode.

The refrigerant-to-water HX is mounted below the indoor fan coil. Therefore, an additional line set is included to (1) conduct refrigerant from the compressor discharge through the wall dividing the chambers to this HX and (2) return refrigerant from this HX to the reversing valve mounted in the outdoor section.

A check valve (lower in Fig. 3.7) is installed in parallel with the indoor expansion valve to enable refrigerant expansion in the cooling mode and refrigerant bypass in the heating mode. A similar arrangement is employed for the expansion valve at the outdoor unit to enable refrigerant expansion in the heating mode and refrigerant bypass in the cooling mode. Other valves shown on the outdoor chamber side of Fig. 3.10 are refrigerant ball valves that maintain up flow through the refrigerant flow meter irrespective of the flow direction in the interconnecting liquid line, facilitating the use of a single calibrated refrigerant flow meter for all testing modes.

The return air plenum of the indoor fan coil case is modified to accommodate connection to a length of 4-in. round ducting between it and the outdoor chamber, enabling the introduction of ventilation air. A bank of removable filters is mounted on the return air plenum, upstream of the ventilation air duct transition to simulate the slight reduction in static pressure that would be present in an actual installation. Note that the indoor fan is downstream of this area. With the indoor unit placed in a horizontal position in the indoor chamber, the supply air ducting arrangement ensures the required minimum of two flow diameters upstream and downstream of the station where airflow rates are determined (Fig. 3.11). The station includes a honeycomb-type flow-straightening section upstream of the total and static pressure manifolds.

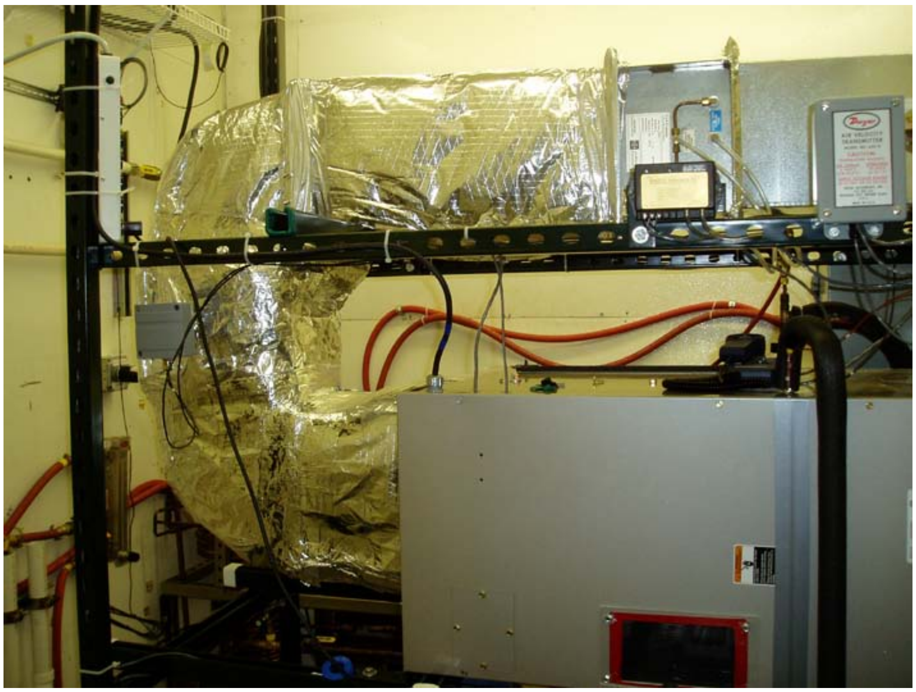

Fig. 3.11. Air supply duct and flow station arrangement.

Shown also in Fig. 3.10 is the hydronic loop comprising the hot water storage tank (American Water Heater Company Proline Model E62-50R-045 DV, 50-gallon nominal capacity), the refrigerant-to-water HX, the water pump, water flow meters, and manual valves to direct heated water to the reheat coil or to the bottom of the water tank, or both at the same time. The connection to the bottom of the tank is an annular fitting in which the hot water from the pump passes through the center of the fitting into the tank; water going to the refrigerant-to-water HX passes out of the tank through the annulus of the fitting. This arrangement ensures that the cooler water resident (by stratification tendencies) at the bottom of the tank is circulated to the inlet of the refrigerant-to-water HX, maximizing its effectiveness.

3.8 Instrumentation

The laboratory prototype system is instrumented to provide relevant air, water, refrigerant, and electrical measurements as shown in Fig. 3.10. Generally, low voltage transducer outputs are employed. However, where necessary, 4–20-mA current transducers are used to eliminate electrical noise that might affect the integrity of voltage outputs. Table 3.2 indicates the respective locations of measurements and the instruments employed to accomplish them. All purchased instruments are manufacturer-calibrated, except for the chilled mirror system for determining dew point, which is calibrated by ORNL.

Table 3.2: Laboratory prototype instrumentation

Instrument symbol	Measurement	Location	Instrument or transducer
T1	Temperature	Water in top sixth of WT	Thermocoouple, Type T
T2	Temperature	Water in 5/6 of WT	Thermocoouple, Type T
T3	Temperature	Water in 4/6 of WT	Thermocoouple, Type T
T4	Temperature	Water in 3/6 of WT	Thermocoouple, Type T
T5	Temperature	Water in 2/6 of WT	Thermocoouple, Type T
T6	Temperature	Water at bottom sixth of WT	Thermocoouple, Type T
T7	Temperature	Water (recirculating) entering WT bottom	Thermocoouple, Type T
T8	Temperature	Water (recirculating) leaving WT bottom	Thermocoouple, Type T
T9	Temperature	Water (recirculating) entering HXRW	Thermocoouple, Type T
T10	Temperature	Water (recirculating) leaving HXRW	Thermocoouple, Type T
T11	Temperature	Refrigerant from HXRW entering RV	Thermocoouple, Type T
T12	Temperature	Refrigerant at C suction	Thermocoouple, Type T
T13	Temperature	Refrigerant at C discharge	Thermocoouple, Type T
T14	Temperature	Refrigerant leaving HXRW	Thermocoouple, Type T
T15	Temperature	Refrigerant entering HXRW	Thermocoouple, Type T
T16	Temperature	Refrigerant entering HXRAO (cooling mode)	Thermocoouple, Type T
T17-T21	Temperature	Refrigerant in HXRAO at return bends	Thermocoouples, Type T
T22	Temperature	Refrigerant leaving HXRAO (cooling mode)	Thermocoouple, Type T
T23	Temperature	Refrigerant leaving outdoor check valve (cooling mode)	Thermocoouple, Type T
T24	Temperature	Refrigerant entering EVI (cooling mode)	Thermocoouple, Type T
T25	Temperature	Refrigerant entering HXRAI (cooling mode)	Thermocoouple, Type T
T26-T28	Temperature	Air entering HXRAI	Thermocoouples, Type T
T29-T34	Temperature	Refrigerant in HXRAI at return bends	Thermocoouples, Type T
T35	Temperature	Air leaving HXRAI	Resistance temperature detector, Vaisala Group Model HMD60Y (platinum 1000Ω IEC 751 Class B)
T36	Temperature	Air supply	Resistance temperature detector, Vaisala Group Model HMD60Y (platinum 1000Ω IEC 751 Class B)
T37	Temperature	Refrigerant leaving HXRAI (cooling mode)	Thermocoouple, Type T
T38	Temperature	Water leaving top of WT	Thermocoouple, Type T
T39	Temperature	Water entering WT from main	Thermocoouple, Type T

T40	Temperature	Air return	Resistance temperature detector, Vaisala Group Model HMD60Y (platinum 1000Ω IEC 751 Class B)
T41	Temperature	Air return dewpoint	Condensation optical chilled mirror hygrometer with monitor and sampling module, General Eastern Instruments, Inc. Model D2 two-stage chilled mirror sensor (four-wire platinum resistance thermometer, 100Ω 1/3 Class A DIN 43760) with General Eastern Instruments, Inc. Model M2 Plus-RH monitor and General Eastern Instruments, Inc. SSM vacuum pump sampling system (0.5-5.0 scfh)
RH1	Relative humidity	Air supply	Capacitance thin-film polymer, Vaisala Group Humicap Model HMD60Y
RH2	Relative humidity	Air return	Capacitance thin-film polymer, Vaisala Group Humicap Model HMD60Y
RH3	Relative humidity	Air leaving HXRAI	Capacitance thin-film polymer, Vaisala Group Humicap Model HMD60Y
P3	Pressure	Refrigerant entering RV	Capacitance diaphragm, Setra Systems, Inc. Model C207 (0-500 psig)
P4	Pressure	Refrigerant entering HXRW	Capacitance diaphragm, Setra Systems, Inc. Model C207 (0-250 psig)
P5	Pressure	Refrigerant leaving HXRW	Capacitance diaphragm, Setra Systems, Inc. Model C207 (0-250 psig)
P6	Pressure	Refrigerant at C suction	Capacitance diaphragm, Setra Systems, Inc. Model C207 (0-100 psig)
P7	Pressure	Refrigerant at C discharge	Capacitance diaphragm, Setra Systems, Inc. Model C207 (0-500 psig)
P8	Pressure	Refrigerant leaving outdoor check valve (cooling mode)	Capacitance diaphragm, Setra Systems, Inc. Model C207 (0-500 psig)
P9	Pressure	Refrigerant entering EVI (cooling mode)	Capacitance diaphragm, Setra Systems, Inc. Model C207 (0-500 psig)
P10	Pressure	Refrigerant leaving HXRAI (cooling mode)	Capacitance diaphragm, Setra Systems, Inc. Model C207 (0-250 psig)
FE1	Flowrate	Water (recirculating) leaving WT bottom	Positive displacement nutating disk with magnetic switch, BadgerMeter, Inc. Recordall Model 25 (0.25-25 gpm) with reed switch
FE2	Flowrate	Refrigerant in liquid line	Turbine with rate converter, EG&G Flow Technology, Inc. Omniflo FTO-3NIXW-LHA-1 with RC51-3-C-0000-6 (0.01-0.40 gpm)
FE3	Flowrate	Water entering WT from main	Positive displacement nutating disk with magnetic switch, BadgerMeter, Inc. Recordall Model 25 (0.25-25 gpm) with reed switch
FE4	Flowrate	Water leaving WT for HXWA	Positive displacement nutating disk with magnetic switch, BadgerMeter, Inc. Recordall Model 25 (0.25-25 gpm) with reed switch
FE5	Flow velocity	Air supply (at middle of duct)	Air velocity transmitter (including probe), Dwyer Instruments, Inc. Model 640-0 (0-12,000 fpm)

FE6	Flowrate	Air supply (at flow station)	Airflow measuring station with differential pressure transmitter, Air Monitor Corporation Fan-Evaluator (10-in. x 10-in. nominal rectangle, 0.69 ft^2) with Modus Instruments, Inc. Model T10-01E5 (0-0.1 in. water = 0-5VDC
FE7	Flow velocity	Air ventilation (at middle of duct)	Air velocity transmitter (including probe), Dwyer Instruments, Inc. Model 640-0 (0-12,000 fpm)
FE8	Flow volume	Condensate from HXRAI	Tipping bucket rain gauge, Texas Electronics, Inc. Model TR-525I
E1	Electric power	P	Hall-effect AC watt transducer; Ohio Semitronics, Inc. Model PC5-104CX5 (0-200 W = 0-5 VDC)
E2	Electric power	C	Hall-effect AC watt transducer; Ohio Semitronics , Inc. Model PC5-011CX5 (0-2 kW = 0-5 VDC)
E3	Electric power	FO	Hall-effect AC watt transducer; Ohio Semitronics, Inc. Model PC5-002X5 (0-1 kW = 0-5 VDC)
E4	Electric power	FI	Hall-effect AC watt transducer; Ohio Semitronics, Inc. Model PC5-002X5 (0-1 kW = 0-5VDC)

3.9 Data Acquisition

Instrument and transducer signals are wired to two Hewlett-Packard Company Model HP-E1345A 16-channel low-offset relay multiplexer VXI modules and four Hewlett-Packard Company Model HP-E1347A 16-channel thermocouple low-offset relay multiplexer VXI modules. Sequential voltage measurements of the multiplexer outputs are made by a Hewlett-Packard Company Model HP-E1326B 5½-digit multi-meter VXI module. All modules are controlled by two Hewlett-Packard Company Model HP-75000 Series B mainframes [one E1300A (plain front) and one E1301A (keyboard/ display front)]. These mainframes are, in turn, slaved to a Dell Optiplex GX620 personal computer (Intel Pentium 4, 2.99 GHz, 1.00 GB of RAM) with a Microsoft Windows XP Professional Version 2002 Service Pack 2 operating system that runs the associated Hewlett-Packard Company H-P VEE 4.01 data acquisition and reduction software. All of the data related to water-heating tests are collected in a separate data acquisition arrangement using a Campbell Scientific, Inc. CR23X Micrologger running C-S version 2.3. Data for the laboratory prototype tests are normally collected at 30-second intervals.

4. CONTROL STRATEGY

4.1 Primary Functions

The integrated heat pump is a single system intended to perform a variety of energy-related functions with efficiencies targeted to meet requirements for a ZEH of the future. The primary functions include

- space heating,
- space cooling,
- dehumidifying,
- air ventilating, and
- water heating.

4.2 System Components and Control Types

To accomplish these functions, various components must be combined to form the system. To achieve the desired capacities and efficiencies, they must be connected in an appropriate arrangement and controlled effectively. The approach builds, where possible, on methods employed in previous industry attempts to market AS-IHPs, including the Carrier/EPRI HydroTech 2000 and the Nordyne/EPRI Powermiser models (U.S. Patents 1991a, 1991b, 1991c, 1992, 1993, and 1997; Carrier 1989a, 1989b, and 1989c; Nordyne) noted earlier. Priority is given to heat pumping system operation in order to provide the needed home energy services as efficiently as possible. Only when heat pumping operation is unable to fully meet these needs is use made of less efficient secondary systems.

The major energy-consuming components for heat pumping are shown in Table 4.1. Minor energy-consuming components are shown in Table 4.2.

Table 4.1. Major energy-consuming components for heat pumping

Component	**Control type**
Refrigerant compressor	On/off, variable-speed
Indoor fan	On/off, variable-speed
Outdoor fan	On/off, variable-speed
Water pump	On/off, multiple-speed

Table 4.2. Minor energy-consuming components for heat pumping.

Component	**Control type**
Thermostat	Mode, time, temperature, humidity
Microprocessor(s)	Input/output
Refrigerant reversing valve actuator	Biposition (cooling or heating)
Electronic refrigerant expansion valve actuator(s)	Variable position (opening)
Heating water valve actuator	Biposition (open or closed)
Tempering water valve actuator	Variable position (opening)
Return air damper actuator	Biposition (open or closed)

Ventilation air damper actuator	Biposition (open or closed)

Sizing of the system is such that, barring component failure, heat pumping should provide adequate capacity for the space cooling, dehumidifying, and ventilation steady-state loads in the design house. The only loads likely to exceed temporarily the heat pumping system capabilities would be space heating and/or water heating under more extreme conditions (low outdoor temperatures and/or concentrated hot water usage). For these short-duration situations, substantially less efficient secondary energy-consuming components shown in Table 4.3 would be activated.

Table 4.3. Secondary energy-consuming components.

Component	**Control type**
Electrical resistance air heating element (indoor air handler)	On/off
Electrical resistance water heating element (upper, hot water storage tank)	On/off
Electrical resistance water heating element (lower, hot water storage tank)	On/off

Crucial to achieving the required performance is the incorporation of efficient variable-speed and/or multiple-speed operation over wide ranges in the major energy consuming components. The compressor and fans are essentially continuously variable over their entire ranges. The water pump motor has three discrete speeds. Thus, for each of these components, the control system must determine, for given conditions, whether the component should be on or off and, if on, how fast it should be running. For refrigerant expansion and water tempering control valves, the appropriate variable opening needs to be set per calls and conditions to provide the desired control condition, such as prescribed values of condenser subcooling, liquid tube temperature, or supply air temperature.

4.3 Operational Strategy

The general intent for the variable- and multiple-speed components is to optimize their speeds for any particular combination of loads so as to provide required capacities at maximum system efficiency. The reduction in HX loadings to just meet the current conditioning loads is the major contribution to higher system efficiency. Also inherent in the strategy is reducing system cycling losses by maximizing run times of the highly efficient components. Generally this implies operation at the lowest speeds that will meet the load requirements. Of course, this must be accomplished within the established performance envelope of each component. For example, in addition to the usual discharge temperature limit for a single-speed compressor, there are generally additional restrictions for a variable-speed compressor such as limits for suction pressure, discharge pressure, and compression ratio that vary for each speed range. In addition, there will likely be limits on ramp (increasing or decreasing) rates when speeds are to be changed. Other variable components, such as expansion valves, will be controlled over their available ranges to accommodate the desired capacities for selected modes. The remaining components require only binary decisions from the control system. In particular, the refrigerant reversing valve is either in the “cooling” position or the “heating” position; the water circuit valves are arranged so that water flows through the refrigerant-to-water HX, the water-to-air HX, or both; the return and ventilation air dampers are either open or closed; the air-heating electrical resistance elements are either on or off; and the water-heating electrical resistance elements are either on or off (upper and lower elements are not permitted to operate simultaneously).

The ASHRAE 62.2 (2004) requirement as applied to the candidate ZEH implies an average calculated air flow from the outdoors to the indoors. The intent of the control strategy is to use the system indoor fan to induce this amount of ventilation air flow while the system functions in nearly all the cooling, heating, and dehumidification control modes. When the system does not operate in one of these modes, a ventilation/flow timer will activate the indoor fan to induce about three times the calculated flow for 20 minutes of each hour in a ventilation operating mode to meet the requirement, while maintaining adequate air distribution uniformity.

4.4 Inputs to the Control System

To decide which components to turn on or off and at what speed or position, the control system requires various inputs (Table 4.4). Some are occupant-selected and some are inputs gathered from various sensors and clocks or timers.

Table 4.4. Control inputs and sources

Input	**Source**
Fan mode	Occupant-selected at thermostat
Heating/cooling or season selection	Occupant-selected at thermostat
Thermostat air temperature setting	Occupant-selected at thermostat
Thermostat air humidity setting	Occupant-selected at thermostat
Time-related options (setback, etc.)	Occupant-selected at thermostat
Thermostat air humidity	Sensor in thermostat
Thermostat air temperature	Sensor in thermostat
Supply air temperature	Sensor in indoor air handler section
Ambient air temperature	Sensor in outdoor air handler section
Compressor refrigerant discharge temperature	Sensor in compressor section
Accumulator refrigerant suction temperature	Sensor in compressor section
Indoor refrigerant liquid tube temperature	Sensor in indoor air handler section
Outdoor refrigerant liquid tube temperature	Sensor in outdoor air handler section
Upper tank water temperature	Sensor on upper water storage tank
Bottom tank water temperature	Sensor on bottom hot water storage tank
Indoor mid-coil temperature	Sensor in indoor air handler section
Outdoor mid-coil temperature	Sensor in outdoor air handler section
Indoor coil exit temperature	Sensor in indoor air handler section
Outdoor coil exit temperature	Sensor in outdoor air handler section
Clock	
Ventilation timer	
Indoor fan delay	
Compressor restart timer	
Defrost timer	

The most familiar occupant-selected inputs are fan mode and heating/cooling or season selections at the thermostat. Other common occupant inputs are the air temperature and air humidity set points at the thermostat, as well as time-related options such as day or night setback/setup settings. Common air sensor inputs for space conditioning are thermostat humidity and air temperatures at the thermostat, in the supply from the indoor air handler, and in the outdoor ambient air. Various refrigerant line temperatures including compressor discharge, accumulator suction, indoor liquid tube, and outdoor liquid tube are also employed. For water heating purposes, two additional temperature sensor inputs are normally employed: one near the bottom of the water storage tank and one in the upper section of the water storage tank. Other temperature sensors on the refrigerant-to-air HXs may be required for optimum control. Selected clock and timer inputs are also generally incorporated.

4.5 Operating Modes

Microprocessors will determine the operating mode of the system based on load demands indicated by the various inputs. The load demands may be for space cooling or heating, water heating, dehumidification, ventilation, outdoor coil defrosting, or selected combinations of these. The available primary system operating modes corresponding to the loads are:

- space cooling (AC),
- space cooling with enhanced dehumidification (ACEAD),
- space cooling plus water heating (ACWH),
- space cooling with enhanced space dehumidification plus water heating (ACEADWH),
- space heating (AH),
- space heating plus water heating (AHWH),
- water heating (WH),
- dehumidification (AD),
- dehumidification plus water heating (ADWH),
- ventilation (AV),
- ventilation with ventilation air dehumidification (AVVAD),
- ventilation plus water heating (AVWH), and
- outdoor coil defrosting (OCD).

The mode decision determines which components will operate (Table 4.5) and how they will be controlled. A description of the logic employed by the system for each primary mode follows.

Table 4.5. Mode/component matrix

Mode	Component										
	C	RV	FO	FI	Air return damper	Air ventilation damper	P	Water heating valve	Water tempering valve	Air resistance element	Water resistance elements
AC	on	cool	on	on	open	open					
ACEAD	on	cool	on	on	open	open					
ACWH	on	cool		on	open	open	on	open			either
ACEAD WH	on	cool		on	open	open	on	open			either
AH	on	heat	on	on	open	open				either	
AHWH	on	heat	on	on	open	open	on	open		either	either
WH	on	heat	on				on	open			either
AD	on	cool	on	on	open	open	on		open		
ADWH	on	cool		on	open	open	on	open	open		either
AV				on	open	open					
AVVAD	on	cool	on	on		open	on		open		
AVWH	on	heat	on	on	open	open	on	open			either

OCD	on	cool		on	open					either	

4.5.1 Space Cooling (AC)

When the space air temperature exceeds the thermostat cooling temperature set point, a space cooling load is indicated. In the absence of other indicated loads, the refrigerant reversing valve is situated in the cooling position, the return and ventilation air dampers are open, and the heat pump system provides air cooling in proportion to the load by varying the compressor speed (within the permissible envelope) at the rate needed to stay within the thermostat temperature deadband. The coincident indoor and outdoor fan speeds are adjusted in a prescribed manner based on the compressor speed. Heat removed from the indoor air and energy input by the compressor is rejected to the outdoor air. The refrigerant-to-water HX is active at low pump speed if beneficial water heating can be provided by the desuperheating function. When in desuperheating mode, it is recommended that water be allowed to exceed the nominal water heater set point of 120°F up to some specified maximum upper tank temperature of 140 to 155°F to take maximum advantage of heat recovery opportunities. (Note that local codes may require anti-scald measures for hot water at these temperature levels. See section 6 for discussion of AS-IHP performance with and without desuperheating operation.)

4.5.2 Space Cooling With Enhanced Dehumidification (ACEAD)

When (1) the space air temperature exceeds the thermostat cooling temperature set point and (2) the space relative humidity exceeds the thermostat humidity set point, both air cooling and dehumidification loads are indicated. In the absence of other indicated loads, the reversing valve is situated in the cooling position, the return and ventilation air dampers are open, and, as in the previous case, the heat pump system provides air cooling in proportion to the load by varying the compressor, indoor fan, and outdoor fan speeds. However, in this case, the air moisture removal rate is increased by reducing the indoor fan speed relative to the compressor speed.

The goal is to increase latent cooling capacity by setting the compressor and fan speeds so as to satisfy the sensible cooling load with a lowered indoor coil temperature to increase dehumidification. Heat removed from the indoor air and energy input by the compressor is rejected to the outdoor air. If the dehumidification load requirement is met before the space cooling load requirement, the control system transitions to the space cooling mode. If the space cooling load requirement is met before the dehumidification mode requirement, the control system transitions to the dehumidification mode. The refrigerant-to-water HX is active at lowest pump speed if beneficial water heating can be provided by the desuperheating function.

4.5.3 Space Cooling Plus "On Demand" Water Heating (ACWH)

When (1) the space air temperature exceeds the thermostat cooling temperature set point and (2) the lower water storage tank temperature is below its set point, both space cooling and water heating loads are indicated. In the absence of other indicated loads, as shown in Fig. 4.1, the reversing valve is situated in the cooling position, the return and ventilation air dampers are open, the water valve to the refrigerant-to-water HX circuit is open, the water pump is activated, and the heat pump system provides air cooling in proportion to the air cooling load by varying the compressor and indoor fan speeds.

