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Field Tests of a "Drop-In" Residential Heat Pump Water Heater

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ABSTRACT

Sixteen pre-production prototype “drop-in” residential heat pump water heaters were instrumented, pre-tested in a laboratory environment, installed in a wide variety of host homes across the United States, and monitored to determine performance over a nominal one-year period. The units were single-package models in which the 50-gallon storage tank was integral with the heat pump. Provision was made in each of these field test units (1) to store data from the instrumentation, (2) to download the data on command to a central data acquisition computer, (3) to display real-time operational information for installation and troubleshooting purposes, and (4) to allow remote switching between operation in the (primary) heat pump mode and the (secondary) resistance mode. Results indicated that performance was sensitive to hot water usage (amount and pattern), ambient temperature, inlet water temperature, and thermostat setting. Seasonal, weekly, daily, hourly variations were examined. Measured energy savings averaged about 55% as compared to an efficient, electric resistance water heater. Reductions in diversified electricity demand peaks were also found to be substantial. For the sites analyzed, no significant difference was found in hot water deficits for the two modes. Tank loss, dehumidification, and space conditioning impacts were quantified for various situations.

1. DESCRIPTION OF THE HEAT PUMP WATER HEATER

The integrated heat pump water heater (HPWH) unit employed in the field tests described here was a pre-production prototype of a “drop-in” replacement for conventional residential electric resistance water heaters. Development of the design was a collaborative effort among Arthur D. Little, Inc. (ADL), ECR International (ECR, the manufacturing partner), and Oak Ridge National Laboratory (ORNL) with sponsorship from the U.S. Department of Energy (DOE) and the New York State Energy Research and Development Authority (NYSERDA). In 1999, ECR developed the original prototype design based on concepts from two ADL patents (U.S. 5,906,109, May 1999 and U.S. 5,946,927, September 1999). ADL created the related control subsystems. Subsequently, ORNL substantially improved and refined the design, resulting in the achievement by a prototype of an energy factor (EF) rating of 2.47 (Tomlinson, 2000) per the DOE Simulated Use Test Procedure (Federal Register, 1998). Based on this prototype, 10 units were assembled by ECR and delivered to ORNL for a durability test program that was conducted in selected laboratory environments under an accelerated cycling scenario to simulate 10 years of residential use in less than 1 year of testing (Baxter and Linkous, 2002). A second group of 18 similar units was delivered to ORNL for field test purposes. We received data on sixteen units installed in homes as described in this report.

1.1 CONSTRUCTION

As a “drop-in” replacement, the HPWH had a footprint (22.25 in. diameter) common for conventional 50-gal water heaters and had the same installation requirements for power wiring (240 V, single-phase, 30 A circuit with ground) and for water connections (cold inlet from source to dip tube and hot outlet from tank to house usage). The temperature/pressure relief valve, the high temperature cutout switch, and the upper and lower resistive elements were retained from the conventional water heater assembly.

The main visual difference was the addition of a small vapor compression heat pump system (compressor, evaporator, fans, thermal expansion valve, and connective tubing) enclosed in a shroud on top of the tank. This increased the unit height to 60 inches and the unit weight to 180 pounds. The heat pump used a finned-tube refrigerant evaporator to remove heat from the ambient air induced through a rear grille with filter by two fans. In these prototype units, one additional connection was required to conduct condensate from the pan beneath the evaporator to a drain. Although the manufacturer had expended some effort toward offering a condensate management system option, it was not available on the prototype units.

Heat was rejected from a refrigerant condenser comprised of a copper-tubing coil wrapped securely around the outside of the bottom third of the water tank. This heat was transmitted through a highly conductive heat transfer mastic to the tank wall and, thereby, to the water. Polyurethane foam insulation was blown into the space between the tank wall and the outer water heater metal jacket to hold the condenser in place and to minimize heat loss from the tank to the surroundings. The total refrigerant charge of the unit was 16 ounces of R-134a.