In this mode, the outdoor fan is not active, so that most of the combined heat removed from the indoor air and energy input to the refrigerant from the compressor is transferred to the circulating water (in the refrigerant-to-water HX). The remainder of the heat is rejected by natural convection processes in the refrigerant line set and the outdoor refrigerant-to-air HX. If the water heating load requirement is met before the space cooling load requirement, the control system transitions to the space cooling mode. If the space cooling load requirement is met before the water heating mode requirement, the control system transitions to the water heating mode.

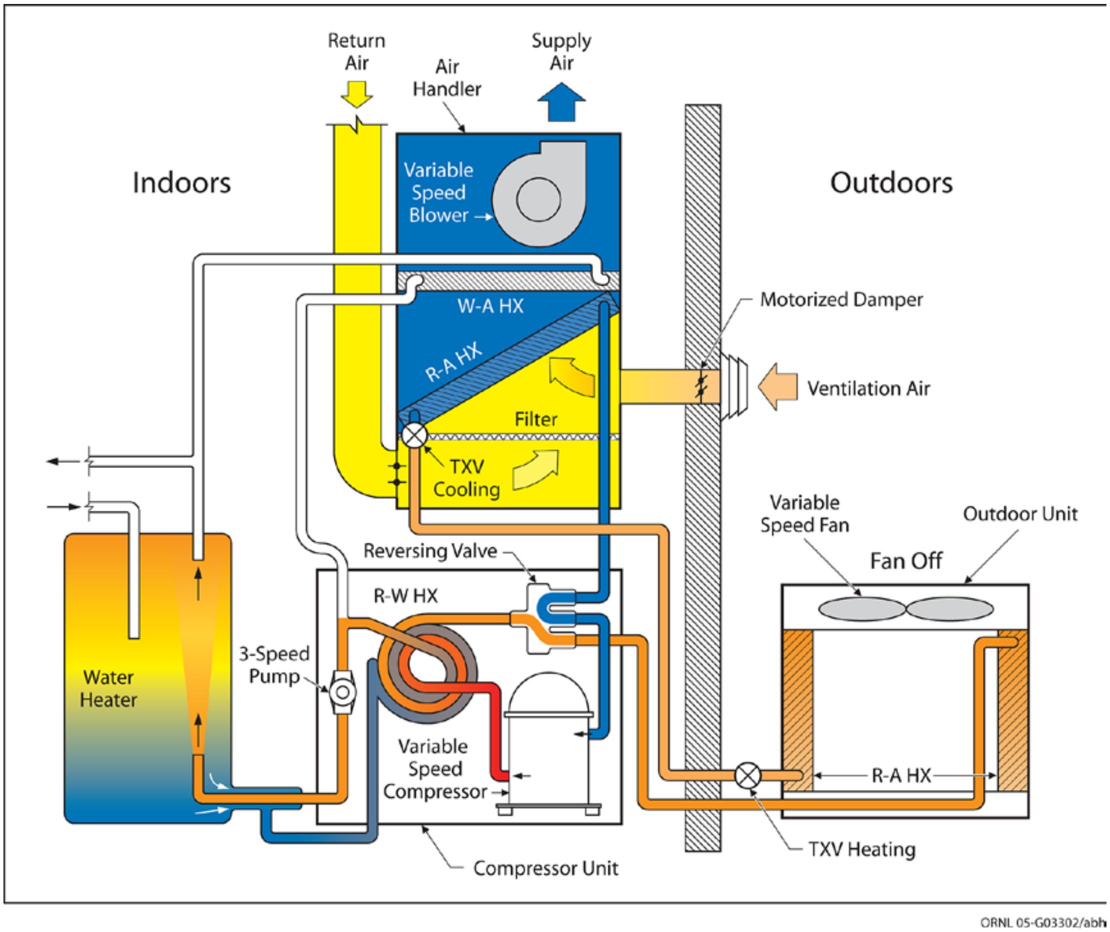

Fig. 4.1. Space cooling plus "on-demand" water heating (ACWH).

When (1) the lower water storage tank temperature is below its set point and (2) the upper water storage tank temperature is below its set point, a critical water heating load is indicated. In this situation, the control system activates the upper electrical resistance element to minimize the chance of running out of hot water.

4.5.4 Space Cooling With Enhanced Dehumidification Plus "On-Demand" Water Heating (ACEADWH)

When (1) the space air temperature exceeds the thermostat cooling temperature set point, (2) the space relative humidity exceeds the thermostat humidity set point, and (3) the lower water storage tank temperature is below its set point, three loads are indicated: space cooling, dehumidification, and water heating. In the absence of other indicated loads, the reversing valve is situated in the cooling position, the return and ventilation air dampers are open, the water valve to the refrigerant-to-water HX circuit is open, the water pump is activated, and the heat pump system provides air cooling in proportion to the space cooling load by varying the compressor and indoor fan speeds.

In this case, the air moisture removal rate is increased by reducing the indoor fan speed relative to the compressor speed. As before, the goal is to increase latent cooling capacity by setting the compressor and fan speeds so as to satisfy the sensible cooling load with a lowered indoor coil temperature to increase dehumidification. Also in this mode, the outdoor fan is not active, so that the majority of the combined heat removed from the house and energy input to the refrigerant from the compressor is transferred to the circulating water (in the refrigerant-to-water HX). The remainder of the heat is rejected by natural convection processes in the refrigerant line set and the outdoor refrigerant-to-air HX. If the dehumidification load requirement is met first, the control system transitions to the space cooling plus water heating mode. If the air cooling load requirement is met first, the control system transitions to the dehumidification plus water heating mode. If the water heating load requirement is met first, the control system transitions to the air cooling with enhanced dehumidification mode.

When (1) the lower water storage tank temperature is below its set point and (2) the upper water storage tank temperature is below its set point, a critical water heating load is indicated. In this situation, the control system activates the upper electrical resistance element to minimize the chance of hot water running out.

4.5.5 Space Heating (AH)

When the space air temperature is below the thermostat heating temperature set point, a space heating load is indicated. In the absence of other indicated loads, the reversing valve is situated in the heating position, the return and ventilation air dampers are open, and the heat pump system provides space heating in proportion to the load by appropriately varying the compressor, indoor fan, and outdoor fan speeds. The control logic varies the compressor and outdoor fan speeds to meet the space heating load at highest efficiency while the indoor speed is varied to maintain comfortable supply air temperatures. Heat removed from the outdoor air and energy input by the compressor is provided to the indoor air. If the space heating load exceeds the heat pump capacity, the control system activates the electrical resistance air heaters in the indoor unit. At outdoor temperatures below a specified minimum, the compressor is locked out and the total space heating load is met using auxiliary resistive heating. The refrigerant-to-water HX is active at lowest pump speed if beneficial water heating can be provided by the desuperheating function.

4.5.6 Space Heating plus "On-Demand" Water Heating (AHWH)

When (1) the space air temperature is below the thermostat heating temperature set point and (2) the lower water storage tank temperature is below its set point, both space heating and water heating loads are indicated. In the absence of other indicated loads, as shown in Fig. 4.2, the reversing valve is situated in the heating position, the return and ventilation air dampers are open, the water valve to the refrigerant-to-water HX circuit is open, the water pump is activated, and the heat pump system provides space heating in proportion to the space heating load by varying the compressor, indoor fan, and outdoor fan speeds. The heat rejected from the refrigerant is shared by the space (indoor refrigerant-to-air HX) and water (refrigerant-to-water HX) heating loads. The distribution of heat between these two loads depends primarily upon the indoor fan speed, which is controlled to meet the space heating load. As indoor fan speed increases, so does the proportion of rejected heat supplied to the indoor air. The compressor speed is to be set as a prescribed function of outdoor ambient in this mode between minimum and maximum water heating speeds with the indoor fan speed providing the control to meet the space heating load.

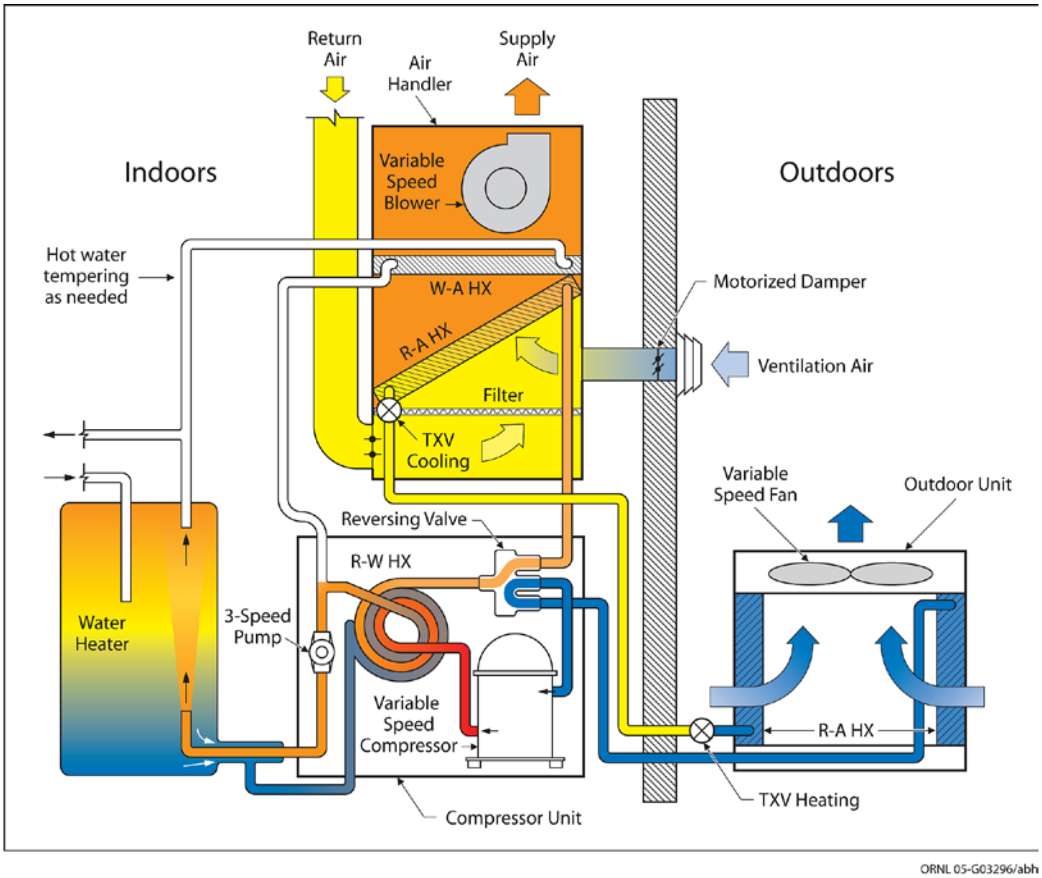

Fig. 4.2. Space heating plus "on-demand" water heating (AHWH).

If the space heating requirement exceeds the capacity of the heat pump, the electrical resistance air heaters in the indoor unit are activated to provide supplemental heat, and the water pump is locked out (terminating water heating by the heat pump). In this circumstance, the lower electric resistance heating element in the storage tank is activated to provide water heating. If the water heating load requirement is met before the space heating load requirement, the control system transitions to the space heating mode. If the space heating load requirement is met before the water heating mode requirement, the control system transitions to the water heating mode. At outdoor temperatures below a specified minimum, the compressor is locked out and both the total air heating load and the total water heating load are met using their respective electrical resistance heating elements.

When (1) the lower water storage tank temperature is below its set point and (2) the upper water storage tank temperature is below its set point, a critical water heating load is indicated. In this situation, the control system activates the upper electrical resistance element to minimize the chance of running out of hot water.

4.5.7 Demand Water Heating (WH)

When the lower water storage tank temperature is below its set point, a water heating load is indicated. In the absence of other indicated loads, the reversing valve is situated in the heating position, the return and ventilation air dampers are closed, the water valve to the refrigerant-to-water HX circuit is open, the water pump is activated, and the heat pump system provides water heating in proportion to the water heating load by varying the compressor and water pump speeds. In this mode, the indoor fan is not active, so that most of the combined heat removed from the outdoor air and energy input to the refrigerant from the compressor is transferred to the circulating water (in the refrigerant-to-water HX). If the capacity of the heat pump is insufficient to meet the water heating load, the control system will activate the lower electrical resistance water heating element in the hot water storage tank. At outdoor temperatures below a specified minimum, the compressor is locked out and the total water heating load is met using electrical resistance heating means.

When (1) the lower water storage tank temperature is below its set point and (2) the upper water storage tank temperature is below its set point, a critical water heating load is indicated. In this situation, the control system activates the upper electrical resistance element to minimize the chance of running out of hot water.

4.5.8 Dehumidification (AD)

When the relative humidity exceeds the thermostat humidity set point, a dehumidification load is indicated. In the absence of other indicated loads, the reversing valve is situated in the cooling position, the return and ventilation air dampers are open, and the heat pump system cools the circulated air and removes moisture from it in proportion to the dehumidification load by varying the compressor, indoor fan, and outdoor fan speeds. In this case, the air moisture removal rate is enhanced by reducing the indoor fan speed relative to the compressor speed. Heat removed from the indoor air and energy input by the compressor is rejected to the water tank first (via desuperheating in the refrigerant-to-water HX) and then to the outdoor air. The water valve to the water-to-air tempering HX circuit is open and the water pump is activated to allow hot water from the storage tank to be used to provide reheat to maintain the thermostat air temperature set point.

4.5.9 Space Dehumidification Plus Water Heating (ADWH)

When (1) the space relative humidity exceeds the thermostat humidity set point and (2) the lower water storage tank temperature is below its set point, both dehumidification and water heating loads are indicated. In the absence of other indicated loads, as shown in Fig. 4.3, the reversing valve is situated in the cooling position, the return and ventilation air dampers are open, and the heat pump system cools the circulated air and removes moisture from it in proportion to the dehumidification load by varying the compressor, indoor fan, and outdoor fan speeds. In this case, the air moisture removal rate is enhanced by reducing the indoor fan speed relative to the compressor speed. Both water circuit valves are open and the water pump is activated to permit flow through both the refrigerant-to-water HX and the water-to-air HX in the indoor unit. Heat is rejected to the water from the discharge refrigerant and (a smaller amount) rejected by the water to the dehumidified air in the indoor unit to provide reheat to maintain the thermostat air temperature set point. Water-pump speed may be increased to maximum to provide sufficient water flow for both functions. Refrigerant discharge heat in excess of that which can be absorbed in the refrigerant-to-water HX is rejected by natural convection through the outdoor refrigerant-to-air HX. If the dehumidification load requirement is met before the water heating load requirement, the control system transitions to the water heating mode. If the water heating load requirement is met before the dehumidification load requirement, the control system transitions to the dehumidification mode.

When (1) the lower water storage tank temperature is below its set point and (2) the upper water storage tank temperature is below its set point, a critical water heating load is indicated. In this situation, the control system activates the upper electrical resistance element to minimize the chance of running out of hot water.

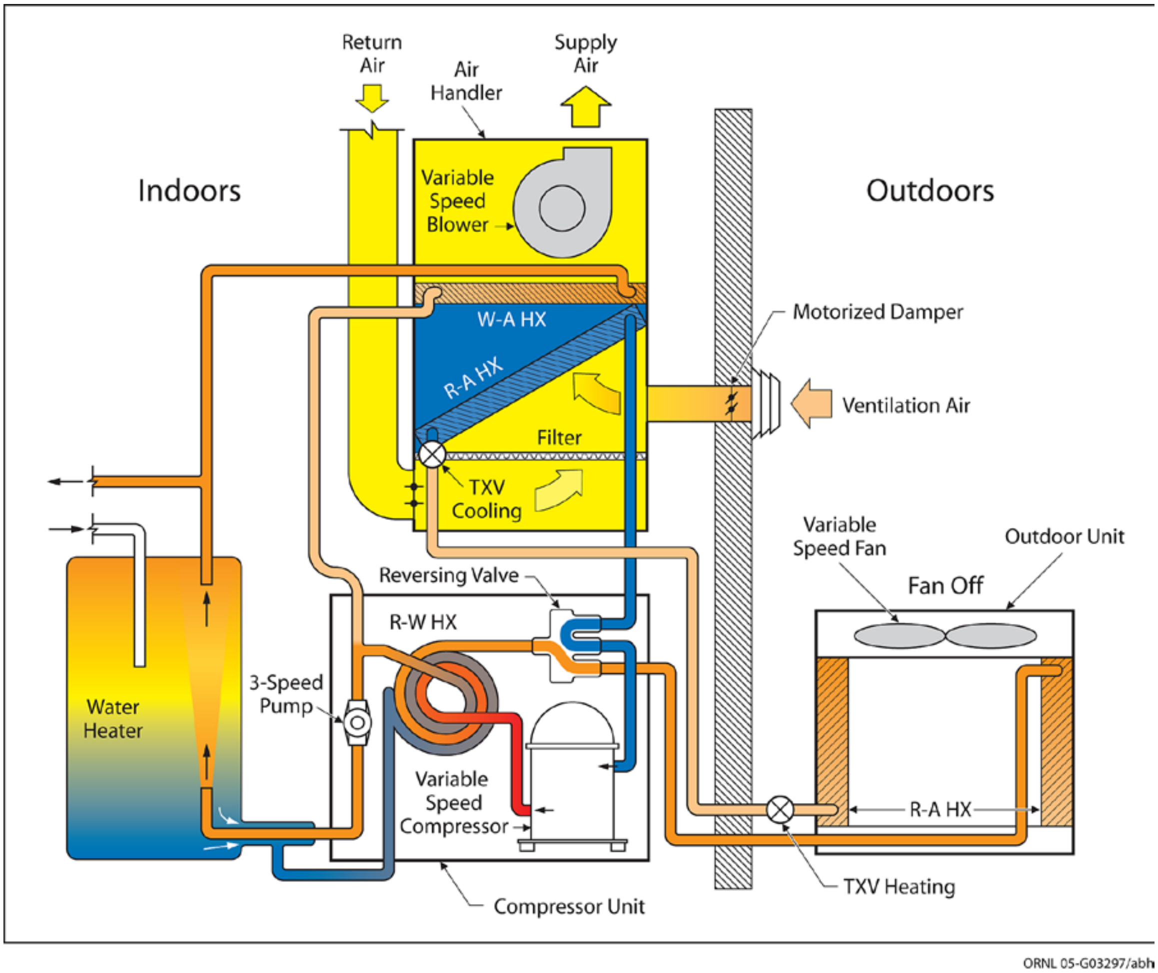

Fig. 4.3. Space dehumidification plus water heating (ADWH).

4.5.10 Ventilation (AV)

When the ventilation air flow/timer signals that outdoor air is needed to meet minimum requirements, a ventilation load is indicated. In the absence of other indicated loads, the return and ventilation air dampers are open and the indoor fan activated to bring in the prescribed amount of outdoor air. The timer gives such a signal if the indoor air handler has not operated in another mode for one hour. Equal amounts of ventilation air and return air are proposed to temper the outdoor air and promote effective air distribution. If further indoor air tempering is desired, water could be circulated through the water-to-air tempering coil to provide this service.

4.5.11 Ventilation With Ventilation Air Dehumidification (AVVAD)

When (1) the ventilation air flow/timer signals that outdoor air is needed to meet minimum requirements and (2) the humidity ratio of the outdoor air (as determined from outdoor sensors) is above a desired set point, both ventilation and ventilation air dehumidification loads are indicated. In the absence of other indicated loads, as shown in Fig. 4.4, the return air damper is closed, ventilation air damper is open and the indoor fan activated to bring in the prescribed amount of outdoor air, unmixed with any return air. In this way, only ventilation air is dehumidified to obtain the maximum moisture removal at a given evaporator coil temperature. The reversing valve is situated in the cooling position and the heat pump system cools the ventilation air and removes moisture from it in proportion to the dehumidification load by varying the compressor and outdoor fan speeds. The compressor speed is increased if the indoor relative humidity level is sensed to increase while in AVVAD mode. Heat removed from the ventilation air and energy input by the compressor is rejected to the water tank (via desuperheating in the refrigerant-to-water HX) and to the outdoor air. The water valve to the water-to-air tempering HX circuit is open, and the water pump is activated to allow heat from the storage tank to be used to provide reheat to maintain the thermostat air temperature set point.

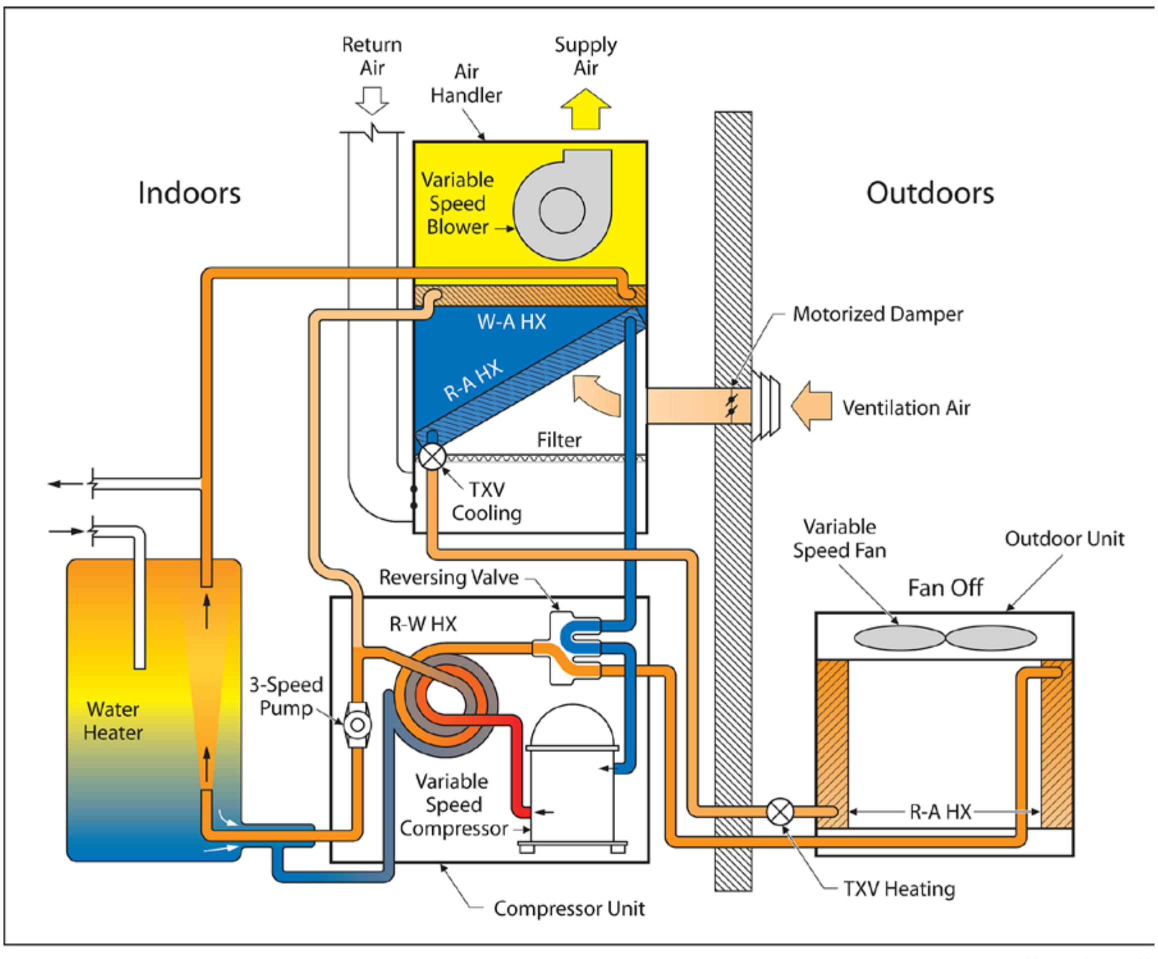

Fig. 4.4. Ventilation with ventilation air dehumidification (AVVAD).

4.5.12 Ventilation Plus Water Heating (AVWH)

When (1) the ventilation air flow/timer signals that outdoor air is needed to meet minimum requirements and (2) the lower water storage tank temperature is below its set point, both ventilation and water heating loads are indicated. In the absence of other indicated loads, as shown in Fig. 4.5, the return and ventilation air dampers are open and the indoor fan activated to bring in the prescribed amount of outdoor air. The reversing valve is situated in the heating position, the water valve to the refrigerant-to-water HX circuit is open, the water pump is activated, and the heat pump system provides water heating in proportion to the water heating load by varying the compressor and outdoor fan speeds. The water valve to the water-to-air tempering HX circuit is open as well to allow hot water from the storage tank to be used to provide air tempering heat as needed to warm the ventilation air (up to the thermostat air temperature set point but no further). The combined heat removed from the outdoor air and energy input to the refrigerant from the compressor is transferred primarily to the circulating water (in the refrigerant-to-water HX) with a small fraction rejected to the supply air through the indoor refrigerant-to-air HX. This combined water heating mode is enabled only if the ventilation air is less than 55-60°F (during shoulder seasons and mild periods during the heating season) when modest heating of the ventilation air will be acceptable. If the capacity of the heat pump is insufficient to meet the water heating load, the control system will activate the lower electrical resistance water heating element in the hot water storage tank.