1.2 CONTROL SYSTEM

Each field unit employed a microprocessor-based control system originally developed by ADL. The microprocessor received and processed inputs from 7 sources (4 thermistors, 2 switches, and 1 voltage divider) as enumerated in Table 1.1. All thermistors were sampled 32 times during each software cycle and averaged to minimize electrical noise distortions of temperature indications.

Table 1.1. Inputs to HPWH control board

Input	Source
Thermostat setting	Voltage divider
Lower tank temperature	Thermistor
Upper tank temperature	Thermistor
Evaporator temperature	Thermistor
Compressor discharge temperature	Thermistor
mode selection	Toggle switch
condensate pan level (overflow)	Float switch

Based on the processed values of the 7 inputs and on the values of 16 adjustable control parameters (Table 1.2) stored in the electrically erasable programmable read-only memory (EEPROM), the control logic of the software determined which, if any, of five devices should be activated by means of associated relays. The controlled devices were the three heating devices (compressor, upper resistive element, and lower resistive element) and the two auxiliary devices (fan 1 and fan 2). To indicate which heating device was active, three colored light-emitting diodes were installed in a vertical column within the shroud so as to be visible through the evaporator air exit grille. When the upper resistive element was active, the top (green) diode was lit from the control board. In similar fashion, the middle (yellow) diode indicated compressor activity and the bottom (red) diode indicated lower resistive element activity.

1.3 CONTROL LOGIC

The control system logic incorporated hysteresis to avoid excessive equipment cycling and permitted only one heating device to operate at any one time. To minimize hot water runouts, the dominant heating device was set up to be the upper resistive element. Therefore, whenever the upper tank temperature fell below the thermostat setting minus the upper tank temperature hysteresis value, the upper resistive element was activated until the upper tank temperature exceeded the thermostat setting. If the upper tank temperature criterion was satisfied, lower tank heat was called for whenever the lower tank temperature fell below the thermostat setting minus the upper/lower tank temperature differential minus the lower tank hysteresis value. If lower tank heat was called for, the preferred device to activate was the compressor. It was activated in the absence of all the following six conditions:

1) the mode switch was in the resistance water heater mode;
2) the lower tank temperature equaled or exceeded the lower tank temperature/compressor upper limit;
3) the compressor discharge temperature equaled or exceeded the compressor upper limit;
4) the condensate pan level switch indicated overflow;
5) the evaporator temperature equaled or exceeded the evaporator upper temperature limit; and

6) the evaporator temperature equaled or fell below the evaporator lower temperature limit.

It should be noted that this last condition was intended to eliminate the possibility of frost formation on the evaporator (and, therefore, the potential necessity of a defrost cycle) during operation of the heat pump system in cold situations.

Table 1.2. HPWH control parameters

EEPROM location (Hexadecimal)	Parameter	Field value (Decimal)
00	Evaporator upper temperature limit	200
01	Evaporator lower temperature limit	25
02	Compressor upper temperature limit	220
03	Fan 1 upper temperature limit	65
04	Fan 2 upper temperature limit	60
05	Upper tank temperature hysteresis	27
06	Lower tank temperature hysteresis	20
07	Fan hysteresis	5
08	Upper/lower tank temperature differential	10
09	Compressor discharge lower temperature limit	100
0A	Lower tank temperature/compressor upper limit	140
0B	Thermistor oversampling value	32
0C	Compressor restart timer value 1	13
0D	Compressor restart timer value 2	105
0E	Compressor start timer value 1	6
0F	Compressor start timer value 2	180

Once the compressor was activated, a timer was started. If the compressor discharge temperature equaled or exceeded the compressor discharge lower temperature limit when the compressor start timer value (three minutes in this case) was reached (indicating proper compressor operation), the compressor would remain on until:

1) one of six conditions listed above occurred;
2) the lower tank temperature exceeded the thermostat setting minus the upper/lower tank temperature differential; or
3) the upper tank temperature fell below the thermostat setting minus the upper tank temperature hysteresis value (causing upper resistive element activation).

When one of these conditions caused a compressor shutdown, a second timer started. Until this timer reached the compressor restart timer value (six minutes in this case), the compressor could not be reactivated.