When (1) the lower water storage tank temperature is below its set point and (2) the upper water storage tank temperature is below its set point, a critical water heating load is indicated. In this situation, the control system activates the upper electrical resistance element to minimize the chance of running out of hot water.

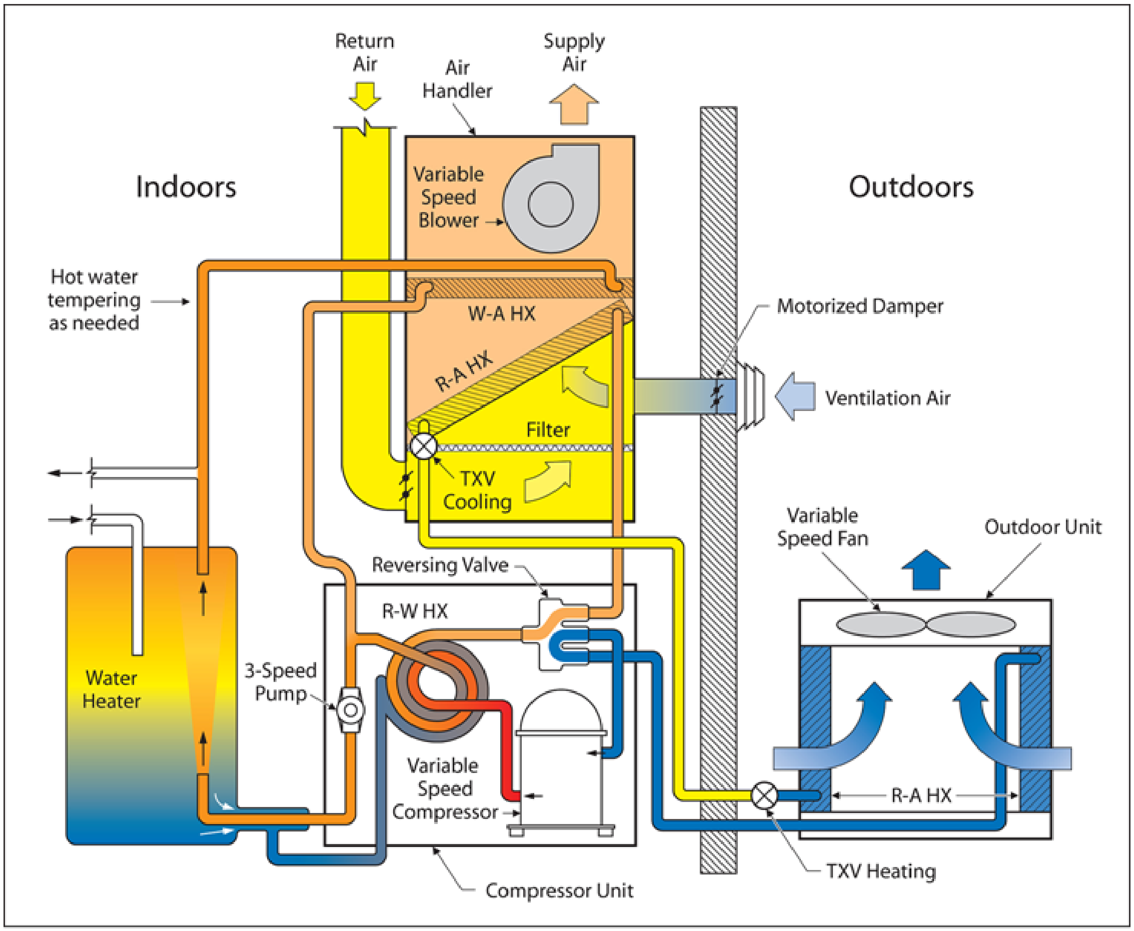

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Fig. 4.5. Ventilation plus water heating (AVWH).

4.5.13 Outdoor Coil Defrosting (OCD)

When (1) the outdoor air temperature is below a specified minimum (45°F, for example), and (2) a "defrost need" signal is received from a demand defrost sensor (e.g., difference between the temperature of refrigerant exiting the outdoor refrigerant-to-air HX and the temperature of the outdoor air exceeds some defined limit) the need for a defrost cycle is indicated. In this case, the refrigerant reversing valve is situated in the cooling position, the return damper is open, and the ventilation air damper is closed, and the heat pump system operates the compressor and indoor fan to remove heat from the indoor air and move the combined heat and energy input by the compressor to the outdoor coil. With the outdoor fan off, the bulk of this energy is employed in melting the accumulated frost layer.

5. TEST DATA, MODEL VALIDATION, AND SYSTEM OPERATIONAL DESIGN AND PERFORMANCE

5.1 Laboratory Testing and Results

5.1.1 Steady-State Space Cooling Mode

We conducted a number of initial steady-state tests to determine the most suitable indoor airflow, compressor speed, and refrigerant charge at the 95°F ambient design condition. The basis for selection of the optimum charge was to conduct a steady-state test with the superheat from the indoor coil set to 3 – 5°F and to examine the level of liquid-line subcooling, adding charge to increase the subcooling or removing charge to decrease it until maximum EER was found. A series of four tests were conducted with the laboratory prototype R-22 heat pump system at the four design conditions prescribed for variable speed cooling mode testing: 95, 87, 82 and 67 °F outdoor dry bulb temperature (DBT). Table 5.1 shows the results from these tests.

Table 5.1 Steady-state cooling mode performance

Test No.	Outdoor DBT (°F)	Outdoor air flow (scfm)	Indoor DB/WB (°F)	Indoor airflow (scfm)	Compressor speed (Hz)	Cooling capacity (Btu/h)	EER (Btu/ W-h)	SHR
16	95	1134	80/67	487	79	14888	12.0	0.743
17	87	994	80/67	349	58	10739	14.8	0.749
18	82	850	80/67	243	36	7216	18.6	0.739
20	67	834	80/67	241	36	7454	23.6	0.727
22	82	823	80/67	169	36	6691	17.0	0.665

Tests No. 16, 17, 18, and 20 were cooling-only under ARI indoor conditions [DBT and wet bulb temperature (WBT)] and outdoor conditions (DBT only) needed for an SEER determination. The tests shown in this table illustrate the range of cooling capacities that the IHP can deliver under ARI conditions, closely matching an expected load for a near-zero-energy house, and also the efficiency in terms of EER. Charge level was optimized at the desired design cooling capacity of 1-1/4 tons by finding the condenser subcooling level giving near-maximum EER at a fixed low evaporator superheat. The compressor speed was varied during this optimization to maintain the design cooling capacity. Test No. 16 is the result of this series of tests. Design indoor fan speed was set to keep the sensible heat ratio (SHR) below 0.75. For tests 17, 18, and 20, the charge was held fixed and a low level of evaporator superheat was maintained while the compressor and fan speeds were lowered with ambient as determined from modeling. These results show that with careful selection of variable-speed components and applying them to a conventional minimum SEER outdoor unit (with replaced compressor), it is possible to reach high steady-state efficiencies.

Test No. 22 was one run where we reduced the indoor airflow to 70% of the airflow in Test 18 to determine the amount of improved dehumidification. The SHR decreased by 10%.

5.1.2 Steady-State Water Heating–Space Cooling Mode

We conducted five tests of the IHP under steady-state conditions of simultaneous space cooling and water heating. For these tests, we maintained a fixed water temperature in the tank by bleeding cool water into the tank as warm water was removed from the tank. These tests allowed us to set the superheat at the exit of the indoor coil to be in the range of 3 – 5°F and, by adjusting the refrigerant charge, to set the subcooling in the range of 5 – 12°F with the system balanced and operating under fixed conditions. For each of these tests, the indoor chamber was maintained at controlled conditions of 80°F/67°F and the outdoor chamber at ARI conditions. The results of these tests are shown in Table 5.2.

Table 5.2 Steady-state space cooling + water heating performance

Calculation	Test case				
	5	6	8	10	12
EER: Space cooling (Btu/W-h)	14.92	16.84	17.15	17.23	17.03
EER: Space cooling + water heating (Btu/W-h)	30.01	32.25	33.14	33.14	32.84
Heat to water using R-W HX (Btu/h)	7029	6469	9941	9499	9729
Cooling to space (Btu/h)	6953	7065	10660	10280	10480
Sensible heat ratio (SHR)	0.746	0.739	0.732	0.775	0.775
Avg. tank temperature (°F)	83.7	83.8	71.3	70.2	71.6
Avg. compressor power (W)	361.9	372.8	569.4	541.3	558.2
Avg. pump power (W)	85.8 (high speed)	28.3 (low speed)	26.4	28.5	28.9
Avg. indoor fan power (W)	18.3	18.5	26.1	27.3	28.4
Indoor fan flow rate (cfm)	276.2	278.1	401.6	409.1	412.3
Outdoor ambient (°F)	82	82	87	87	87*

*Outdoor chamber fan off

The performance of the IHP in this limited test series is impressive, with an overall EER (space cooling and water heating) of more than 30. Test cases 5 and 6 show the impact of running the water pump at different speeds. The effect of the pump speed (and power) on the EER as shown in these test cases suggests that a low pump speed provides a boost in total EER, although as expected, less water heating is accomplished. Since the pump runs any time that water heating and/or air tempering is done, it was important to examine the benefit of pump speed on overall performance. In a series of short-term tests, we determined that the highest pump speed buys little in overall performance as compared to the increase in pumping power.

5.1.3 Dynamic Water Heating–Space Cooling Mode

A series of dynamic water heating tests where we allowed the water tank to heat up were conducted following some of the steady-state cooling + water-heating tests. An example is a test (Case 9) where the tank began at 70°F and was heated by the heat pump (no resistance heating), and for most of the test, a low pump speed was used. The heat to water was provided by heat from the indoor coil as well as compressor work. 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. Staying within the compressor operating envelope is a design requirement. In Case 9, we examined among other things the ability of the IHP in the water-heating mode to stay within the compressor working pressure envelope. The results of this analysis are shown in Figure 5.1. The envelopes cover acceptable compressor discharge and suction pressures for the variable-speed compressor used in the IHP. The blue envelope covers acceptable conditions for compressor operation at a speed of 30 Hz. If the compressor is operated at 100 Hz, the maximum discharge pressure is the same as for 30 Hz operation. The red region contains acceptable compressor conditions in the range of 45 – 90 Hz. As an example, for Case 9, we plotted the compressor suction and discharge pressures for various average tank temperatures during the heating process.

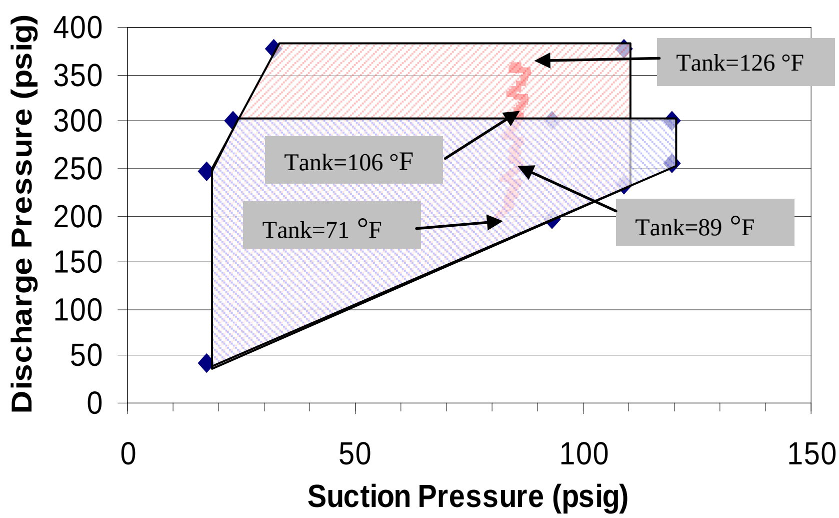

Figure 5.1. Tank heat-up on compressor map.

It can be seen that compressor discharge and suction pressures remain within the acceptable operating envelope as the tank heats up. Even at an average tank temperature of 126°F, the pressures across the compressor are within the envelope.

In Case 9, we also examined how the EER varies with elapsed time (Figure 5.2) and with average tank temperature (Figure 5.3). The top curve in each of these plots shows the combined EER that includes the cooling benefit as well as water heating, and the bottom curve is the EER for cooling alone. The beginning EER is high as expected; however, even at the end of the test with the ending average tank temperature of 126°F, the combined EER is about 17 Btu/W-h, which gives an ending coefficient of performance (COP) of 5.0.

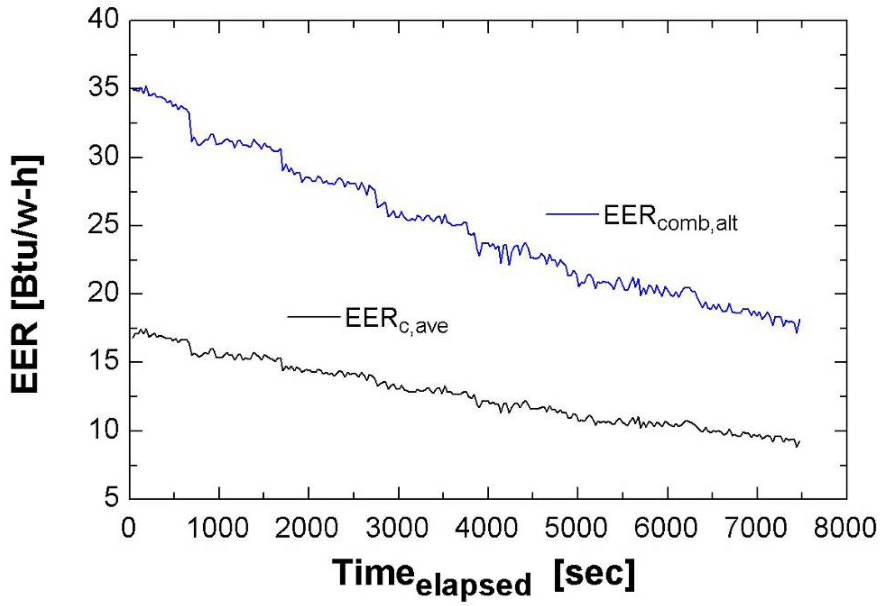

Figure 5.2. EER performance in water heat-up test (Case 9).

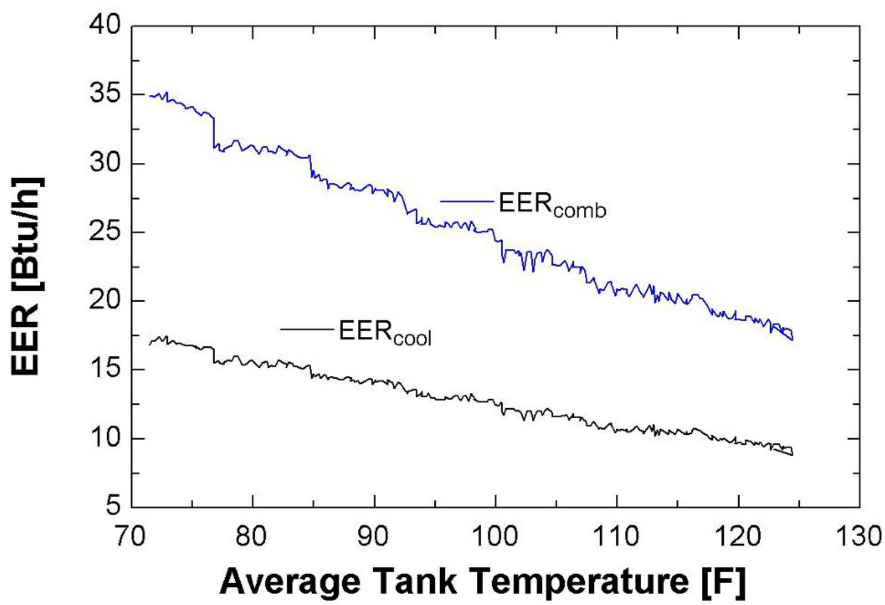

Figure 5.3. EER performance with average water tank temperature in heat-up Case 9.

5.1.4 Dedicated Dehumidification Mode

Tests were also run to examine the performance of the IHP as a dedicated dehumidifier (dehumidification mode). 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 at 120°F inlet water to the tempering coil, there would be sufficient reheating of the air leaving the indoor coil to establish space-neutral conditions. We have completed one test of the dehumidifying/reheating performance of the IHP. We started the test with a cool tank (70°F) of water, and we adjusted the hydronic valves so that water heated by the refrigerant-to-water HX would heat the tank at the same time as water from the tank goes to the reheat coil. Since the water in at the top of the tank is initially cool (70°F), there would be little reheating. However, as the tank warms up, more reheating would take place and the overall sensible cooling of the air from the indoor unit would drop. Figure 5.4 taken from this test shows the performance of the IHP in this mode of operation.

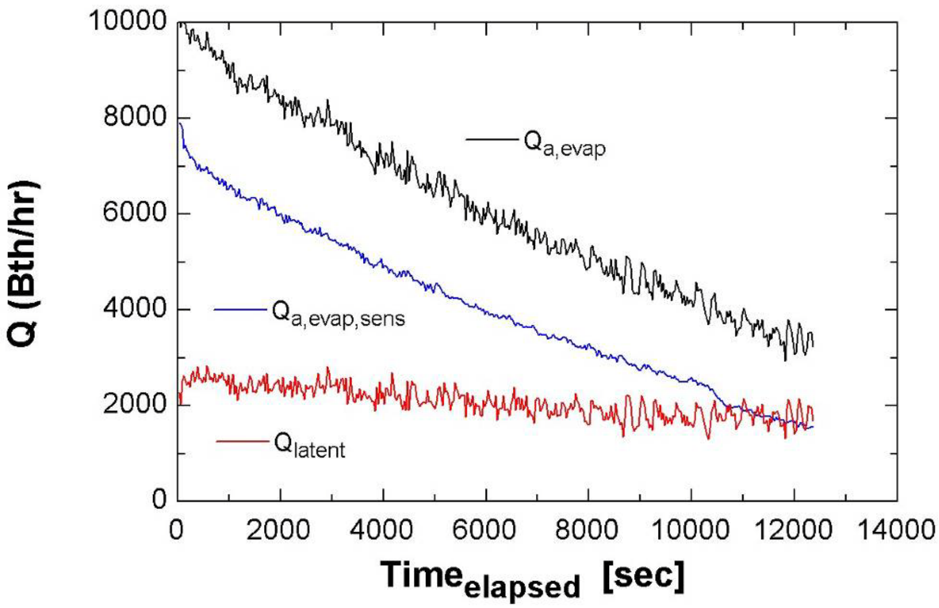

Figure 5.4. Performance of system in dehumidification + water heating mode.

Over the duration of this test, the latent heat removal was about 2000 Btu/h, and the sensible cooling of the overall air stream dropped to about the same level. At this point the SHR was approximately 0.50, indicating a high level of dehumidification while returning the air to the space at about 77°F. (Note that had we chosen to return the supply air at 80°F, using more tempering water flow or higher inlet water temperature, the assumed mixed return and ventilation air temperature used as the entering air temperature, the ending SHR would be zero.)

5.1.5 Tank Loss Characterization

As with any system that provides domestic hot water, there is a difference in the hot water that is delivered to the user and that which is provided to the tank. Tank standby losses to account for this difference as well as the tank heat loss coefficient (UA) were determined by measurement. Starting with a hot tank at about 125°F in the chamber, we closed all flows to the tank and allowed the tank to cool down in the environmental chamber which was maintained at room temperature. The heat losses from the tank were determined from the change in average tank temperature over a measured duration of time and with the assumption that the tank and its contents were thermally “lumped.” We also accounted for the thermal mass of the tank itself [tank weight is 120 lb and the tank is made entirely of steel (c_p = 0.11 Btu/lb/°F)]. Tank cool down is described by Newton’s Law of cooling in which the cooling rate is proportional to the difference between the average tank temperature and the ambient temperature, and a hyperbolic cooling trend is found. However, over the limited tank temperature changes and relatively small time change, the drop in tank temperature with time was very close to linear (R squared = 0.9999). Tank heat losses as determined by the time rate of change of tank and water temperature were set equal to the convective heat losses between the exterior of the tank and the room ambient. From these measurements, an average tank UA = 2.7 Btu/h/°F was calculated. Based on this value, if the tank were at a 120°F setpoint in a 70°F room, standby losses would be 135 Btu/h. Although small, standby losses will affect the heating/cooling loads depending on the location of the water tank inside a home as well as the quality of hot water delivered to the user.

5.2 Model Validation

The ultimate value of the IHP will be determined by its annual energy savings as compared to other alternatives. Energy savings reduce operating costs, make the IHP more affordable, and therefore improve the IHP’s marketability. In the first step of this analysis process, we used the laboratory test data to calibrate the component performance models in a simulation model, such as the compressor, fans, pump, and HXs.

The manufacturer’s compressor performance map was adjusted for the effects of inverter efficiency and operation at speeds above and below the rated speed, as well as for differences between the compressor map and the actual compressor performance. At each speed for which a compressor map was used, we adjusted the compressor power and mass flow corrections available as model inputs to reflect the differences between the measured power and refrigerant mass flow and that predicted by the compressor maps. This was possible because we measured the compressor inlet refrigerant pressure and temperature and the exit pressure, which is the information required to identify an operating point on a compressor performance map.

For all the tests, we used EES (Engineering Equation Solver) data reduction programs to compare the measured compressor flow and power input to the maps for the appropriate speeds and to calculate the required correction factors. This same program was used to calculate the delivered capacities of the cooling and water heating coils, calculate 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 detailed results from the data reduction programs were used to calibrate the predictions of the ORNL HPDM (Rice and Jackson 2002) for the range of space cooling and water heating tests performed.

5.3 Use of HPDM in Design Optimizations of AS-IHP

Using this calibrated model, system simulations were generated for the appropriate operational speeds and ambient conditions for the following important modes of operation with regard to calculating energy use:

1. space cooling, without and with full condensing water heating,

2. space heating, and
3. dedicated water heating, with outdoor coil as the heat source for heating water in the shoulder months and in the heating season.

Initially, speeds of the indoor blower and outdoor fan were individually set relative to the compressor speeds based on optimization simulations done early in FY05. These controls and the nominal flow rates were refined based on the system test data taken in late FY05. In general, the indoor blower speed was set in cooling mode to maintain an equivalent to lower SHR as that at the design cooling condition as the compressor speed was reduced.

This validated model was used in further heat pump design optimization and control assessments for the various operation modes and prototype specification in FY06 based on the laboratory R-22 compressor, air-moving, and HX components. In FY07, a suitable compressor map for a state-of-the-art R-410A rotary compressor over a wide range of compressor speeds (30 to 120 Hz) was obtained and a revised design optimization and control assessment was performed based on the improved compressor characterization and preferred HFC refrigerant, R-410A. Inverter losses of 5% were assumed for the 60 – 120 Hz speed range, while 8% inverter loss was assumed at the minimum speed of 30 Hz. The main air-to-refrigerant HX design change made was to go from the 5-circuit, 3/8-in. tube indoor coil of the breadboard to a 3-circuit, 5/16-in. tube design. This improved the coil performance with R-410A, especially at low-speed operation in space cooling and heating operation.

The following sections describe our present design optimized for the state-of-the-art R-410A brushless, DC-driven rotary compressor and its performance over a range of operating modes. Finally, the predicted performance is compared to the published data for the Carrier HydroTech 2000 integrated heat pump discussed in Section 2.

5.4 AS-IHP System Speed Control Relationships

With a variable-speed heat pump, as the compressor speed varies to match the load, the indoor and outdoor airflows should be adjusted in somewhat similar measure to obtain highest efficiency (Miller 1988, Rice 1992). In addition, the refrigerant flow control should be adjusted with compressor speed to obtain optimal condenser exit subcooling, if possible, while the compressor inlet superheat is maintained at a value sufficient to maintain low superheat levels leaving the evaporator.[1]

As the compressor speed generally has a stronger effect on these optimums than the outdoor ambient conditions, this variable was used as the independent control variable for the AS-IHP design. HPDM was used with the breadboard IHP design and component performance specifications and data to determine an optimal set of control relationships for indoor blower and outdoor fan motor frequencies (directly proportional to speed) and condenser subcooling vs. compressor speed. This was done for a representative (target) set of cooling and heating ambient temperatures vs. compressor speed so that the effect of ambient conditions was also factored into the analysis.