Fan operation was limited to periods when the compressor was active. If, at any time after compressor startup, the evaporator temperature fell below the fan 1 upper temperature limit minus the fan hysteresis value, fan 1 was activated until the evaporator temperature equaled or exceeded the fan 1 upper temperature limit. If, at any time after compressor startup, (with fan 1 already on) the evaporator temperature fell (additionally) below the fan 2 upper temperature limit minus the fan hysteresis value, fan 2 was activated until the evaporator temperature equaled or exceeded the fan 2 upper temperature limit. Thus, compressor operation was accompanied, at various times, by no fan operation, fan 1 operation only, or fan 1 and fan 2 simultaneous operation.

If lower heat was called for, but, for any of the reasons indicated above, the compressor could not operate, the lower resistive element was activated. Once activated, the lower resistive element continued operation until:

1) the condition(s) preventing compressor operation were eliminated (causing compressor activation);
2) the lower tank temperature exceeded the thermostat setting minus the upper/lower tank temperature differential; or
3) the upper tank temperature fell below the thermostat setting minus the upper tank temperature hysteresis value (causing upper resistive element activation).

2. DATA ACQUISITION STRATEGY AND IMPLEMENTATION

Four purposes were planned for the field data acquisition/instrumentation system. The first planned use of the system was as an aid during installation, especially in remote locations where inexperienced installers might need some assistance verifying proper operation. Following installation, the system was to be used for its primary function: to measure, preprocess, and store operating data from the HPWH that would be periodically collected by a central personal computer for evaluation. In certain situations, the system might also be used to diagnose current and/or incipient problems with the HPWH or with the data acquisition/instrumentation system itself. Finally, the system was intended to provide for remote switching of the HPWH between heat pump mode and resistance mode.

To implement the strategy, sensors were installed on each HPWH to provide inputs to its associated data logger, which would preprocess and temporarily store the data in final storage. The data logger was connected to a modem that was, in turn, connected to a dedicated phone line. To accomplish the primary function, on a regularly programmed weekly schedule, the central personal computer (with modem) called each data logger (through its associated modem) to download the data accumulated since the last download. Binary format was used for both data storage in the data logger and data transmission to save storage space and transmission time, but data storage in the central personal computer was in comma-separated ASCII format for scanning ease. To accomplish other functions related to installation, troubleshooting, or mode switching, manual calls using the personal computer were initiated to the relevant data logger.

2.1 DATA LOGGER

A Campbell Scientific Model CR23X Micrologger was employed on each HPWH to acquire and convert signals from the instrumentation at chosen intervals, store selected data, provide control/output options (for local or remote mode switching), and accomplish communication and power functions associated with the dedicated modem and phone line. Twelve single-ended voltage input channels received signals from copper-constantan thermocouples located as described later. Two differential voltage input channels received signals from power transducers. Signals from the thermostat, differential pressure transducer, and humidity transmitter were each assigned to one differential voltage input channel. Signals from the water flow meter were assigned to one pulse-input channel. Signals were generated at one control output channel for transmission to the mode control channel on the HPWH control board.

2.2 POWER SUPPLY

Primary power for the data logger, modem, humidity transmitter, and differential pressure transducer was provided by a Shindengen America FYX600/63G AC-DC multiple output power supply. Backup power for the data logger and modem (to record hot water consumption in case of possible primary power outages or power supply failures) was supplied by a set of 10 alkaline D cells installed inside the data logger case.

2.3 MODEM

A Campbell Scientific COM200 telephone modem served as the communication link between each data logger and dedicated phone line. Both power and communication connections were provided by a 9-pin subminiature D connector cable between the datalogger and the modem.

2.4 SOFTWARE

PC208W 2.3 software from Campbell Scientific was used to:

1) create the program the data logger ran to accomplish its various tasks;
2) transfer the program by direct download to the data logger;
3) provide control outputs to the HPWH;
4) establish and maintain communication with the data logger;
5) implement automated and/or manual downloading of data;
6) display real time data values and graphs as required; and
7) prepare weekly summaries of unit performance.