[1]Obtaining this optimal control over a range of ambients will generally require some adjustable level refrigerant charge storage means such as a suction line accumulator or other devices which can hold excess charge at some conditions and deliver needed charge back to the system at other conditions and operation modes.

5.4.1 Compressor Speed Ratio vs. Ambient in Cooling and Heating Modes

Figures 5.5 and 5.6 show the assumed target relationships between the compressor speed ratio (operating speed to nominal, design speed) and the ambient temperature in cooling and heating modes, respectively, where ratios are shown for generality. (As all of the motors here are synchronous, the frequency ratios and the speed ratios are the same.) The design compressor frequency (at which the design cooling capacity is achieved) in our breadboard system is 79 Hz (speed or Hz ratio = 1.0). The desired speed range is wider in the heating mode than in the cooling mode to provide more heating capacity at ambients below about 32°F where typically the capacity of a single-speed compressor (at a speed ratio of 1) becomes insufficient to meet the heating load. Here we are proposing a maximum speed ratio of 1.5 or 50% overspeed to 118 Hz in this case. Rice (1988) has shown that compressors can be operated in constant power overspeed conditions in the heating mode since the torque requirements decrease along with the ambient temperatures. Because of this, the motor can be run at reduced volts/hertz ratios (fixed line voltage / increasing frequency) at these lower temperature heating conditions. This overspeed operation results in a significant increase in the rated HSPF per the DOE rating procedure (Domanski 1988). The minimum assumed speed for our analysis was 28 Hz for both cooling and heating modes (0.35 speed ratio).

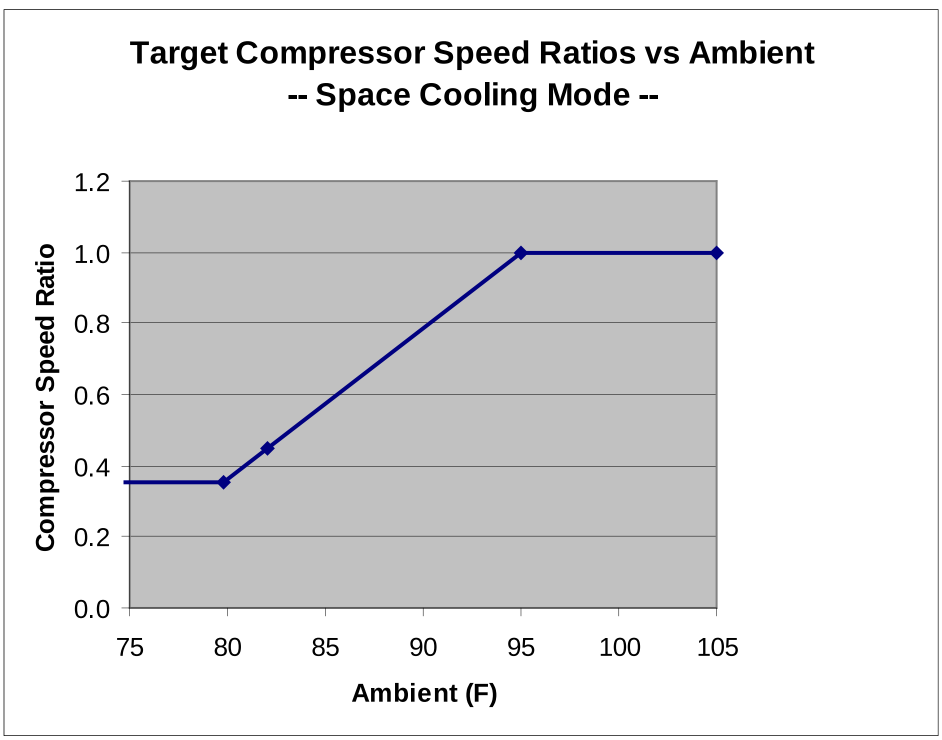

Fig. 5.5. Target compressor speed ratios vs. ambient in the space cooling mode.

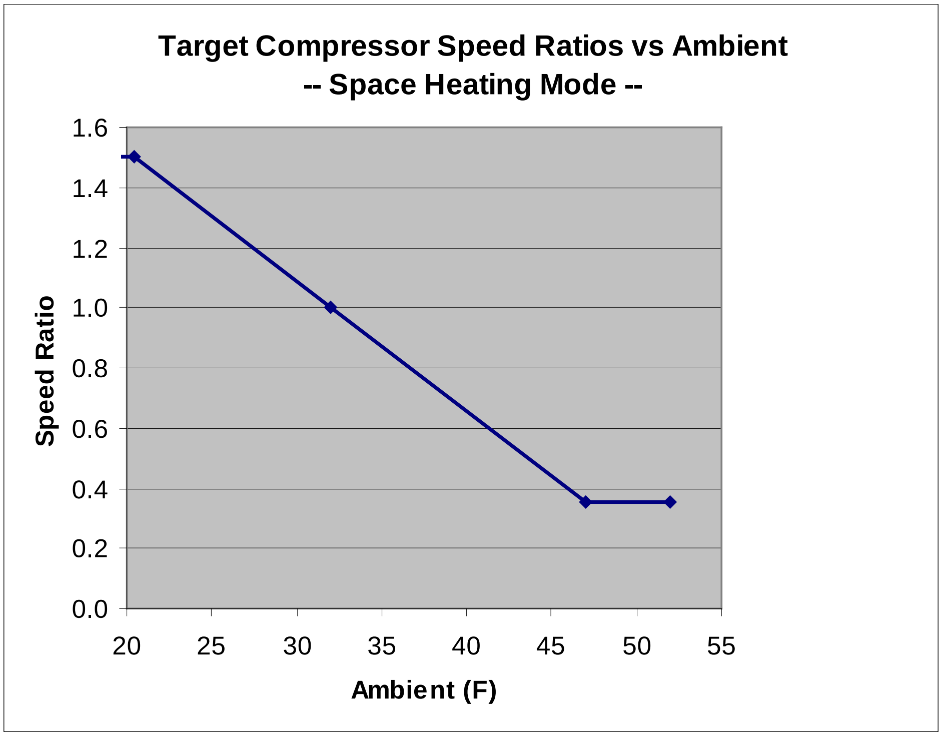

Fig. 5.6. Target compressor speed ratios vs. ambient in the space heating mode.

5.4.2 Control Parameters vs. Compressor Speed Ratio in the Space Cooling and Heating Modes

The selected indoor blower and outdoor fan speed ratios and condenser subcooling control are shown in Figs. 5.7 and 5.8 as functions of the compressor speed ratios for cooling and heating mode, respectively. The nominal airflows are 500 cfm indoor and 1200 cfm outdoor. In the cooling mode, shown in Fig. 5.7, the airflow ratios drop off more slowly than the compressor speed, since the capacity and thus HX loading drops more gradually than compressor speed as well. (A one-to-one speed ratio relationship is shown by the dotted gray line.) This is because as the speed is lowered and the HX unloads, the evaporator pressure rises with increases in the refrigerant suction density entering the compressor. This higher density tends to resist the capacity drop from the compressor speed reduction. The selected indoor airflow trends with compressor speed are also strongly determined by the requirement to maintain approximately the same sensible-heat-ratios over the ambient range. In the heating mode, the airflows again drop off more slowly than the compressor speed for similar HX loading reasons. There is also a need based on comfort considerations to maintain supply air temperatures around 95°F or higher over the range of compressor speeds. At reduced compressor speeds in both modes, the optimal subcooling levels are lower as found by Miller (1988).

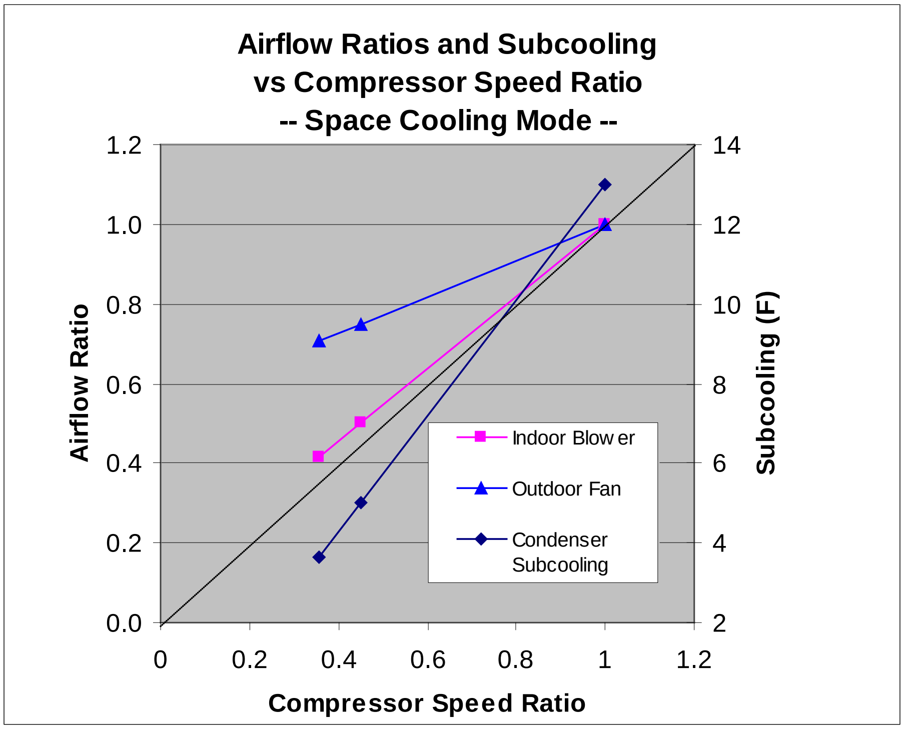

Fig. 5.7. Control parameters vs. compressor speed ratio in the space cooling mode (dotted gray line signifies 1:1 compressor speed to air flow ratio).

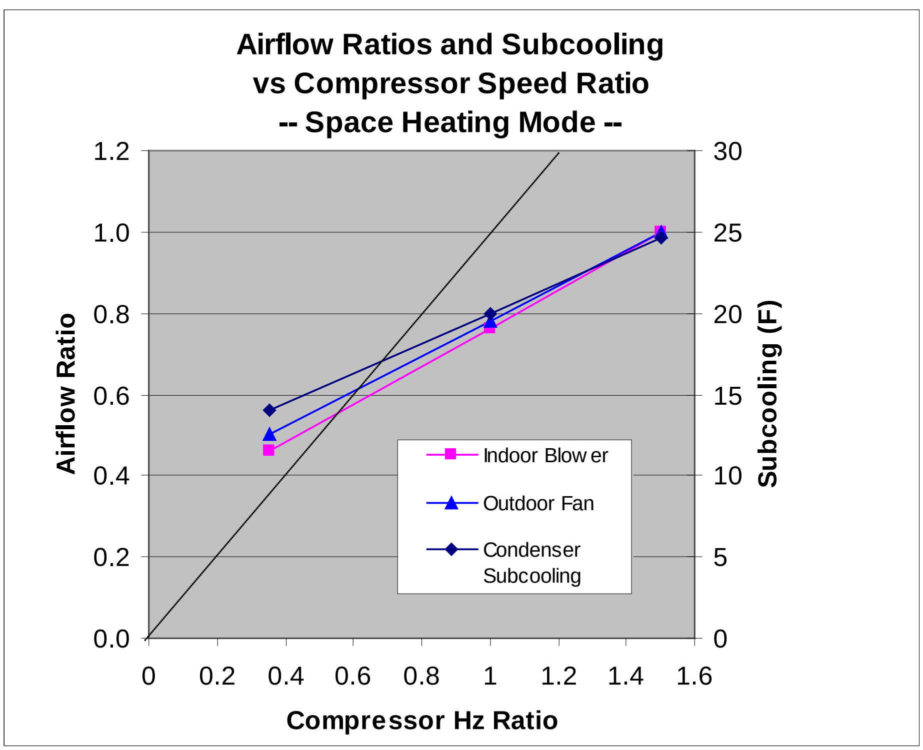

Fig. 5.8. Control parameters versus compressor speed ratio in the space heating mode (dotted gray line signifies 1:1 compressor speed to air flow ratio).

5.4.3 Target Speed Ratios and Refrigerant Flow Control vs. Ambient in Space Cooling and Heating Modes

In Fig. 5.9, the target speed ranges in cooling mode for all three modulating components are shown as a function of ambient temperature along with the specified condenser subcooling and compressor inlet superheat levels. This plot shows the speed ranges for an expected average cooling load matching with ambient. As the cooling load varies from the expected load relationship, the compressor speed will adjust to match the load seen by the thermostat, and the airflows and subcooling levels would be adjusted based on the revised compressor speed.
Fig. 5.10 shows a similar set of control values expected for an expected average heating load matching over the range of ambients. Again, depending on the actual heating load characteristics of a given building, the compressor speed would adjust to meet the actual load at a given ambient, and the other control parameters would be adjusted accordingly.

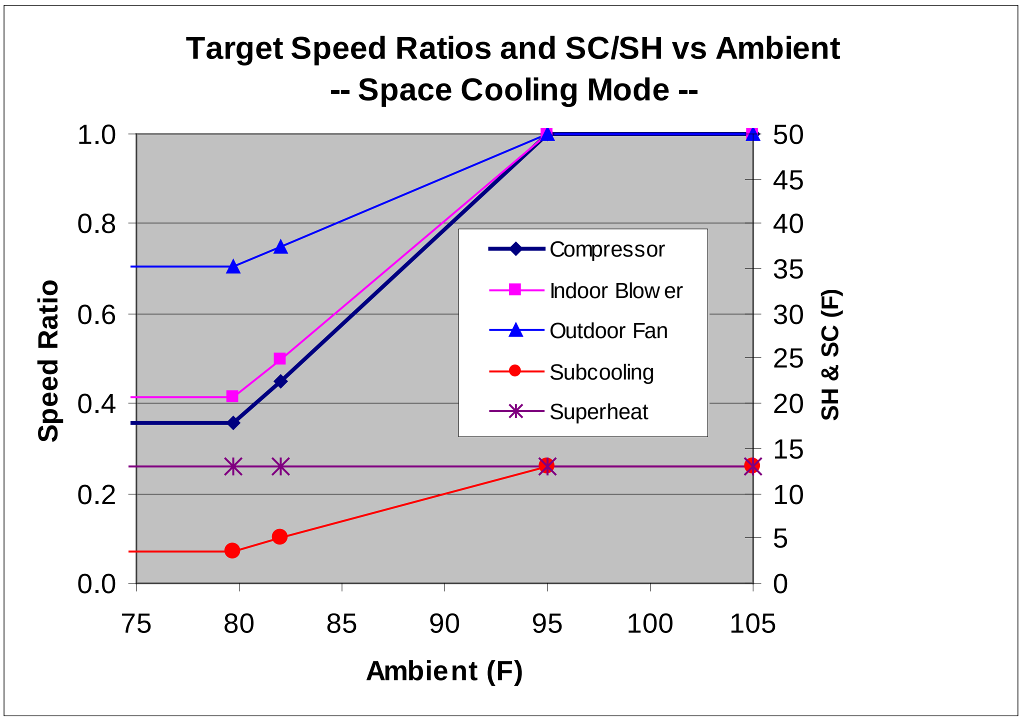

Fig. 5.9. Target speed ratios and refrigerant superheat (SH) and subcooling (SC) levels vs. ambient in the space cooling mode.

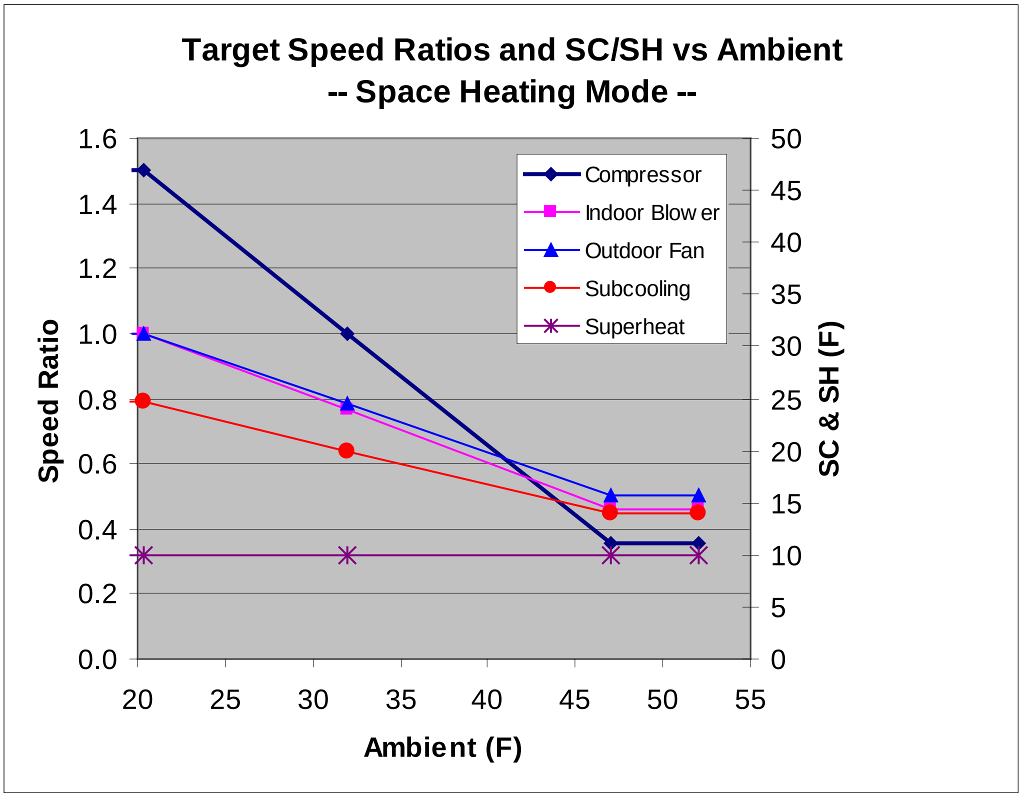

Fig. 5.10. Target speed ratios and refrigerant superheat (SH) and subcooling (SC) levels vs. ambient in the space heating mode.

5.4.4 Target Compressor Speed Ratios For Space Heating, Space Cooling, Water Heating, and Ventilation Cooling Modes vs. Ambient

In Fig. 5.11, the target compressor operating speed ratios vs. ambient temperature are summarized for space heating, space cooling, water heating, and ventilation cooling. Dedicated water heating is shown to operate at target maximum speed at 45°F and below, slowing to minimum speed at 65°F (from 45 to 90 Hz in this case). Expected operation range is from 40 to 80°F, as beyond these ambients, space conditioning is expected to take priority with water heating provided by the other combination modes as described earlier, including desuperheating, heat recovery, and combined space and water heating. The compressor cannot be oversped in water heating mode as much as in space heating because the condensing saturation temperatures must reach 130°F or higher to heat the water to the 120°F set point. In this mode, the outdoor airflow rate relationship is the same as in space heating while the condenser subcooling is slightly lower than in space heating, being controlled between 14°F at 45 Hz and 18°F at 90 Hz.

Target speed ranges for ventilation cooling are also shown in Fig. 5.11. Two curves are shown for different humidity removal requirements, from 100% relative humidity outdoor air to space-neutral and from an average outdoor humidity ratio to space-neutral. In this mode, the airflow across the indoor coil is fixed at the ventilation flow rate (e.g., 144 cfm for the timed 20-min duration) and the compressor speed is increased to provide more dehumidification as needed based on the indoor relative humidity sensor (from 28 to 37 Hz for the average humidity case and 29 to 64 Hz for the high humidity case). The outdoor airflow and the condenser subcooling are controlled as in space cooling in the ventilation cooling mode.

It should be noted that the low speed requirements for the variable-speed equipment are based primarily on the speeds needed in ventilation cooling to meet the dehumidification requirements at average humidity levels without removing too much moisture and requiring higher tempering heat output. Higher minimum ventilation cooling speeds will use more energy than needed to provide this function. The low speeds used in the space heating and cooling modes improve performance at light loads (and also rated efficiency numbers) but are used mainly because they are available from the ventilation cooling requirements. (Moderately higher minimum space cooling and heating speeds should still provide relatively high space conditioning efficiencies because the increase in compressor efficiency with speed would offset a portion of the HX and blower unloading gains realized at the current minimum speeds.)

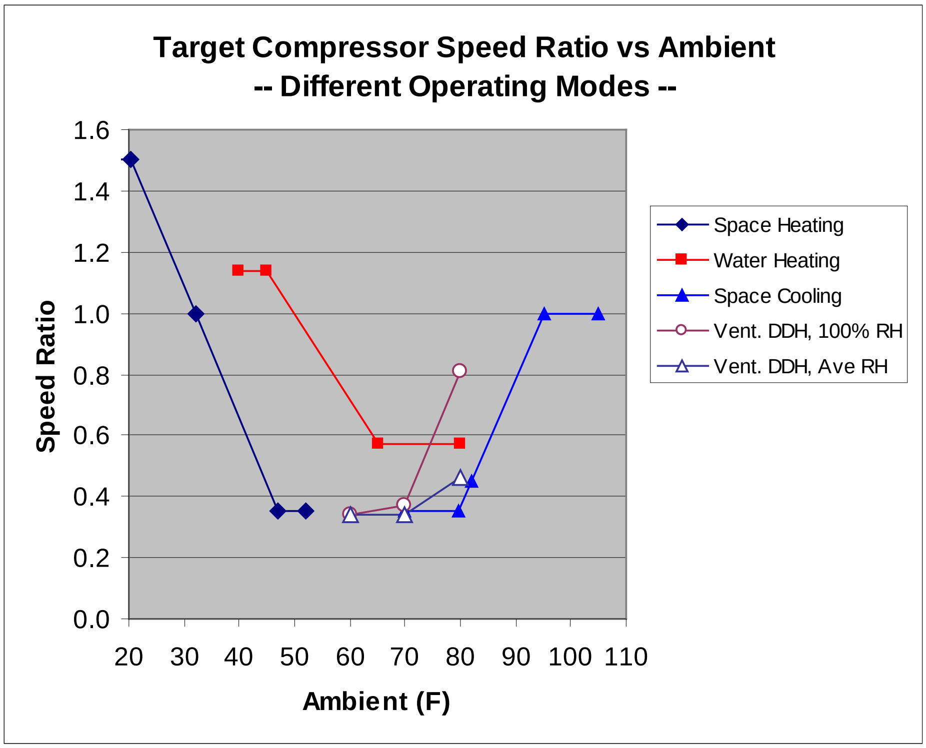

Fig. 5.11. Target compressor speed ratios for various operating modes versus ambient. (DDH = dedicated dehumidification; RH = relative humidity.)

These target speeds are to be used by the microprocessor as starting points for the various operating modes, to be adjusted by the thermostat controllers to meet the required indoor dry bulb temperature, humidity, or domestic hot water set points, with load following where possible for maximum efficiency. In the case of water heating, higher capacity output from the heat pump at the lower ambients is selected as the starting operation point rather than higher COP to avoid the need for resistance heat elements, which may be needed if the unit cannot keep up with hot water demand when space heating takes priority. However, if the thermostat determines that the water heating load is being met too quickly, the control logic will shift to lower speeds.

5.5 AS-IHP System Performance Trends for Selected Operating Modes

5.5.1 Target Air-Source IHP Space Heating and Cooling Performance vs. Ambient With Proposed Control Relationships for Load Tracking

In Fig. 5.12, the target performance of the AS-IHP is shown for space heating and cooling for the assumed load tracking behavior. This is where the compressor and indoor and outdoor fan speed ratios as well as the subcooling and superheat are assumed to follow the relationships given in figures 5.9 and 5.10 for space cooling and heating, respectively. The respective EERs are shown by the solid lines, and the delivered capacities are given by the dotted lines. The points where the trend lines change slope are where the minimum and maximum compressor speeds are reached and the system reverts to ambient trends similar to a single-speed unit, but at minimum and maximum speeds. It can be seen from this plot that the design cooling capacity at 95°F is just over 15,000 Btu/h or 1.25 tons. Similarly at the maximum overspeed operation in heating mode, a heating capacity of 13,600 Btu/h is reached at about 20°F ambient. Typically a single-speed heat pump has about the same heating capacity at 47°F as the design cooling capacity and then drops with ambient to a much lower capacity at 20°F, having a similar capacity to the variable-speed system shown here only at the design speed of 79 Hz (speed ratio of 1.0) at 32°F ambient. A constant outdoor relative humidity of 73% was assumed, as used in the ARI rating conditions.

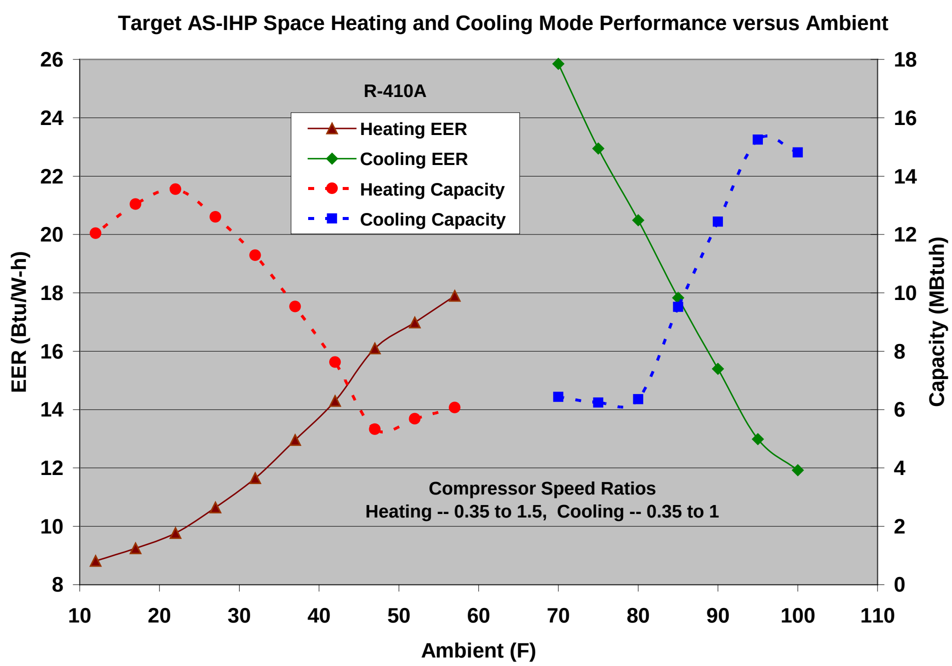

Fig. 5.12. Target air-source IHP space heating and cooling performance vs. ambient with proposed control relationships for load tracking.

5.5.2 Target AS-IHP Water Heating Performance vs. Ambient With Proposed Control Relationships

The target water heating performance is shown in Fig. 5.13 over the expected ambient range for this mode of operation where hot water is produced with outside air as the source with an assumed outdoor relative humidity of 73%. The assumed inlet water temperature was 108°F, which is consistent with the rating point used in the HydroTech 2000 performance ratings. (This is a representative temperature of the water leaving the bottom of the domestic water tank during a call for water heating that is to be supplied by the heat pump to reheat after a moderate draw or sufficient tank heat loss. At tank bottom temperatures much below 100°F, the control logic would likely be calling for resistance heat.) Here the water heating capacity ranges from about 14,000 Btu/h (equal to a 4.1-kW heating element) at 40°F ambient to just below 10,000 Btu/h at 65°F ambient. In the latter case, the lower heating output was selected to provide higher water heating COP, as there is no compelling need to heat the water faster at this ambient where there will be little if any call for coincident space cooling or heating. Accordingly, the delivered COPs for dedicated water heating range from about 2.8 at 45°F to 4.8 at 80°F, the highest ambient expected for outdoor source water heating. By way of comparison to heat pump water heater rating conditions of inlet evaporator air of 67.5°F ambient, 50% relative humidity, the water heating COP is 4.0 for the assumed inlet water temperature of 108°F. Note that in the cooling season, most water heating is expected to be done in heat recovery mode where both space cooling and hot water are delivered outputs and the effective COP including water heating is much higher.

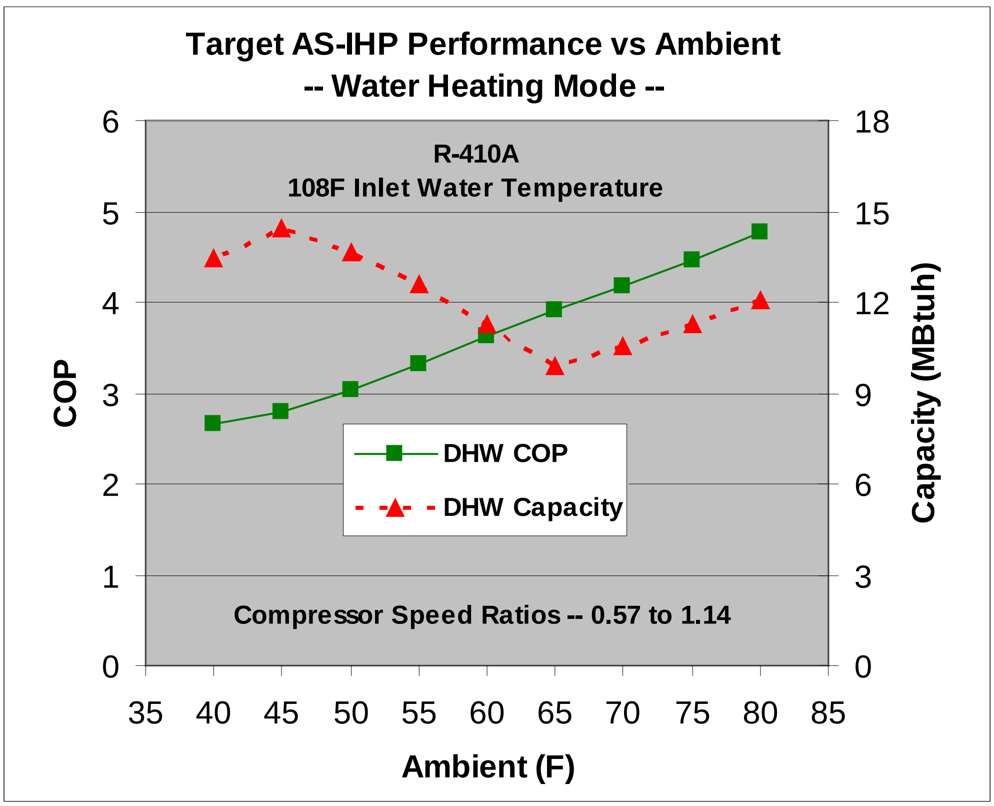

Fig. 5.13. Target air-source IHP water heating performance vs. ambient with proposed control relationships.

5.5.3 Target AS-IHP Ventilation Cooling Performance vs. Ambient With Proposed Control Relationships

Performance in the ventilation cooling mode from 70 to 80°F ambients is shown in Fig. 5.14, where an average outdoor humidity ratio of 0.0155 lbm water/lbm dry air is assumed with a constant ventilation flow rate of 144 cfm. This ventilation rate is that required over a 20-minute operation period to provide the required 48-cfm/hr ventilation rate. By using a 20-minute ventilation period rather than continuous, more reasonable minimum airflow rates and compressor speeds can be utilized. It is also assumed for ventilation air cooling, as opposed to ventilation air-only operation (section 4.5.10), that the indoor return air damper is closed and that only outdoor air is being circulated in the house.

The compressor speed is controlled in this case to provide a constant dehumidification rate and thereby supply air with space-neutral humidity with the outdoor coil airflow rate and subcooling adjusted according to compressor speed as in regular space cooling mode.

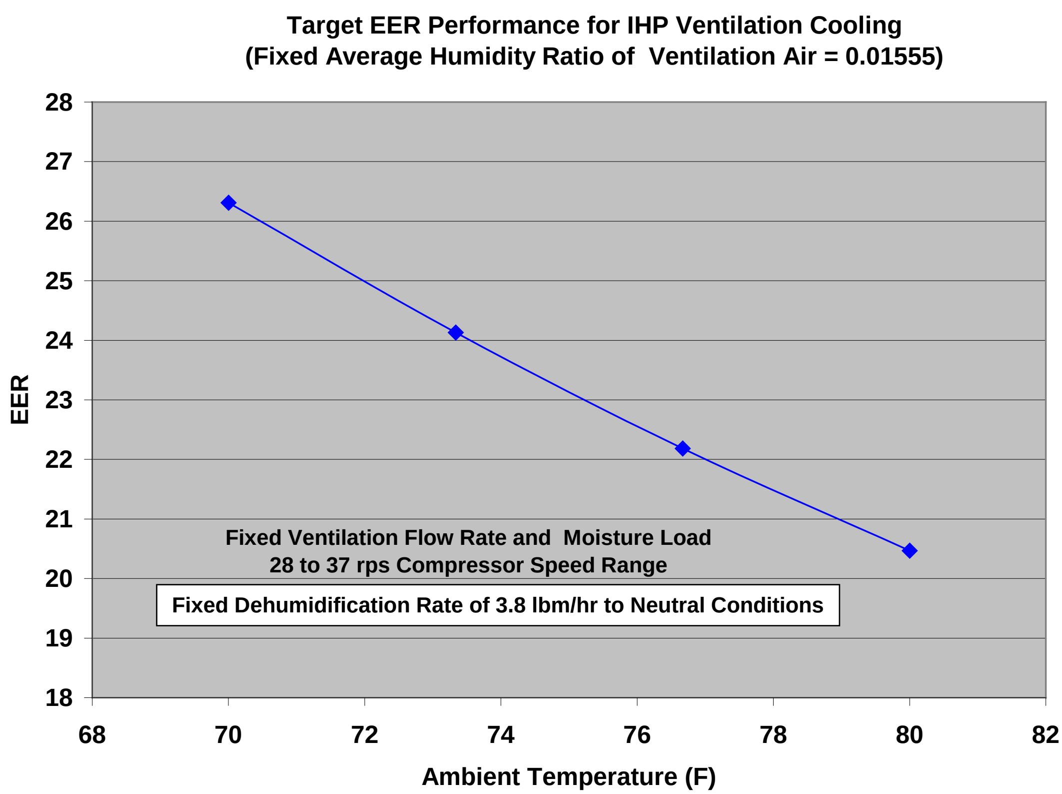

Fig. 5.14. Target AS-IHP ventilation cooling performance vs. ambient with proposed control relationships.

In Fig. 5.15, the delivered SHR is seen to range from 0.37 to 0.48 by directly working on the humidity ratio of the outdoor ventilation air without any dilution with indoor return air. This provides a high operating EER and even more importantly minimizes the tempering heat that is needed to offset the part of the accompanying sensible cooling that exceeds the required cooling load for the 20-minute operation period.

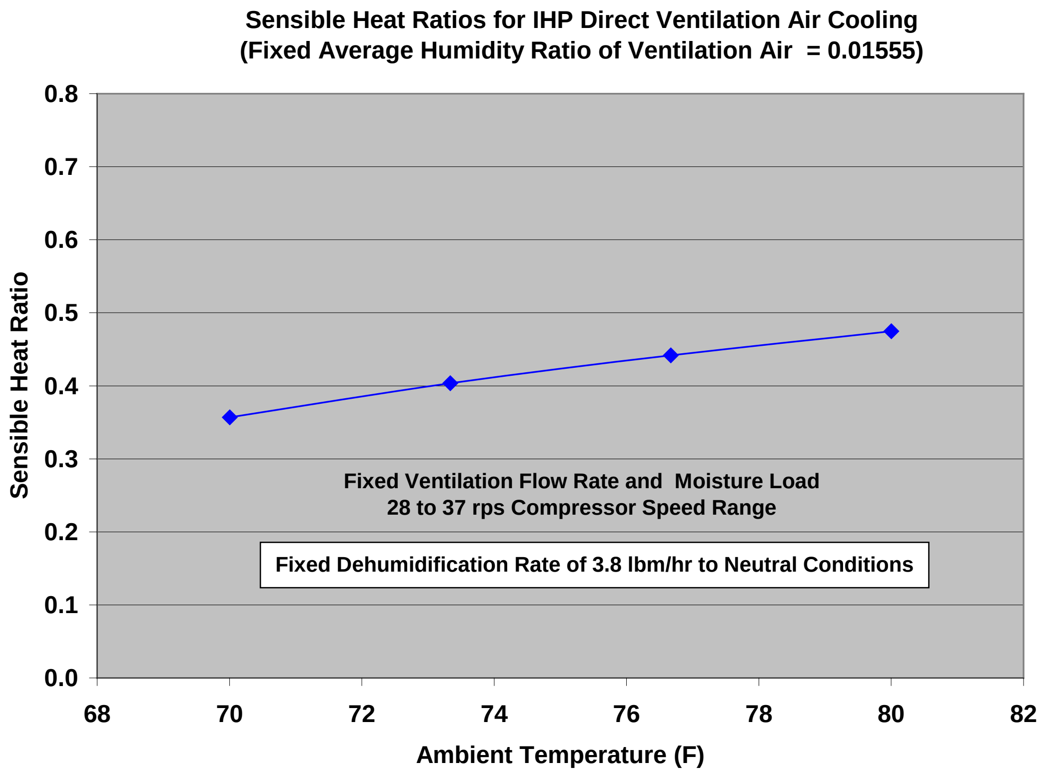

Fig. 5.15. Target AS-IHP ventilation cooling sensible heat ratio (SHR) vs. ambient with proposed control relationships.

5.6 Comparison to HydroTech 2000 Performance

From the example results provided, it can be seen that the AS-IHP is capable of high performance over a range of operating modes. This performance is maximized by variable-speed compressors that can maintain high efficiency reasonably well over the range of speed ratios required for load matching.

5.6.1 Comparison of Space Cooling Performance

Comparison of predicted AS-IHP performance to that of the earlier HydroTech 2000 variable-speed system (Carrier 1989c) shows significant performance improvements with the higher efficiency rotary compressors presently available

This is shown in the following figures for the current R-410A AS-IHP design as compared with the published HydroTech 2000 product performance data for their 2-ton design using R-22. The first comparison is for space cooling operation. Here the Carrier product was rated at low, intermediate, and high (design cooling) speeds (32, 40, and 55 Hz in the cooling mode for the 2-ton design). The intermediate speed is, by rating procedure convention, one-third of the way between the low and high speeds.

We evaluated the ORNL AS-IHP design for a similar set of speeds, although with the wider available speed range. The cooling mode speed ranges relative to rated design cooling speed were from 0.58 to 1 for the Carrier 2-ton design versus 0.35 to 1 for the ORNL AS-IHP.

Figure 5.16 shows the comparative capacities for the nominal 1.25 vs. 2-ton designs, with the ORNL design of similar relative speed given by the dashed lines with the same symbol. Here the capacity trends are seen to be similar for the three speed levels, with the nominal capacities of 1.25 and 2 tons, respectively, being obtained at high speed and 95°F ambient.

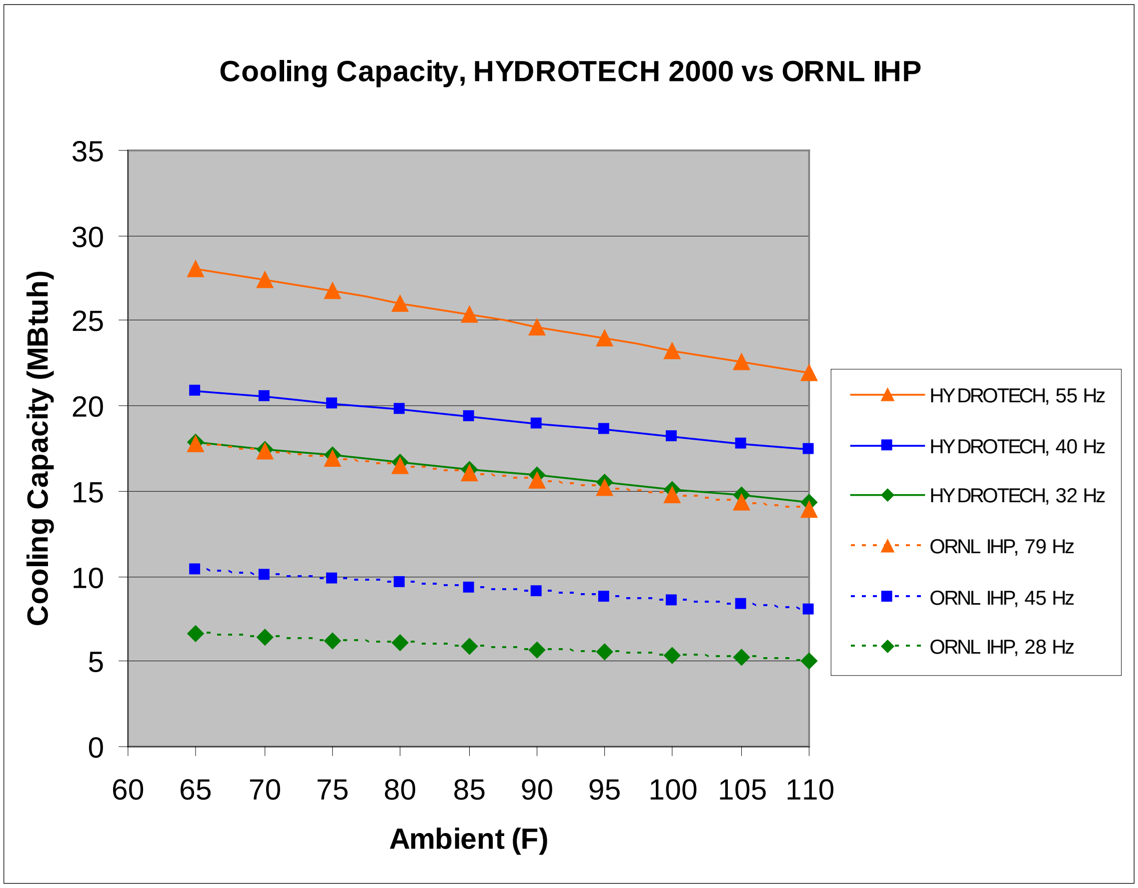

Fig. 5.16. Comparison of space cooling capacities for 1.25-ton ORNL AS-IHP vs. 2-ton HydroTech 2000 designs.

Next the comparative EER levels are shown for the AS-IHP versus the HydroTech 2000. Here the performance of the present design in Figure 5.17 is seen to be much higher than that of the Hydrotech, especially at the intermediate and low speeds where most of the operating hours occur. This higher predicted performance is due in part to the wider low-end speed modulation range and in part from the higher efficiency of the current variable-speed rotary compressors.

5.6.2 Comparison of Space Heating Performance

Next, comparisons are shown for the space heating mode. Here the 2-ton design HydroTech had a wider speed range than in cooling mode and was again rated at low, intermediate, and high heating mode speeds of 32, 46, and 73 Hz..

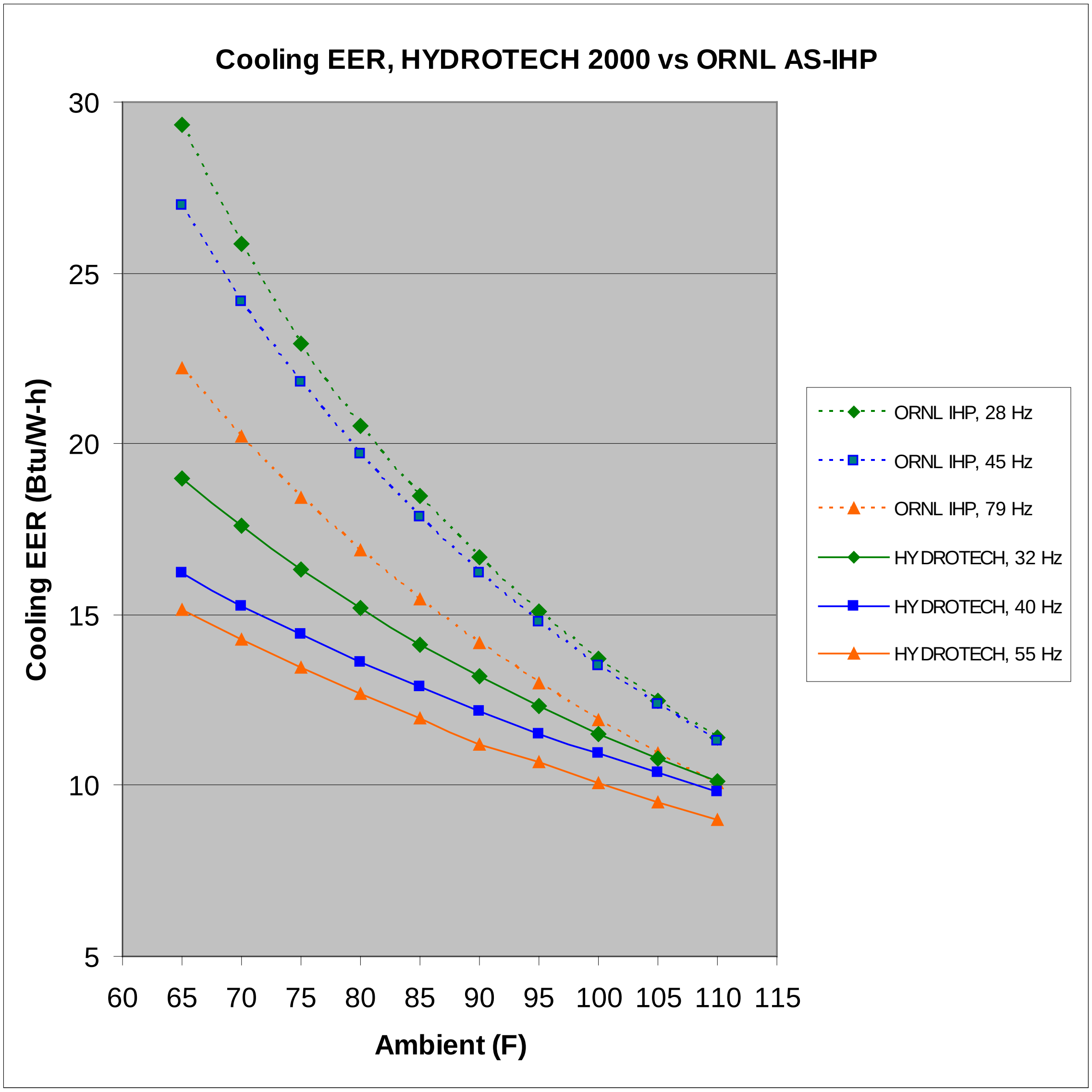

Fig. 5.17. Comparison of space cooling EERs for 1.25-ton ORNL AS-IHP vs. 2-ton HydroTech 2000 designs.

As in cooling mode, we evaluated the ORNL AS-IHP design for a similar set of speeds, but again with a wider available speed range. The heating mode speed range for the Carrier 2-ton design was from 0.58 to 1.33 vs. 0.35 to 1.5 for the ORNL AS-IHP.

In Figure 5.18, we show the comparative capacities for the nominal 1.25-ton vs. 2-ton designs. Here the capacity trends are seen to be similar for the three speed levels, except for the bend in the highest capacity HydroTech curve caused by the inclusion of integrated frosting/defrosting effects on the provided data.