In order to establish preliminary parameters, a "typical hot water day" was postulated based on six events with total hot water usage of 65 gallons being heated from 58°F to 135°F as given in Table 2.1. The estimated "on" time for the heat pump system to supply this quantity of hot water was 654 minutes.

Table 2.1. Typical hot water day postulate

Task	Flow (gpm)	Duration (min)	Usage (gal)
Shower 1	1.33	15	20
Shower 2	1.33	15	20
Handwash 1	1.00	1	1
Handwash 2	1.00	1	1
Handwash 3	1.00	1	1
Handwash 4	1.00	1	1
Dishwash	3.00	7	21
Total		41	65

Distinct strategies were employed to conserve data logger storage space, extend required download intervals, and minimize download time. First, the measurements were divided into three time-interval groups: "rapid" (every 2 seconds, see Table 2.2), "moderate" (every 30 seconds, see Table 2.3) and "slow" (every 10 minutes, see Table 2.4) so as to optimize the data streams required for performance assessment. Second, data transmittal to the final storage area of the data logger was programmed to be event-triggered. That is, data was only transmitted from input storage to final storage when hot water was being drawn (as indicated by the flow meter) and/or power was being drawn (as indicated by the total power transducer).

Calculations for the postulated "typical hot water day" with all relevant measurements, array identifier and time stamps indicated a final storage requirement of 19,680 bytes for rapid data, 36,624 bytes for moderate data, and 1,452 bytes for slow data. With the given 1,172,992 bytes of allocated final storage memory, the data logger then had an effective capacity of more than 20 days of "typical" data. With weekly downloads, this gave a substantial margin to account for unusually heavy hot water usage, faulty unit operation, or communication breakdowns.

Table 2.2. Array ID 2 (every 2 seconds) output variables

Output variable	Final storage (bytes)
Array ID	2
Time	8
Water flow	2
Inlet water temperature	2
Outlet water temperature	2
Array ID 2 Total	16

The estimated minimum (with "clean" direct RS-232 connection to the data logger and minimum background software running) download time for data from 1 week consisting of 7 "typical hot water days" was 421 seconds. Measured download time for a simulated typical week's data under these conditions was 432 seconds. With the switch to modem/phone line connections, download time for the same data was measured to be 542 seconds or about 9 minutes. Again, to allow for larger amounts of data and/or poorer communication links (lower effective baud rates), scheduled intervals between automated downloads were spaced no closer than 30 minutes.

To perform the enumerated tasks, a 784-line program was created and loaded into each data logger using the PROGRAM (EDLOG) portion of the PC208W software package. In recognition of the program run time (approximately 530 ms) and the "rapid" data requirement, the program execution interval was set at 2s. In addition to the operations related to converting sensor signals to the associated engineering units as outlined for final storage above, the program provided for input storage monitoring of reference temperature, logger supply voltage, program run time, overruns, and watchdog (processor, timer, or counter) errors. These values, along with flag operational indicators for water flow, compressor power, upper element power, lower element power, and mode selection and output (heat pump or resistance), gave real-time information during live connections (using the CONNECT portion of PC208W) to each data logger. Live connections also allowed tabular and graphical viewing of the three time-interval groups of variables that were being sent to final storage for subsequent download.

Table 2.3. Array ID 30 (every 30 seconds) output variables

Output variable	Final storage (bytes)
Array ID	2
Time	8
Total power	2
Upper element power	2
Top tank temperature	2
Next tank temperature	2
Next tank temperature	2
Next tank temperature	2
Next tank temperature	2
Bottom tank temperature	2
Condensate level	2
Array ID 30 Total	28

Communication and storage configurations for the modems, dataloggers, and COM ports were created using the SETUP portion of PC208W. Automatic downloads from data logger final storage were also scheduled using SETUP. The STATUS portion of PC208W was employed to monitor communication and data collection status of all components. Downloaded data was stored in unit-enumerated files on the hard drive of the central personal computer. Examination of downloaded data files was accomplished with the VIEW portion of PC208W.