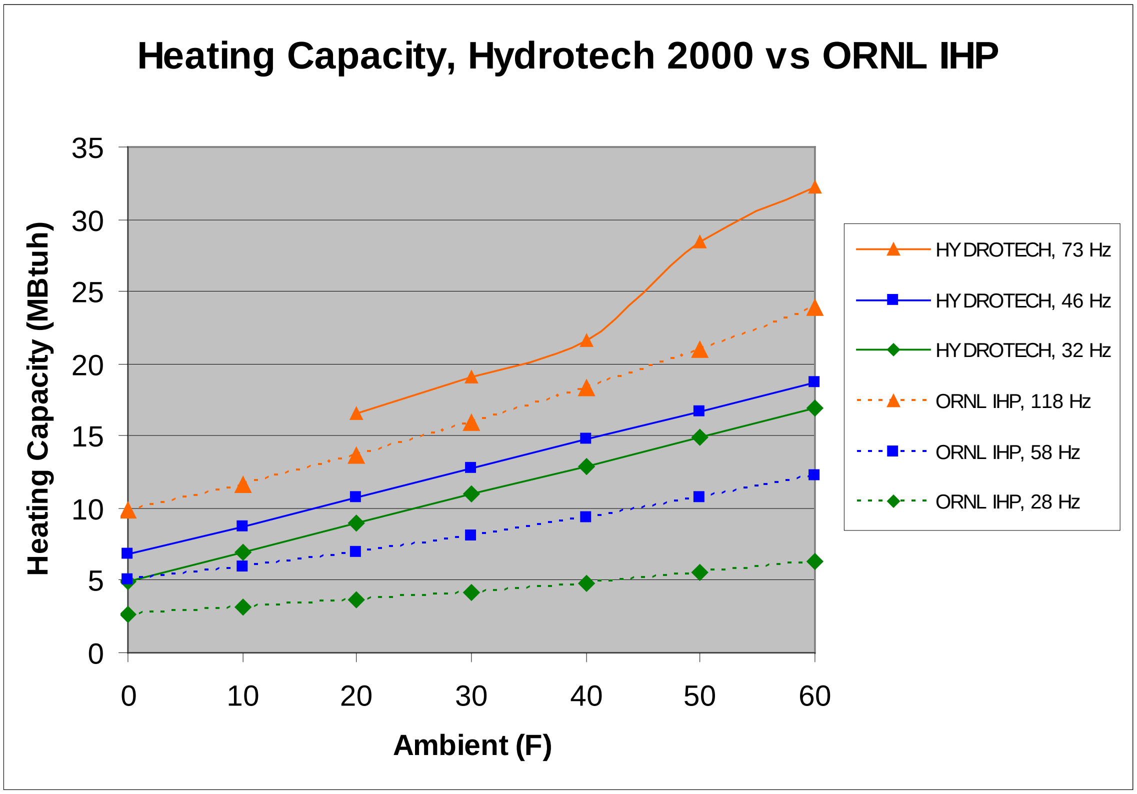

Fig. 5.18. Comparison of space heating capacities for 1.25-ton ORNL AS-IHP vs. 2-ton HydroTech 2000 designs.

Next the relative heating COP levels are shown for the AS-IHP versus the HydroTech 2000. Here the performance of the present design in Fig. 5.19 is seen to be much higher than that of the Hydrotech, including the intermediate and low speeds where the bulk of the operating hours occur. At the 47°F heating rating point, a 4.7 COP is predicted for the AS-IHP as compared to a COP of 3.5 for the Carrier product. At the low ambient rating point of 17°F, the AS-IHP has a 2.7 COP vs. 2.3 for the Hydrotech, while operating at a higher speed and relative HX loading. Again the wider low-end speed modulation range and higher efficiency of the current variable-speed rotary compressors are believed to be the major contributors here.

5.6.3 Comparison of Dedicated Water Heating Performance

The performance of the AS-IHP when exclusively heating water using the outdoor air source is shown for a range of ambients in the next two figures. An entering water temperature (EWT) of 108°F from the domestic water heater was used for this comparison as this was the only EWT for which the HydroTech water heating performance was published. Performance for a range of compressor speeds was not provided in the Carrier water heating data, but rather for a programmed speed for each ambient ranging from 32 to 70 Hz. Accordingly, our comparisons are made for an estimated range of corresponding speeds.

In Fig. 5.20, the published HydroTech 2000 dedicated water heating capacity curve for 20 to 60°F ambient temperature is compared to AS-IHP water heating performance when controlled as discussed in section 4.4.2.

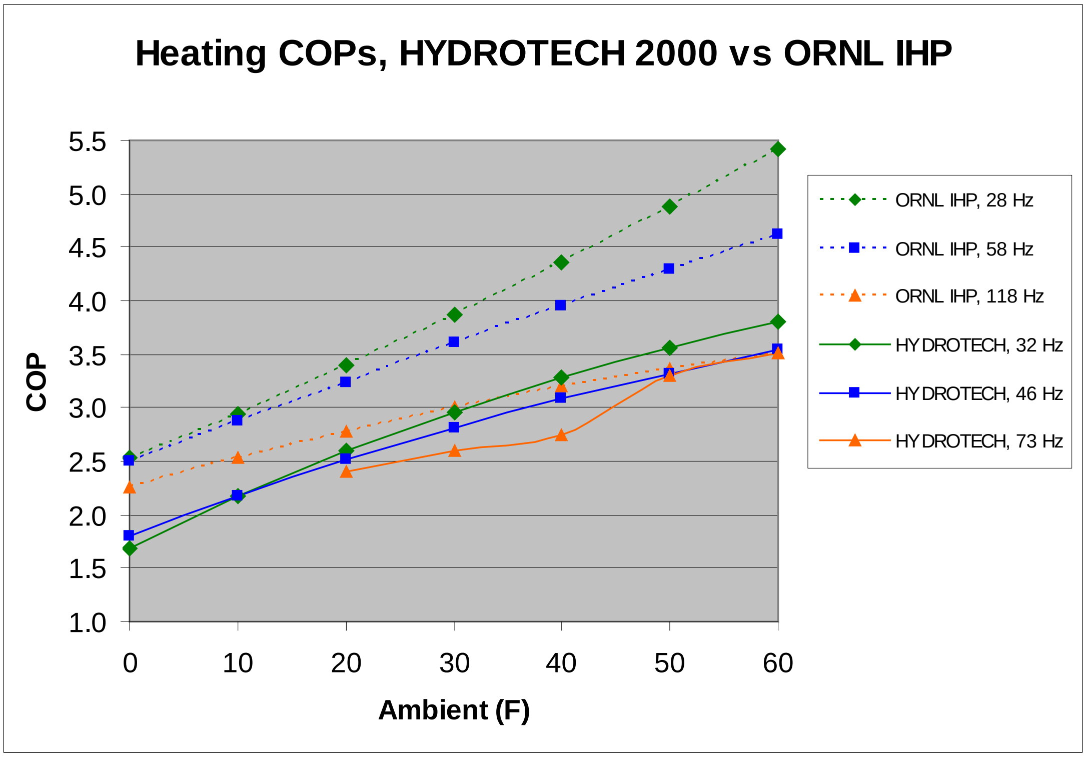

Fig. 5.19. Comparison of space heating COPs for 1.25-ton ORNL AS-IHP vs. 2-ton HydroTech 2000 designs.

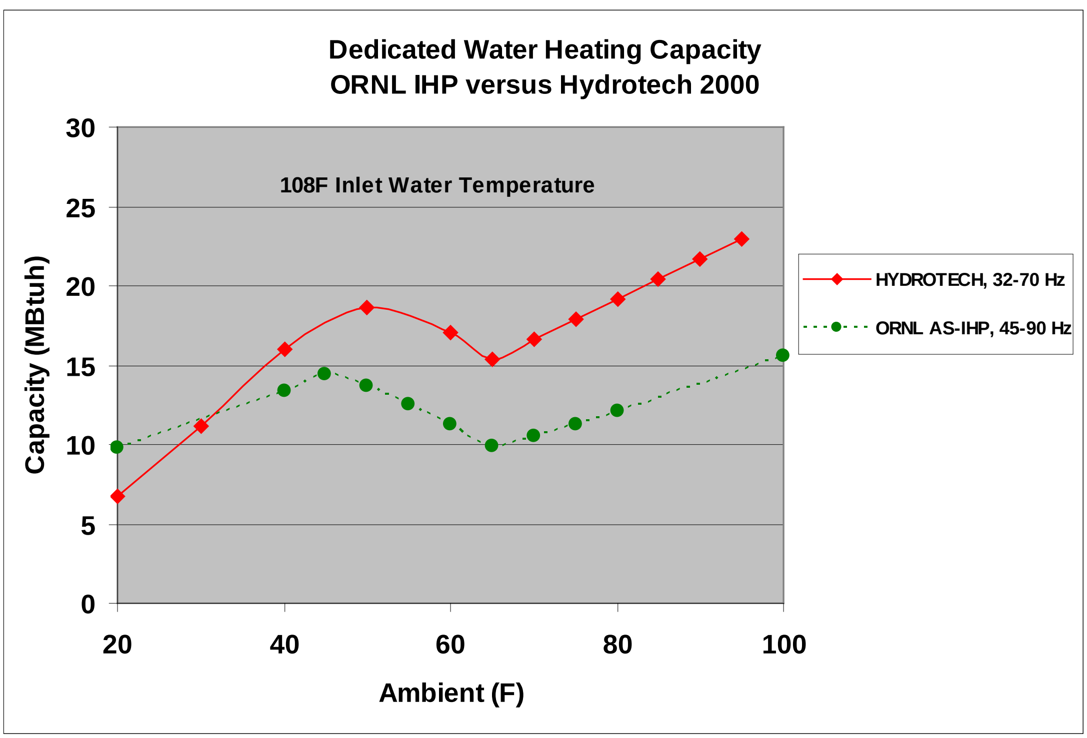

Fig. 5.20. Comparison of dedicated water heating capacities for 1.25-ton ORNL AS-IHP vs. 2-ton HydroTech 2000 design.

From observing the HydroTech capacity trends with ambient, it is estimated that the compressor speed was at the 70 Hz maximum water heating speed around 50°F and below, and that the speed was reduced as the ambient increased from 50 to 65°F, where the minimum speed of 32 Hz was reached and maintained for higher ambients. In comparison, the currently recommended water heating speeds for the AS-IHP are 90 Hz at 45°F and below, decreasing to a minimum of 45 Hz at 65°F. The capacity trends of the two systems are seen to be similar, but with a faster drop-off in capacity for the HydroTech unit below 40°F. This is most likely due to the poorer volumetric efficiency of the reciprocating compressor from the higher clearance volume effects as compared to the rotary type used in the AS-IHP.

The comparative water heating COPs are shown in Figure 5.21. At 50°F, the COP advantage is the narrowest for the AS-IHP system at 37%, increasing to between 47 and 53% higher between 60 and 80°F. Below 50°F, the COP of the current design increases from 40% higher at 40°F to 123% higher at 20°F, although our current control approach does not operate a dedicated mode at this low an ambient.

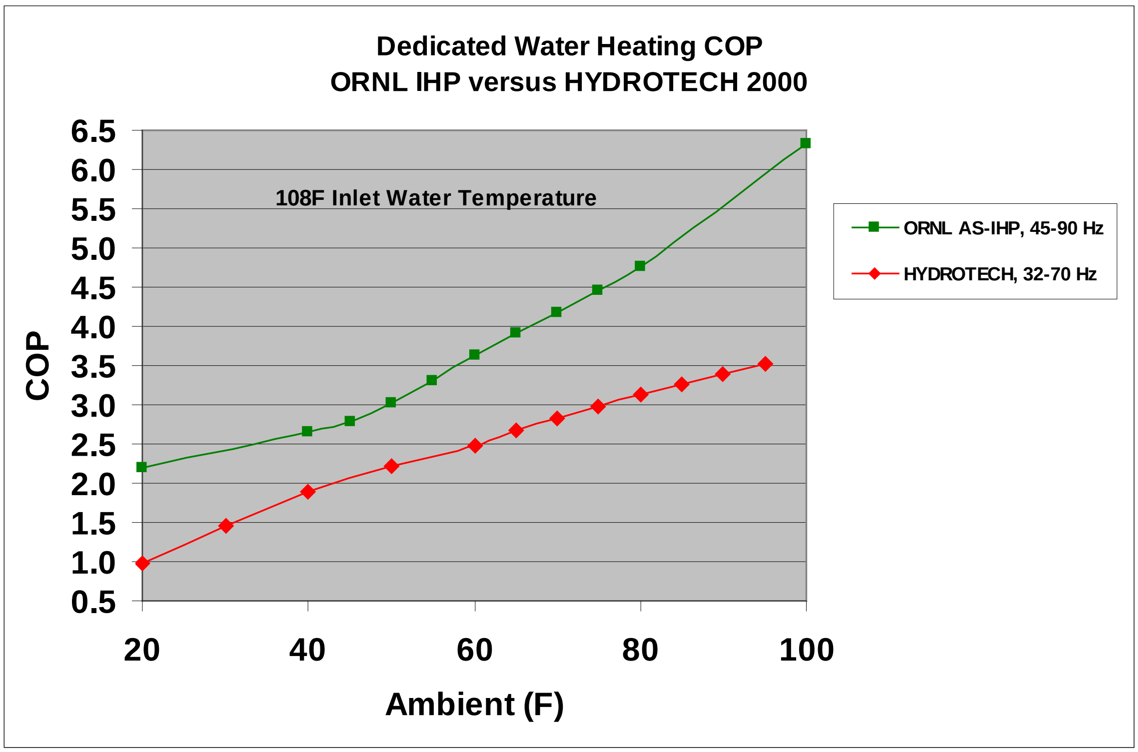

Fig. 5.21. Comparison of dedicated water heating COPs for 1.25-ton ORNL AS-IHP vs. 2-ton HydroTech 2000 design.

5.6.4 Comparison of Combined Space Cooling and Water Heating Performance

For the case of combined space cooling and heat recovery water heating, the HydroTech design rejects heat from both the water-to-refrigerant condenser and the outdoor condenser operating in series. We surmise that this was done to limit the maximum condensing temperature and possibly compressor torque requirements for the selected size of water-to-refrigerant HX. As such, combined mode cooling performance is a function of ambient temperature.

In contrast, the proposed AS-IHP design employs full heat recovery and so performance is independent of ambient temperature at a fixed compressor speed. As a result, the combined cooling and water heating performance is somewhat higher than for the HydroTech design, as shown in Figs. 5.22 – 5.24. As before, the water heating performance is calculated for a constant inlet water temperature to the water-to-refrigerant HXs of 108°F.

The delivered space cooling and water heating outputs for the two designs are compared in Fig. 5.22. The 2-ton HydroTech unit provides a nearly constant 7000 Btu/h of water heating while that of the AS-IHP increases along with unit cooling output from about 6000 Btu/h to over 17,000 Btu/h at the design cooling condition at 95°F ambient.

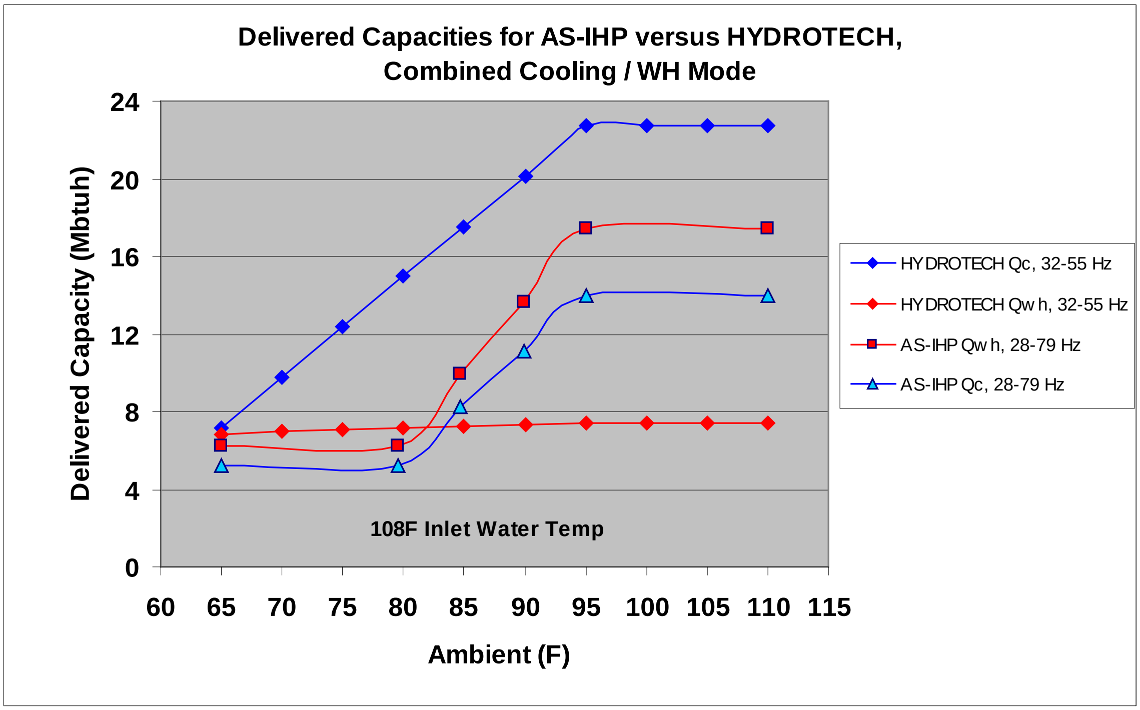

Figure 5.22. Space cooling and heat rejection capacity vs. ambient for rated inlet water temperature.

In Figure 5.23, the cooling-only EERs of the two systems while in this combined output mode are compared. Here the EER of the AS-IHP ranges from 11 at the lower speeds to around 10 at the design cooling condition, as compared to a steadily increasing EER from 10 to over 13 for the HydroTech at design cooling. This is because the full condensing AS-IHP has an increasing water-to-refrigerant HX loading with speed/ambient and the single 108°F heat sink temperature, while the dual condensers in the HydroTech case shift more of the heat rejection load to the cooler outdoor sink as the compressor speed is increased. The result is that the AS-IHP operates at an increasingly higher condensing temperature with ambient.

However, when one compares the combined EERs of each system, calculated as the sum of the useful cooling and water heating output divided by the input power, the AS-IHP system shows much higher performance. In Figure 5.24, the combined EERs for the AS-IHP range from 24 to 22 as the ambient increases vs. 10 to 13.5 for the HydroTech 2000. This is due to the much higher water heating output from the full waste heat recovery operation for the present AS-IHP design, which more than offsets the higher condensing temperatures required.

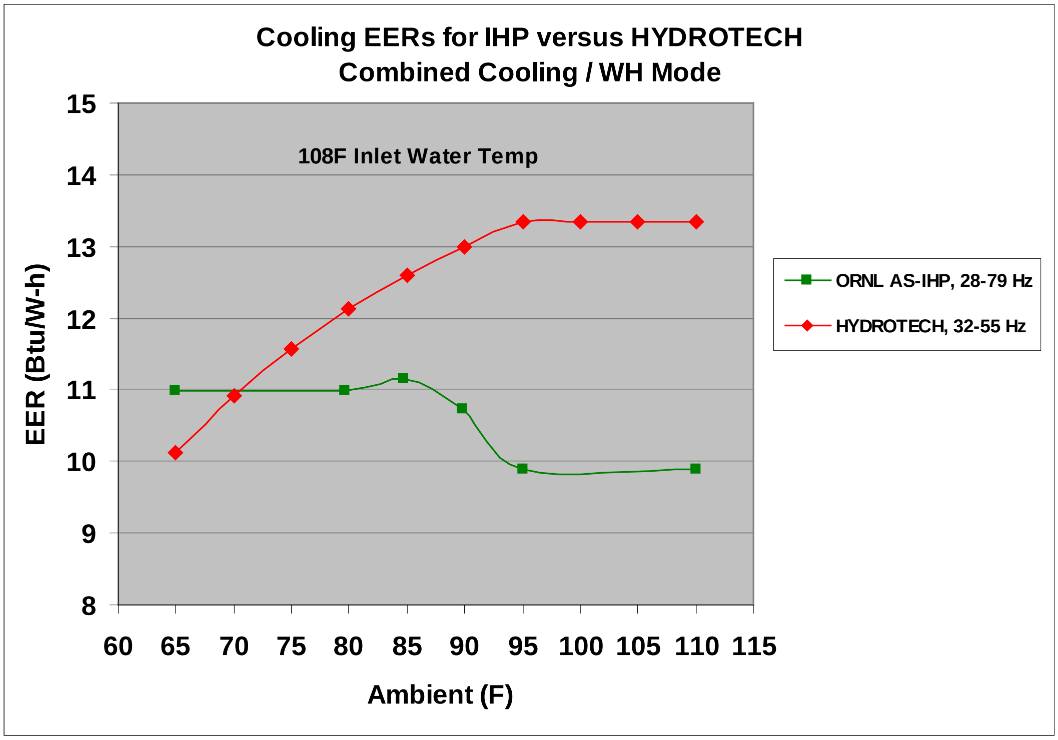

Figure 5.23. Space cooling EER va. ambient for rated inlet water temperature.

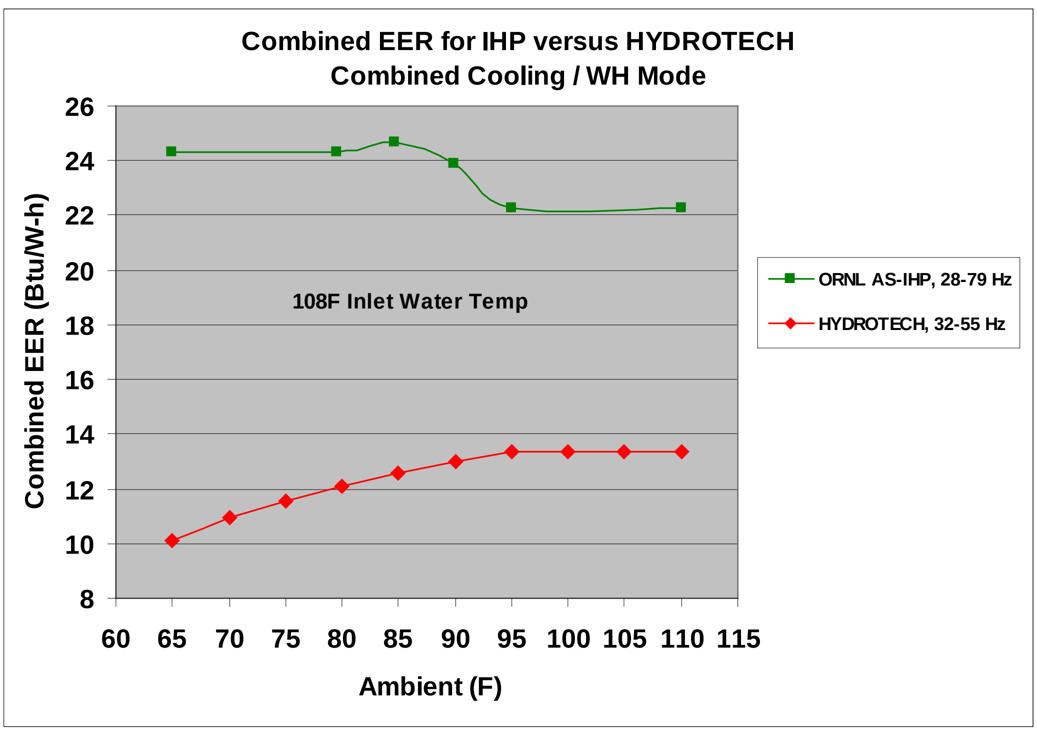

Figure 5.24. EER for combined cooling / water heating outputs vs. ambient for rated inlet water temperature.

6. INTEGRATION OF HPDM ANALYSIS INTO TRNSYS SIMULATIONS OF ANNUAL ENERGY USE AND SAVINGS OF AS-IHPs IN ZEHs

6.1 Development of Energy Savings Estimates for AS-IHPs

Once validated steady-state estimates of integrated equipment performance are available, the next challenge is to estimate the energy use of such equipment over a season or year of operation in a specified house and climate. This step is somewhat more involved for integrated equipment than for separated units in that it requires a coupling with the usage history and thermal state of the domestic water heater. In addition, there may be competing calls for different modes of operation, for which the control logic must determine the appropriate calls. An appropriate way to properly account for these interactions is to perform a time-series-based calculation based on a suitable starting point for a year of operation. For conventional HVAC systems without strong coupling to the domestic hot water (DHW) system, this is done on an hourly basis with energy use simulation codes such as DOE-2. However, in the case of an integrated heat pump such as is under development here, the use of one HVAC system to provide multiple outputs requires a sub-hourly analysis to most accurately account for the various interactions, the competing operating modes, and representative inlet conditions that will be seen simultaneously by the water-to-refrigerant DHW and source HXs while heating water.

6.2 Simplified Performance Estimates

6.2.1 Initial Performance Calculations

When we first had a validated heat pump simulation for the major operating modes at the end of FY05, such a time-series-based analysis tool was not available to us. Accordingly, our first estimates were based on separate estimates of the seasonal space cooling and space and water heating energy use. System simulations using the DOE HPDM for the major operating modes were used to estimate the equipment seasonal space conditioning and water heating COPs under expected average operating conditions for each space conditioning and/or water heating function. In the space cooling and heating modes, the binned DOE rating procedures were used with the rating point data at different speeds and appropriate ambients provided by calibrated HPDM simulations. For the water heating energy factor (EF) calculations, estimates were made of the summer, winter, and shoulder month average energy factors using the HPDM water heating mode calculations based on estimated average inlet water temperatures. Estimated contributions from desuperheater operation and required resistance water heating in the winter months were also included. The estimated SEER, HSPF, and overall water heating energy factor by this initial approach were 17.9, 11.3, and 2.93, respectively, and yearly energy savings were estimated for an 1800-ft^2 ZEH in Atlanta.

6.2.2 Initial Energy Savings Estimates

The total electric energy requirements needed to meet the ZEH loads were determined by applying the calculated SEER, HSPF, and EF values to the appropriate loads for space cooling, space heating, and water heating. This energy use was compared with a baseline HVAC/WH system consisting of a 13-SEER, 7.7-HSPF heat pump and 0.90-EF water heater, and with a state-of-the-art heat pump and electric water heater system. Results of this analysis and an energy savings breakdown are shown in Table 6.1.

Table 6.1. Initial performance comparison of IHP and state-of-the-art equipment

Loads (1800 ft^2 ZEH in Atlanta, from TRNSYS)		Equipment							
		Baseline		State-of-the-Art Electric			IHP		
Source	kWh	Efficiency	Energy use (kWh)	Efficiency	Energy use (kWh)	Energy reduction compared to baseline	Efficiency	Energy use (kWh)	Energy reduction compared to baseline
Space heating	3920	7.7 HSPF	1737	9.8 HSPF	1365	21.4%	11.3 HSPF	1183	31.9%
Space cooling	2313	13 SEER	607.1	20 SEER	394.6	35.0%	17.9 SEER	440.7	27.4%
Water heating (total)	3019	0.90 EF	3354	0.95 EF	3178	5.3%	2.93 EF	1032	69.2%
Water heating (heating season)	1474	0.90 EF	1637	0.95 EF	1552	5.3%	2.11 EF	698.5	57.4%
Water heating (cooling season)	801	0.90 EF	890.3	0.95 EF	843.4	5.3%	5.37 EF	149.1	83.3%
Water heating (shoulder seasons)	744	0.90 EF	826.5	0.95 EF	783.0	5.3%	4.04 EF	184.2	77.7%

The results predicted that the IHP could save 53.4% in yearly energy use in Atlanta compared to the baseline 2006 minimum efficiency HP and electric water heater. In contrast, the state-of-the-art 20-SEER HP and 0.95-EF electric water heating system, with no heat-pump-supplied water heating, saved only 13.4% relative to the minimum efficiency baseline.

It should be noted (in relation to later house loads reported in this section) that the loads in Table 6.1 were computed by TRNSYS assuming maximum utilization of window opening to minimize cooling loads without consideration of the impacts on indoor space humidity. These results did not include active humidity control and so the energy use effects of enhanced, dedicated, and ventilation air dehumidification modes were not included.

6.3 Development of Time-Series-Based Energy Savings Calculations

6.3.1 Direct HPDM Call Implementation

In FY06, much improved calculations of the yearly energy use were developed by linking the HPDM with TRNSYS, a time-series-dependent simulation model capable of determining the energy use of building cooling and heating equipment as applied to a defined house on a sub-hourly basis. This required an extensive effort to couple the HPDM to TRNSYS in a fully consistent manner so that the outputs of the TRNSYS from modeling the time-dependent indoor space water heater conditions would become inputs to the HPDM. The HPDM output conditions of the indoor air and water leaving the equipment HXs were then also coupled back to the

TRNSYS house and DHW modules to update their operating states. Further details of the house and controls modeling are described by Baxter (2007).

From the summer of FY06 into the fall of FY07, indoor humidity control was also added to our TRNSYS-based analysis capability, first for stand-alone dehumidifiers and then for the AS-IHP by including the control logic for enhanced, dedicated, and ventilation air dehumidification. The direct-mode HPDM coupling along with the indoor humidity control was first used for an initial business case analysis for the AS-IHP, as reported by Baxter (2007).

The HPDM modules within TRNSYS were set up to call one of three heat pump description data files depending on which of three basic modes of operation were active as determined by the temperature and relative humidity thermostat calls: space conditioning, space cooling and water heating, and dedicated water heating. The desired settings of air- and water-flows, subcooling and superheat as a function of compressor speed, and mode as discussed in Chapter 4 were defined in an IHP system control routine. The sub-modes of space cooling operation for enhanced, dedicated, or ventilation air dehumidification were activated by TRNSYS as needed by the temperature, relative humidity, and/or outdoor humidity thermostat calls.

This direct HPDM/TRNSYS coupling provided for the first time the ability to simulate in a sub-hourly analysis the annual performance of a multi-function, multi-mode, integrated heat pump without having to provide a detailed set of curve-fitted equations representing the system performance over a range of conditions and airflows in multiple operating modes. It also provided the ability to modify the components and/or the compressor speed vs. HX flow controls of the system and evaluate the annual energy use implications without having to provide new sets of performance equations.

The observed drawbacks of the initial direct HPDM/TRNSYS coupling after use for the initial business case analyses were the following.

1) Increase in computational time. Although an HPDM call typically executes in a fraction of a second, with the TRNSYS model running on a 3-minute time step, it usually required more than 160,000 calls to the HPDM for a yearly simulation. This resulted in an increase in the total run time by a factor of 4 to 5.

2) Decrease in model robustness. With more than 100,000 calls to an iterative solution model such as the HPDM, the odds of having the solution crash once in the yearly simulation were difficult to reduce to zero. While this was not a major issue for air-source analysis where one-year runs would need to be restarted occasionally, for our companion ground-coupled IHP analyses where 10- or 20-year runs were occasionally needed (to confirm proper ground HX sizing by looking at the long-term effect on ground temperature), these occasional solution glitches became more problematical.

3) IHP system design control logic hard-coded into a TRNSYS module. The need to incorporate the speed control equations shown in Chapter 4 into TRNSYS code required code recompiling for each design change and so made it somewhat inconvenient to change the system internal design controls.

6.3.2 Map-Based IHP Modeling Implementation

The HPDM was directly linked to TRNSYS for the needed assessment capabilities for two primary reasons. First, it was seen as the most immediately achievable way to obtain this capability to model multi-function, multi-mode performance. Second, direct call provided the ability to experiment with design changes without having to regenerate a new set of performance curves, an exercise that was particularly onerous, time-consuming, and error prone. However, mainly because of the model run-time and robustness issues noted above, an alternative way to provide this capability while minimizing or eliminating the drawbacks was considered. What emerged from this rethinking process was a map-based approach combined with multi-dimensional interpolation.

By tabulating all the operating modes and range of operating conditions needed to represent the envelopes of IHP performance, we determined that it was not unreasonable to generate performance arrays that would encompass the full range of possible operation. While the number of required calls to obtain close interpolations of IHP performance in all modes was not small, at a few thousand, this was much smaller than the more than 160,000 calls needed for the direct HPDM approach. As importantly, once these runs were completed successfully once, the data array could be saved and reused by recall at the outset of successive runs for different climates, or houses, or even control logic strategies.

In mid-FY07, new TRNSYS modules were prepared to generate a full set of AS-IHP performance maps at the outset of the first TRNSYS run with a new AS-IHP design. These performance maps were set up to save all output values (presently numbering 30) that might be needed for linkage to the rest of the TRNSYS equipment and house models. (Because desuperheater performance was dependent on not only time-dependent inlet water-to-refrigerant HX temperatures from the DHW tank but also outlet conditions from the heat pump model, this component model was handled outside of the HPDM calls using interpolated output HPDM values along with current exiting DHW tank temperatures.)

Once the performance map array was generated by running the HPDM model in map generation mode, and written to disk for later reuse, the HPDM module would perform similarly to a direct HPDM call case. However, in this case, it is by performing multi-parameter interpolations (of three to four independent parameters) of heat pump performance for the active operation mode for the current 3-minute time step. Such interpolations are much faster than an HPDM call and so the computational slowdown is eliminated save for an initial IHP performance map generation step. This initial set of calls typically takes 5 to 6 minutes and needs be done only once for each IHP design.

Compressor speed values and step sizes are made consistent with the presently allowed five or six speed steps available for the space conditioning thermostat controls and four speed steps used in the TRNSYS controller for the water heating thermostat speed control logic. This eliminates compressor speed interpolation error, which is potentially the largest source of error with a limited number of speed steps, from the current implementation.

To specify the heat pump configurations needed for the IHP performance maps, we use five heat pump input data files: for space cooling, space heating, space cooling with heat recovery water heating, outdoor source water heating, and ventilation air cooling.

To define the range of parametrics and the heat pump control design, we use eight parametric input control files, one for each possible mode of heat pump operation. The parametrics needed for the performance mapping, in addition to compressor speed, are indoor temperature and relative humidity and outdoor temperature for space cooling modes and outdoor temperature and relative humidity and indoor temperature for space heating. For the water heating modes, the range of possible inlet water temperatures is used in place of outdoor temperature for the combined space cooling mode. For the outdoor source water heating, the inlet water temperature parametric replaces that of the indoor air temperature. In the ventilation air cooling mode, only outdoor temperature and relative humidity are needed.

The parametrics data files were set up consistent with existing parametric capability of the HPDM (Rice 1991). The main change was to extend the number of possible parametrics from two to five and to enable the parametric control input to operate properly with up to five possible independent variables changing. The capability to handle five-variable parametrics was included to accommodate possible future needs to handle split condensers rejecting heat from two sources. In such cases, parametrics for two sink temperature ranges could be needed.

Because the parametrics files are input to the HPDM/TRNSYS analysis tool, all the mapping and heat pump system design control information are now modifiable outside of the source code. This structure resolved the remaining "lesson learned" as noted earlier in this section from the direct call/control experiences. By having all of the heat pump equipment design specifics external to the TRNSYS code, it is much easier to modify and track the current design approach being used.

Further, because the performance map generation process is easily done as a one-time computation once the input heat pump data and parametric control files are prepared, a change of heat pump designs is much less time consuming than would be the case if equipment performance curve fits were required to be generated and input to the program. This meets a long-standing need for a way to conveniently yet accurately incorporate advanced heat pump designs with two or more operating modes into the more detailed hourly and sub-hourly whole-house energy use simulations.

To test the accuracy of the performance mapping approach, we compared the energy use results for each operating mode with the direct HPDM call approach. This was possible because both options were preserved in the current TRNSYS implementation. The comparison proved useful as well in debugging the mapping and interpolation implementation. In the end, we found close agreement between the two approaches with the mapping approach having only a minimal increase in run time (using a saved performance map) compared with a baseline case using curve-fitted performance equations.

The map-based interpolation approach was used with the new system design described in Section 5 for the wider-range R-410A rotary compressor. The results of this analysis for an 1800-ft^2 ZEH in five climates are described in the following section.

6.4 TRNSYS-Based Systems Energy Consumption Analyses — AS-IHP Using R-410A

TRNSYS capabilities were used to simulate not only the annual performance of the current AS-IHP design but also for a suitable suite of baseline equipment for use in determining the potential energy savings of the AS-IHP in a ZEH providing the same energy services.

6.4.1 Baseline HVAC/WH/DH/H System

A standard split-system (separate indoor and outdoor sections), air-to-air heat pump provides space heating and cooling under control of a central thermostat that senses indoor space temperature. It also provides dehumidification when operating in space cooling mode but does not separately control space humidity. Rated system efficiencies were set at the DOE minimum required levels in effect for 2006 (SEER 13 and HSPF 7.7). Water heating is provided using a standard 50-gallon-capacity electric storage water heater with EF set at the current DOE-minimum requirement (EF = 0.90) for this size. Ventilation meeting the requirements of ASHRAE Standard 62.2-2004 (ASHRAE 2004) is provided using a central exhaust fan. A separate dehumidifier is included as well to meet house dehumidification needs during times when the central heat pump is not running to provide space cooling.

Dehumidifier location, sizing, and efficiency level. Rudd et al. (2005) indicates that perhaps the most cost-effective approach for adding separate dehumidification capability to a house is to locate a stand-alone dehumidifier in the conditioned space, preferably in close proximity to the main HVAC system return air grill. That is the approach adopted in the present analysis. A manufacturer of typical stand-alone dehumidifiers, Heat Controller, includes a table on their Web site that suggests a 30-50 pint/day (7-12 L/d) capacity would be sufficient for a 2000-ft^2 house (http://www.heatcontroller.com/products/pdf/dehumidbroch.pdf). A 40-pt/d size was chosen and this proved to be adequate for the ZEH in all locations. In this case "adequate" was taken to mean that indoor relative humidity (RH) levels would exceed 60% for no more than about 1 – 2% of the year. The 60% criterion matches that used by Rudd et al. (2005) in their study. Other studies use 65%, including a recent one by Witte and Henninger (2006) for ASHRAE that evaluated humidity control capability of various unitary system designs. For the cooling set point of 76°F used in our analyses, ASHRAE's thermal comfort standard indicates a maximum acceptable RH of about 65% for spaces with activity levels typical of offices (Figure 5.2.1.1 in ASHRAE Standard 55-2004).

There is currently no DOE-mandated minimum efficiency value for residential dehumidifiers. However, amendments to the Energy Policy and Conservation Act of 1975 included in the Energy Policy Act of 2005, P.L. 109-58, expanded DOE's energy conservation program to include certain commercial equipment and residential products, including dehumidifiers. In compliance with this directive, DOE/BT has recently specified a default minimum dehumidifier energy factor (EF_d) for 40-pt/d dehumidifiers of 1.3 L/kWh effective 2007 and a default minimum of 1.4 L/kWh effective 2012 (DOE/BT 2006). According to comments submitted by Whirlpool to EPA regarding their recent revision of the Energy Star requirements for dehumidifiers, the 35-54 pt/d capacity range represents nearly 60% of all dehumidifier shipments (Hoyt 2005). DOE will focus its rulemaking analysis for dehumidifiers on the 35 – 45-pt/d size range only. The Energy Star Web site (http://www.energystar.gov/index.cfm?c=dehumid.pr_dehumidifiers) indicates that the current efficiency of Energy Star-qualified dehumidifiers of the above capacity ranges from 1.3 to 1.5 L/kWh (rated at 80°F and 60% RH indoor conditions). Based on the above it was decided to use EF_d = 1.4 for the baseline system dehumidifier efficiency in the present analysis.

A whole-house humidifier similar to a model offered by Research Products Corporation (http://aprilaire.com/index.php?znfAction=ProductDetails&category=5&item=550) was included to provide the winter humidification function. Product data for the model (sized for <3000-ft^2, tightly constructed homes) specifies a fixed water input flow of 0.5 gal/hr when operating. Hot water from the DHW tank was used for the humidifier supply based on manufacturer specifications for application with heat pump systems (http://aprilaire.com/themes/aa/en/manuals/400.pdf). Figure 6.1 provides an illustration representative of how such a humidifier might be installed. Some of the indoor air stream is diverted or bypassed through the humidifier where water is evaporated from a distribution pad. Energy consumption of the system will be increased compared to operation without a humidifier in two ways: 1) extra water heater consumption to cover the humidifier water usage, and 2) extra heat pump energy use to overcome the cooling effect of the water evaporation on the air stream. The type of 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.

System control set points were as follows: 71°F ±2.5°F and 76°F ±2.5°F for first-stage space heating and cooling, respectively; 66 F ±2°F for second-stage space heating (electric backup heater); 120°F ±5°F for water heating; 55% RH ±4% for dehumidification; and 34% RH ±4% for humidification.

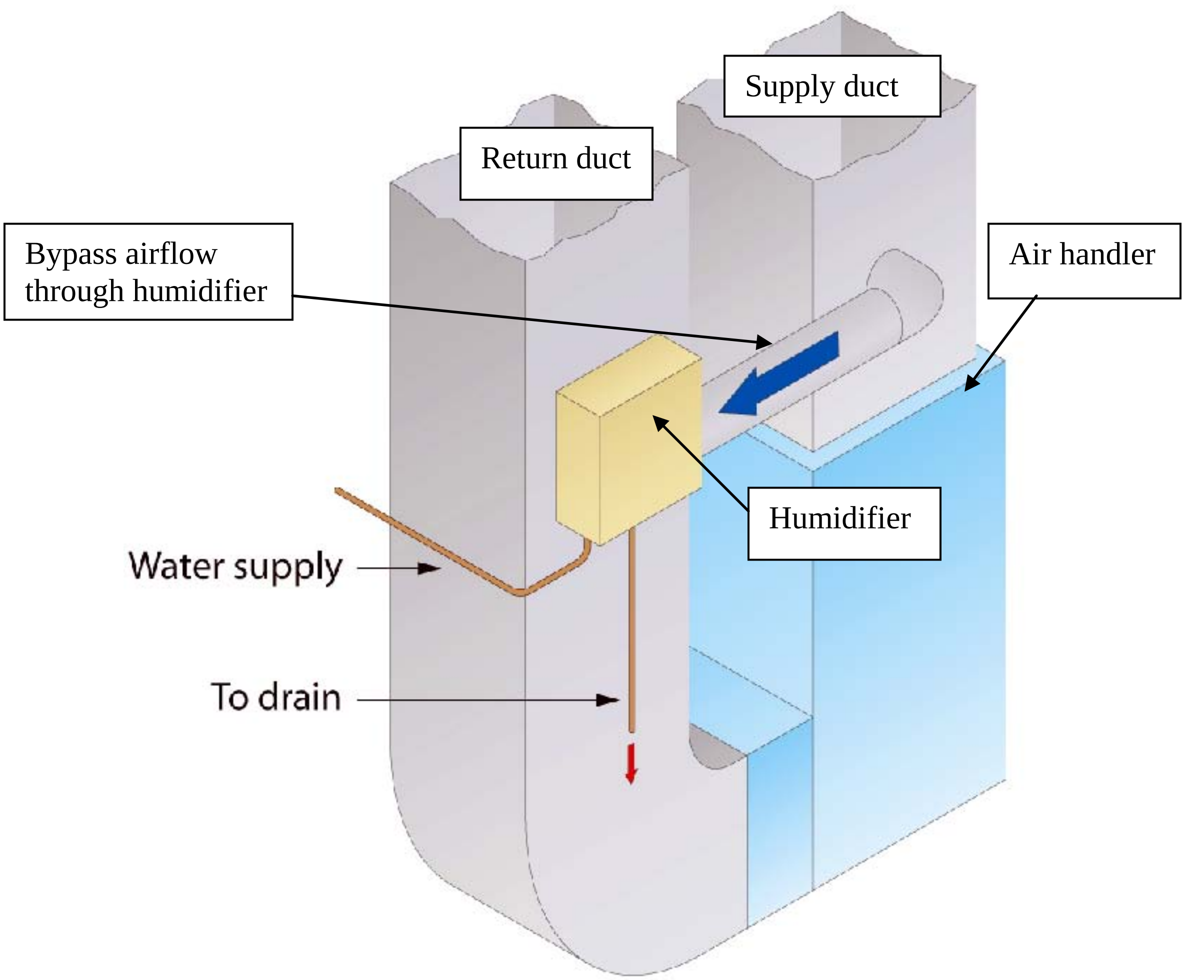

Fig. 6.1. Representative humidifier installation.

The hot water draw schedule assumed for the TRNSYS analyses is shown in Table 6.2. The total daily hot water consumption assumed was ~65 gallons.

Table 6.2. Daily hot water draw schedule assumed for analyses.

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 sink	6:00	1	0.014
Lavatory sink	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 sink	p.m. 12:15	1	0.014
Kitchen sink	12:30	1	0.014
Lavatory sink	4:45	1	0.014
Lavatory sink	5:15	1	0.014
Dishwasher (1^{st} wash)	7:30	1.5	0.048
Dishwasher (2^{nd} wash)	8:00	1.5	0.048
Lavatory sink	9:45	1	0.014
Lavatory sink	10:15	1	0.014
Lavatory sink	10:30	1	0.014

6.4.2 Air-Source Integrated Heat Pump

This system concept, as shown in Figure 3.1, uses one variable-speed modulating compressor, two variable-speed fans, one multiple-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 loads. A 50-gallon water heater tank (same size as for baseline) is included. The same type humidifier as used for the baseline system was assumed to be included with the AS-IHP. Initially the same humidifier water flow as for the baseline case was used as well. But simulations using this flow rate showed water use more than double that of the baseline system. To limit excessive water consumption we cut the water supply rate in half.

The set points for first- and second-stage space heating, space cooling, dehumidification, and humidification as used for the baseline were also used for the AS-IHP. For water heating, the first-stage (IHP water heating) set point was 115°F ±5°F with a second-stage set point of 107.5°F ±2.5°F to control an electric resistance back-up heating element in the upper portion of the DHW tank. The second-stage set point was intentionally set lower than the first-stage set point to maximize the amount of water heating supplied by the IHP.

Our initial IHP system design and operational approach intent (as noted in Sections 3 and 4) had been to utilize available refrigerant desuperheat energy from the IHP compressor discharge gas for water heating whenever the IHP was operating for space cooling, space heating, or dehumidification. A maximum temperature limit was included in the control strategy to shut off the IHP water pump if the DHW tank water temperature exceeded 155°F in the desuperheating mode. A HX effectiveness of 60% was assumed for the desuperheater with the water pump operating at lowest speed. Examination of the analyses results with and without the desuperheater indicated that overall IHP efficiency was better without it. Its elimination also allowed a somewhat simpler system design. See further discussion in the following section.

Also, the initial IHP control strategy assigned priority to space heating over water heating in winter. When there is a call for space heating, the only water heating allowed would be by desuperheater operation as discussed in section 4.5.5 for this control approach. Findings from the annual performance analyses (discussed in the following section) indicated that revising the control strategy to assign priority to water heating over space heating in winter was very advantageous to the IHP. The basic control priority is summarized in the following paragraph.

When there is a call for water heating while in space heating mode, then the unit switches to water heating mode at maximum compressor speed and runs there until either the water heating need is satisfied or there is a call for backup resistance space heating. If the latter occurs, the unit switches back to space heating and runs at maximum speed until the backup resistance heat call is satisfied. Then the unit switches back to water heating mode. Once the water heating demand is met, the unit switches back to space heating operation at the compressor speed specified by the controller and continues until the space heating need is met or there is another call for water heating.

The option of combined space heating and water heating, as discussed in section 4.5.6, was not included in this analysis, nor was that of ventilation with water heating discussed in section 4.5.12. This was because the simulation model cannot yet model these dual HX modes. As such, there are dual HX opportunities to reduce the water heating use in winter that have not yet been simulated. There are also other possible approaches to apportion priority for space heating and water heating, one of which was investigated here.

A unique aspect of the IHP is that the ventilation air is conditioned by the heat pump in both space cooling and space heating modes, and on demand if neither heating nor cooling is required. The ventilation dehumidification mode logic used in the TRNSYS simulation was to initiate when (1) there is no space conditioning call for 1 hour, (2) the outdoor humidity ratio exceeds both a standard indoor humidity ratio of 0.0095 and the indoor humidity ratio, and (3) the outdoor temperature exceeds 68.5°F.

The unit also cycles on demand to dehumidify the space whether or not heating or cooling is required. The air-to-water HX uses waste hot water generated in the desuperheating and space cooling and dedicated dehumidification heat recovery modes to temper the ventilation air, as needed, for space-neutral conditions. A HX effectiveness of 60% was assumed for the tempering coil in the TRNSYS simulation. Compressor, indoor fan, and water pump speed modulation is used to control both indoor humidity and temperature when needed. For the TRNSYS simulations, when in the demand dehumidification mode described in section 4.5.8, the water flow to the tempering coil was modulated by a bypass valve to maintain a neutral supply air temperature.

Another potentially attractive aspect of the IHP concept is that, being a single equipment package, it is better suited than the baseline suite of equipment for being able to curb demand when the grid is stressed in response to a signal from a utility or independent system operator.

6.4.3 Analysis Approach and Results

The annual energy use simulations for the baseline and IHP HVAC systems were performed using the TRNSYS 16 platform (Solar Energy Laboratory et al. 2006). This required conversion of the 1800-ft^2 prototype ZEH description to TRNSYS Type 56 representations. Annual, sub-hourly simulations were performed for the baseline system and AS-IHP for five locations: Atlanta, mixed-humid type climate; Houston, hot-humid; Phoenix, hot-dry; San Francisco, marine; and Chicago; cold. Annual simulations for the IHP systems required that HPDM (Rice and Jackson 2002) be integrated into the TRNSYS simulation system as described earlier in this chapter.

As described previously, the IHP is a continuously variable-speed device and would likely use a proportional/integral/differential (PID) type scheme to set the speeds of the compressor and fans in response to inputs from the various control thermostats. In these simulations this PID control approach was approximated by assuming several discrete speed and capacity levels for the various operating modes — six levels for space heating, five levels for space cooling and demand dehumidification, and four levels for demand water heating.

*Analysis Results with Priority Space Heating***.** Table 6.3 provides summary results of TRNSYS/HPDM sub-hourly simulations for the baseline HVAC system for an 1800-ft^2 prototype net-ZEH for each of the five locations examined in this study. [Note that these values are changed from those given in earlier reports (e.g., Baxter 2007). An error was belatedly discovered in the rated cooling and heating performance data for the baseline heat pump. Correction of the error resulted in somewhat lower annual energy use by the base heat pump except in Chicago, where its estimated energy use increased slightly.] Table 6.4 provides results for the AS-IHP including hourly peak kW demand. Maximum peaks occurred in the winter and generally during the 6 – 8 a.m. time frame (roughly coincident with winter utility peak periods). The water use schedule assumed for the analysis included a significant draw during that time of day, making electric backup element activity likely (adding to backup electric space heating in the colder locales). Maximum summer peaks are somewhat lower and generally occurred during the 6 – 8 a.m. time period as well for the same reason. Summer hourly peaks during the noon – 7 p.m. time period (roughly coincident with summer utility peak time period) were about 1.6 – 2.4 kW for the baseline system vs. about 0.8 – 1.7 kW for the AS-IHP.

Detailed results from the simulations for the ZEH are given in Table 6.5. The total energy consumption and consumption by individual modes for the baseline system are from the hourly TRNSYS simulations. For the IHPs 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 IHPs 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. Temperature control for the IHPs (average indoor temperature and magnitude and duration of extreme high and low periods) was equal to or better than for the baseline in all cities. Indoor space relative humidity control by the IHP met the criteria of no more than 1 – 2% of hours with RH>60% in all locations. Average annual domestic hot water temperature with the IHP was generally a few (2 – 4°F) degrees warmer than for the baseline system. However, the water temperature at the top of the DHW tank in the case of the IHP was 4 – 5°F cooler than for the baseline, indicating somewhat less stratification in the tank water temperatures with the IHP. At no time in any of the cities did the average hourly hot water delivery temperature fall below 105°F for either the IHP or the baseline system.

Table 6.3. Annual site HVAC/WH system energy use and peak for 1800-ft^2 ZEH house with baseline HVAC/WH system

Location	Heat pump cooling capacity (tons)	HVAC/WH site energy use, kWh	HVAC/WH hourly peak kW demand (W/S/SA)*
Atlanta	1.25	7230	8.6/4.6/2.1
Houston	1.25	7380	6.1/4.4/2.2
Phoenix	1.50	6518	6.1/3.9/2.1
San Francisco	1.00	4968	5.7/5.6/1.6
Chicago	1.25	10773	9.7/6.1/2.4

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

Table 6.4. Estimated annual site HVAC/WH system energy use and peak for 1800-ft^2 ZEH house with AS-IHP system (winter humidification active)

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

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

The results summarized in Tables 6.4 and 6.5 show that the AS-IHP exceeds 50% savings over the baseline system in three locations, closely approaches 50% in Phoenix, and achieves ~46% in Chicago. The summer cooling performance of the current R-410A IHP design at extreme hot ambients is not quite high enough to enable reaching 50% annual savings in Phoenix. In Chicago the energy service loads are dominated by heating — space heating and water heating together constitute ~84% of the total load — and the IHP heating efficiency suffers during the extremely cold temperatures encountered in this climate.

Winter peak kW ranged from about 25 to 75% lower for the IHPs than for the baseline. Maximum summer peaks were about 55% to 75% lower, while summer mid-afternoon IHP peaks were ~20 to 60% lower than those of the base system, depending upon location.

The analysis results summarized in tables 6.4 and 6.5 are for water heating priority control in winter and also with desuperheating eliminated. Results using the water heating priority approach for the AS-IHP showed that overall IHP efficiency was clearly improved. While energy use in space heating mode increases somewhat when compared with performance using a space heating priority control approach, the reduction in water heater backup electric element usage more than compensates.

Table 6.5. IHP performance vs. baseline system in ZEH (with humidifier)

Loads (1800-ft^2 ZEH from TRNSYS)		Equipment		
		Baseline	AS-IHP	
Source	kWh	Energy use, kWh (I^2r)	Energy use, kWh (I^2r)	Energy reduction compared to baseline
Atlanta				
Space Heating	4775	1789 (51)	1251	30.1%
Space Cooling	5735	1643	1073	34.7%
Water Heating	3032	3402	924 (142)	72.8%
Dedicated DH	158	208	82	60.4%
Ventilation fan	-	189	20	89.6%
Totals	13701	7230	3349	**53.7%**
Humidifier water use	499 kg		618 kg	
Houston				
Space Heating	1766	648	474	26.9%
Space Cooling	9927	2853	1894	33.6%
Water Heating	2505	2816	556 (91)	80.2%
Dedicated DH	704	875	482	44.9%
Ventilation fan	-	189	12	93.7%
Totals	14902	7380	3418	**53.7%**
Humidifier water use	75 kg		87 kg	
Phoenix				
Space Heating	1580	535	336	37.1%
Space Cooling	9759	3317	2296	30.8%
Water Heating	2189	2477	696 (19)	71.9%
Dedicated DH	-	-	-	na
Ventilation fan	-	189	33	82.7%
Totals	13527	6518	3361	**48.4%**
Humidifier water use	170 kg		229 kg	
San Francisco				
Space Heating	2881	932	607	34.8%
Space Cooling	88	26	23	12.5%
Water Heating	3387	3767	957 (100)	74.6%
Dedicated DH	42	54	11	80.3%
Ventilation fan	-	189	32	83.2%
Totals	6398	4968	1629	**67.2%**
Humidifier water use	34 kg		38 kg	
Chicago				
Space Heating	11425	5448 (1415)	3686 (614)	32.3%
Space Cooling	2550	729	436	40.1%
Water Heating	3807	4286	1644 (327)	61.6%
Dedicated DH	94	121	83	31.9%
Ventilation fan	-	189	17	91.1%
Totals	17877	10773	5865	**45.6%**
Humidifier water use	1369 kg		1639 kg	

Examination of the analysis results in preparation of this report indicated that the contribution of desuperheating to the total water heating energy delivery of the IHP ranged from a low of 4% in San Francisco to 18% in Chicago. It was estimated that the effective COP for water heating for the desuperheating operation (useful water heating energy from desuperheating divided by water pump power during desuperheating operation) was about 1.6 for Atlanta. Given the relatively low water heating energy output and efficiency for the desuperheating operation we elected to examine the impact on IHP overall efficiency from elimination of desuperheating operation. The results for each city are given in Table 6.6 below. Overall the AS-IHP gained about 1 – 1.7% in energy savings vs. the baseline system depending upon location when desuperheating was eliminated. Other advantages from eliminating desuperheating operation include switching from a multiple-speed to a single-speed water pump, elimination of a water temperature control valve, and overall simplification of the IHP control scheme . See the discussion in the GS-IHP status report by Murphy et al. (2007b) for a fuller discussion of the system design implications. Elimination of desuperheating operation also yielded a major reduction in the run time for the pump, with expected attendant benefits of increased pump life. Given these advantages we decided to drop desuperheating operation from IHP designs going forward.

Table 6.6. Comparison of AS-IHP performance vs. baseline HVAC/WH system with and without use of desuperheating (DS) for water heating

Location	HVAC site energy use, kWh			% energy savings vs. baseline HVAC	
	Baseline	IHP with DS	IHP without DS	IHP with DS	IHP without DS
Atlanta	7230	3459	3349	52.2	53.7
Houston	7380	3545	3418	52.0	53.7
Phoenix	6518	3430	3361	47.4	48.4
San Francisco	4968	1704	1629	65.7	67.2
Chicago	10773	5997	5865	44.3	45.6

7. FIELD TEST PROTOTYPE PERFORMANCE SPECIFICATION

Based on previous efforts, a viable embodiment of the IHP concept in a field test prototype is suggested to be a split-system air-source heat pump consisting of three main sections: an indoor compressor section, an indoor air handler section, and an outdoor air handler section.

Included in the indoor compressor section will be the refrigerant compressor, refrigerant accumulator, refrigerant reversing valve, water pump, refrigerant-to-water HX, and selected temperature sensors (with microprocessor as necessary).

Included in the indoor air handler section will be the indoor fan, refrigerant-to-air HX, refrigerant expansion device, water-to-air HX, and selected temperature sensors (with microprocessor as necessary) contained in an enclosure with return air and ventilation air dampers.

Included in the outdoor air handler section will be the fan, refrigerant-to-air HX, refrigerant expansion device, and selected temperature sensors (with microprocessor as necessary). Provision may be made to accommodate additional operating modes such as an outside air economizer mode, if deemed cost-effective in meeting performance goals.

The general performance goal of the system is to provide the energy services required by the 1800-ft^2 ZEH specified for Atlanta (using the National Renewable Energy Laboratory's Building Energy Optimization program) while using no more than 50% of the energy required by the baseline components (13.0 SEER, 7.7 HSPF, 0.90 EF) to provide the same services. The system will operate from a 230-VAC, single-phase, 60-Hz electrical source appropriate for residential service. The preferred system refrigerant is R-410A. The nominal system capacity as conceived is approximately 1.25 tons (15,000 Btu/hr). Based on laboratory prototype tests to date and validated system modeling, the system performance goals are 18.6 SEER and 11.8 HSPF at ARI standard rating conditions in Region IV with the minimum design heating requirement. The target water heating net energy factor is 3.0 with tank losses included. A single-point water heating rating target is 4.0 at 108°F entering water temperature [to the refrigerant-to-water HX, corresponding to the performance condition employed in the HydroTech 2000 performance tables (Carrier, 1989)] and 67.5°F, 50% relative humidity entering evaporator air conditions [corresponding to the entering air conditions specified for the DOE energy factor test (United States Code of Federal Regulations, 2007)].

7.1 Refrigerant Compressor

The concept requires a high-efficiency, hermetic, variable-speed motor/compressor. A suggested option is a rotary compressor with an electronically commutated, BDC-drive motor with a permanent magnet rotor. A minimum R-410A compressor-only EER of 11.2 at ARI air-conditioning rating conditions (ARI 1999) at 60 Hz is recommended. The rated capacity at 60 Hz (3600 rpm) is 9800 Btu/hr to obtain the 1.25-ton design cooling capacity at the 79-Hz design cooling frequency with R-410A. The suggested variable speed ranges to approach the energy savings potential noted in this report are at least 2.8 to 1 in space cooling, 2.0 to 1 in water heating and 3.6 to 1 in space heating. (A maximum speed range of 28 to 100 Hz, for example, was available in a laboratory prototype with applied speeds of 28 to 79 Hz space cooling, 45 to 90 Hz water heating, and 28 to 100 Hz space heating.) A higher maximum speed in heating mode up to 118 Hz, a 4.2 to 1 speed range, is recommended to obtain better heating season performance, as shown in the earlier analysis for the prototype R-410A system.

7.2 Indoor Fan

The concept requires a high-efficiency, variable-speed motor/fan combination. A suggested option is a centrifugal fan driven directly by an integral electronically commutated motor with pulse-width-modulation speed control. The suggested variable speed range is at least 3.5 to 1 with constant airflow control capability.

7.3 Outdoor Fan

The concept requires a high-efficiency, variable-speed motor/fan combination. A suggested option is a multi-bladed propeller fan driven directly by an integral electronically commutated motor with pulse-width-modulation speed control. The suggested variable speed range is at least 2.0 to 1.

7.4 Refrigerant-to-Water Heat Exchanger

The suggested arrangement is counterflow, helical, tube-in-tube with a single refrigerant circuit in the annulus and a single water circuit in a central convoluted water tube. Water-side pressure drop should be no more than 3 psi at 3 gpm water flow. The UA heat transfer rating at maximum water heating speed and 1.8 gpm water flow should be no less than 1075 Btu/hr-F to give 14,000 Btu/hr of water heating at 47°F outdoor ambient temperature. The construction must be double-walled, vented, and approved for potable water use. Suitable provision must be made for either prevention of water-side fouling or access to surfaces subject to such fouling for periodic maintenance cleaning.

7.5 Water-to-Air Heat Exchanger

The suggested arrangement is a perpendicular (relative to the air flow) coil using copper tubing with aluminum fins. The minimum recommended UA heat transfer rating for this coil is 82 Btu/hr-F. Suitable provision must be made for either prevention of water-side fouling or access to surfaces subject to such fouling for periodic maintenance cleaning.

7.6 Indoor Refrigerant-to-Air Heat Exchanger

The suggested arrangement is a sloped (relative to the air flow) coil using grooved copper tubing with enhanced aluminum fins. The minimum recommended UA heat transfer rating for this coil is 800 Btu/hr-F at design cooling conditions.

7.7 Outdoor Refrigerant-to-Air Heat Exchanger

The suggested arrangement is a wrap-around coil using copper tubing with enhanced aluminum fins. The minimum recommended UA heat transfer rating for this coil is 1425 Btu/hr-F at design cooling conditions.

7.8 Electrical Resistance Water Heating Elements

Both lower and upper electrical resistance water heating elements are recommended to have 4.5 kW heating capacity at 230 VAC.

7.9 Water Pump

A three-speed potable water pump rated for 15 ft of head at 3 gpm for duties up to 200°F, 150 psig working pressure is suggested.

7.10 Hot Water Storage Tank

An insulated potable hot water storage tank with a minimum capacity of 50 gallons is recommended.

7.11 Alternative Approaches

Depending on the particular interests of future manufacturing partners, alternative embodiments of the IHP concept may provide the opportunity for cost reductions, performance improvements, or earlier market entries.

7.11.1 Outside Air Economizer Mode

Within the primary design, it is possible to include an outside air economizer mode. In this mode, outdoor and indoor temperature and humidity readings can be used to determine the relative air enthalpies to provide an indication of when outside air can be brought into the space for beneficial cooling. This mode could be initiated when there is a call for cooling while the outdoor air enthalpy is below that of the indoor air by a defined offset. In this mode, the return air damper could be opened to circulate an equal amount of return air along with at least three times the ASHRAE 62.2 continuous ventilation rate (e.g., 3 x 48 cfm = 144 cfm for an 1800-ft^2 house) of outdoor air. This would be equivalent to the flow arrangement for the timed ventilation-only mode, with the difference being that, in the economizer mode, the airflow would be continuous until the cooling thermostat dry-bulb call was satisfied. To assist in effectively ventilating the house in this mode, the bathroom vent fans would be turned on as well (as is done in the Air-Cycler approach of Rudd, 1998). Limiting the outside air economizer flow to the same as for the ventilation-only case would prevent having to increase the size of the ventilation air duct just for this mode of operation. If higher economizer airflow were needed for this function to be effective, the added cost of the larger duct and dampers and added control complexity would need to be weighed against the economizer cooling benefits. This option was addressed in earlier business case analyses (Baxter 2007).

7.11.2 Combined Component Sections

The primary arrangement outlined earlier suggested that components to be located indoors should be housed in two separate sections, the indoor compressor section and the indoor air handler section, as was done in the HydroTech 2000 and Powermiser systems. It is conceivable that housing these components in one single combined indoor section could reduce manufacturing costs. However, the potential impact on installation and maintenance activities would also have to be weighed.

7.11.3 Dedicated Dehumidification Mode with Refrigerant-Source Reheat

An alternative to using heat-pump-provided hot water as the source for air tempering heat in the dedicated dehumidification mode is the use of heat from refrigerant subcooling and some partial condensing in an additional refrigerant-to-air HX located in the air downstream of the indoor evaporator coil. This approach can be used to recover waste heat directly from the refrigerant for air tempering rather than after first heating water.

A drawback of this approach for use in an IHP is that this third refrigerant-to-air HX (replacing the water-to-air HX in the primary embodiment) would need to be employed in series with the outdoor refrigerant-to-air HX and the refrigerant-to-water HX (retained from the primary design approach). This additional HX must be bypassed in some manner in the heating mode. Also, refrigerant charge management could potentially be more difficult with a third potential condenser on the high side of the refrigeration cycle, where most of the refrigerant charge is held.

A major advantage of this approach is that heat that would otherwise have to rejected to potentially higher-temperature sinks (hot water from the storage tank or outside air) can be usefully transferred directly to the air downstream of the indoor evaporator coil (at a relatively low sink temperature). If properly controlled, this can be a quite efficient method of providing air tempering and enhanced dehumidification. However, this heat input must be properly apportioned between the outdoor and indoor refrigerant-to-air HXs to prevent overheating of the indoor air, since the added refrigerant-to-air HX will be quite effective in transferring heat (because the air temperature entering the component will typically be below 60°F). When water heating is also needed, the large temperature difference between the water inlet temperature and air inlet temperature to this component would be expected to limit the amount of water heating that was possible while also tempering the indoor air. In the primary embodiment, air tempering and water heating can be done at the same time. However, this limitation on the water-heating capacity may not be a major drawback, as there is generally an excess amount of rejection heat available in this mode.

As one of our initial design goals was to keep the refrigeration system from becoming too complex, we decided to develop the primary design in lieu of the arrangement described here with three HXs in series. The primary configuration also allows the option for heat-pump-heated water made at moderate ambient temperatures to be used for ventilation air heating in the higher ambient temperature portion of the heating season, if desired. The penalty for using domestic hot water energy for tempering is that it requires more energy input to provide tempering heat by this means than it does to provide such heat from refrigerant in the manner described earlier. The size of this penalty depends on how much domestic hot water energy in the summer and shoulder months with dedicated dehumidification needs is provided by desuperheating (when there is a call for space cooling, but no call for water heating) and heat recovery (when there is a call for both cooling and water heating) during cooling operation, as opposed to dedicated water heating with outdoor air as the heat source.

A major U.S. manufacturer (Lennox 2006) recently released an add-on unit (Humiditrol) for air conditioning and heat pump applications containing the additional refrigerant-to-air HX and a control module for the indoor air handler to enable enhanced dehumidification at SHRs down to 0.3 or lower. To apply this approach to an IHP with dedicated dehumidification capability, the controls and possibly the indoor tempering coil size would need to be modified to more closely approach an SHR of zero by neutralizing more of the sensible cooling provided by the evaporator. Also, the control and performance of simultaneous water heating and air tempering would require further analysis, and the charge management issues of the various modes of multiple condenser operation would need to be addressed. Such an analysis would require that a capability to model systems with two to three condensers in series be developed.

7.11.4 Refrigerant-to-Water Heat Exchanger in Parallel with Refrigerant-to-Air Heat Exchanger(s)

Consideration of a refrigerant-based reheat approach, such as that presented in the preceding section, in conjunction with a parallel arrangement between the refrigerant-to-water HX suggests that this might be a viable alternative for providing dedicated dehumidification and integrated water heating functions. Use of such a parallel arrangement would lessen the charge management challenges of a series configuration. Given that the estimated costs for the hot water based tempering approach from the earlier business case exercise were relatively high ($521 installed from Baxter 2007, pp. 20-21 and p. 34) due to the added component and plumbing requirements, the potential benefits of a refrigerant-based air reheat scheme become more apparent. We also learned from the Lennox experience that the total charge requirement was not increased for a unit with the additional refrigerant-to-air tempering HX (due to the more efficient subcooling process), although managing the charge distribution properly in the two modes is still a technical challenge.

Use of such a parallel arrangement facilitates placement of the compressor in the outdoor unit, the conventional packaging arrangement in split-system heat pumps, which avoids potential noise and vibration issues with a compressor located indoors and enhances compressor cooling options employing ambient air. Further evaluation of this alternative with an augmented HPDM in conjunction with a manufacturing partner is recommended.

7.11.5 Humidification Mode

The primary embodiment makes no provision for humidification. Options are currently available that are either completely “stand-alone” units in that they add moisture directly to air in specific household locations, or “add-on” units that are attached to a central space-conditioning system to add moisture to the ducted supply air stream to achieve some desired general humidity level. If a humidification capability were added to the primary configuration, the main effects (other than the increased complexity of the system and its controls) would likely be an increase in the head capability requirement for the indoor fan and an increased hot water requirement to supply the moisture introduction component. This option was addressed in earlier business case analyses (Baxter 2007).

7.11.6 Multi-Capacity Compressor

Although many compressor options were examined before the primary variable-speed version was selected to meet projected ZEH needs, it is possible that certain multi-capacity models (those having two or more discrete capacities selectable through built-in mechanisms for distinct speed and/or displacement changes) may provide superior cost/benefit results that might facilitate system entry into near-term (pre-ZEH) markets. That is, if, for example, a current multi-speed model with appropriate capacities had a significantly lower cost than variable-speed candidates, but could be employed in a system to achieve most of the targeted IHP savings, then the multi-speed model might be a more attractive component to incorporate into a system for early market entry. Based on the potential interest of a manufacturing partner, it is recommended that such options be examined.

8. CONCLUSIONS AND RECOMMENDATIONS

A laboratory prototype AS-IHP was developed and is described in detail. Salient features include a variable-speed rotary compressor incorporating a BDC motor which provides all refrigerant compression, variable-speed fans for both indoor and outdoor sections, and a multi-speed water pump. The laboratory prototype uses R-22, because of the availability of the needed components that use this refrigerant at the time the prototype was being assembled. It is expected that the HFC R-410A will be used for any ultimate products based on the IHP concept.

Laboratory test data was used to validate component models incorporated into HPDM. Capability has been implemented through an HPDM/TRNSYS linkage to generate full performance maps for advanced integrated heat pumps with multiple modes and compressor speeds of operation and to utilize these performance maps in an automated manner for computationally efficient yet accurate evaluation of advanced equipment performance. This capability has been utilized for subhourly (3-minute time step) analysis of the AS-IHP annual energy use relative to that of an appropriate baseline suite of equipment providing similar energy services in a ZEH in five locations — Atlanta, Chicago, Houston, Phoenix, and San Francisco.

The following specific conclusions from these analyses are highlighted.

1. The AS-IHP system (using R410A) is estimated to closely approach or exceed 50% energy savings vs. the baseline system used in the study in all locations except Chicago. In these four cities the AS-IHP savings for HVAC/water heating/dehumidification energy services ranged from 48.4% to 67.2%. In Chicago, savings were 45.6%. IHP space and water heating efficiency suffers during the extremely low ambient temperatures in Chicago.

2. Modification to the IHP control logic in winter to assign priority to water heating (over space heating) significantly improved overall IHP efficiency and energy savings.

3. Implementation and analysis of split condenser operation in the winter heating mode as discussed in section 4.5.6 is needed to assess the benefits of simultaneous space heating and water heating operation. This same analysis capability can be used as needed to look further at possible dual condenser operation in cooling mode to reduce condensing temperatures under low- or high-speed cooling operation.

4. Use of a desuperheater in addition to full condensing water heating capability is not beneficial in the present IHP design. Alternative ways of providing water heating in the heating season (e.g., assigning priority to water heating as noted in #2 above) were found to be more efficient, as the pump power requirements and loss of coincident space heating was found to more than offset the delivered water heating. Elimination of the desuperheater allows use of a single-speed water pump, results in much lower pump run time, and simplifies the control logic.

The following specific recommendations are made.

1. Based on the conclusions above, the ORNL team recommends that development of the AS-IHP be advanced to Stage 4.

2. Additional IHP controls development is suggested to optimize the efficiency of the IHP water heating mode without compromising indoor temperature control and heating energy consumption.

3. Further analysis is recommended on possible combined-mode space and water heating to improve the water heating performance by more fully utilizing the available capacity of the variable-speed compressor for both thermostat calls. Algorithms for modeling of multiple condensers with different sink conditions within our cycle analysis program (HPDM) need to be developed for this purpose. Such a combined heating mode has the potential of improving the water heating performance for climates like Chicago so that the total energy savings exceeds 50%. This capability will also enable evaluation of an alternative refrigerant-based reheat approach for the AS-IHP in conjunction with an HVAC manufacturer in FY08 that may be more efficient and cost less to implement.

4. To improve space cooling performance in hot-dry climates like Phoenix (to exceed 50% energy savings), some means of evaporative cooling for the condenser should be analyzed. Such an option could be considered, perhaps in lieu of dedicated dehumidification capability which is not used in climates typical of Phoenix.

5. Further evaluation of an alternative parallel water heating arrangement and a refrigerant-based dedicated dehumidification system for air-source application is recommended. This could simplify the charge management issues somewhat.

This report has documented the development of an air-source integrated heat pump through the third quarter of FY2007. Along with two other reports provided previously, this report forms the basis for evaluation of the AS-IHP against DOE’s Technology Development Stage-Gate management criteria for Gate 4, for transition from Stage 3, Advanced Development, to Stage 4, Engineering Development. This report describes the design, analyses, and testing of the AS-IHP and provides performance specifications for a field test prototype and proposed control strategy. The results obtained so far continue to support the AS-IHP as a promising candidate to meet the energy service needs for DOE’s development of a zero-energy home by the year 2020.

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