Table 2.4. Array ID 10 (every 10 minutes) output variables

Output variable	Final storage (bytes)
Array ID	2
Time	8
Suction temperature	2
Discharge temperature	2
Evaporator temperature	2
Ambient temperature	2
Relative humidity	2
Thermostat setting	2
Array ID 10 Total	22

3. INSTRUMENTATION AND MODE SWITCHING

Temperature, water flow, humidity, power, and condensate flow measurements were implemented with sensors installed on each HPWH unit as illustrated in Fig. 3.1. The data

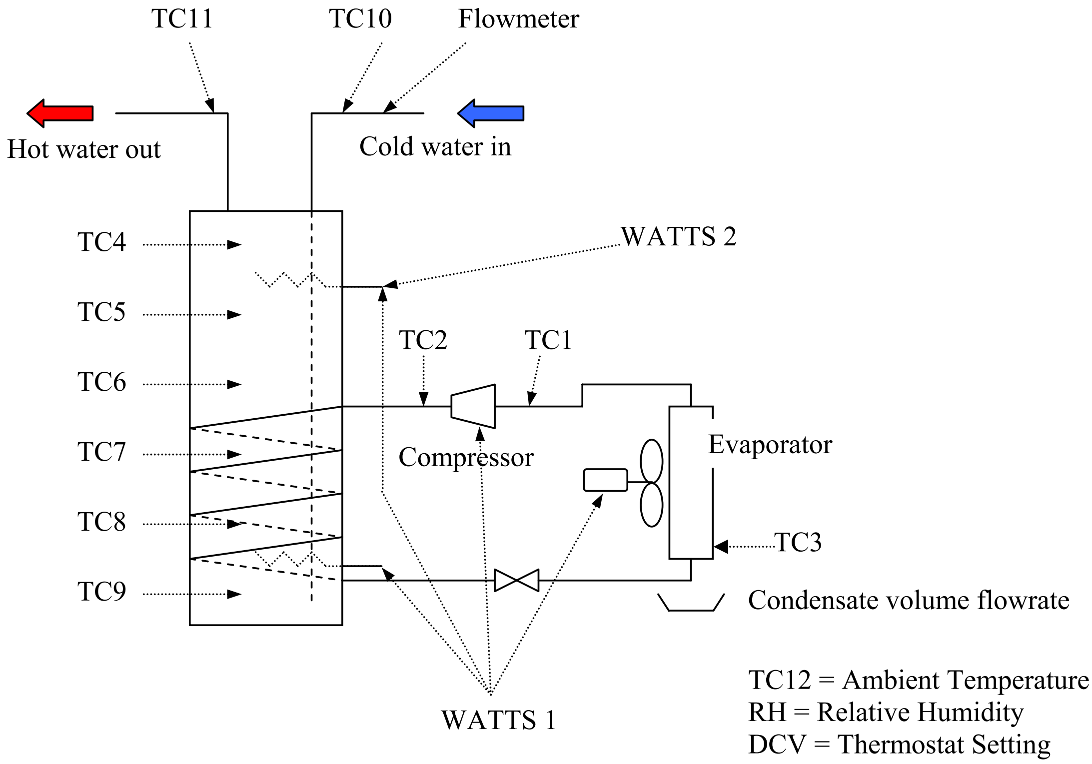

Fig. 3.1. Instrumentation schematic.

logger, power supply, modem, power transducers, and pressure transducer were housed in a covered instrument box (see Fig. 3.2 for view without cover) attached by aluminum brackets to the side of each unit. Associated instrument and power wiring entered the box through connector penetrations. A condensate siphon assembly (see Fig. 3.3) and a humidity transmitter assembly were attached to the outside of each box.

3.1 TEMPERATURE

Copper-constantan thermocouples were used as sensors to provide temperature readings from 12 locations in and around each HPWH. Six of these thermocouples were sheathed at specified lengths, formed into a thermocouple "tree," and installed (and sealed) through the water outlet

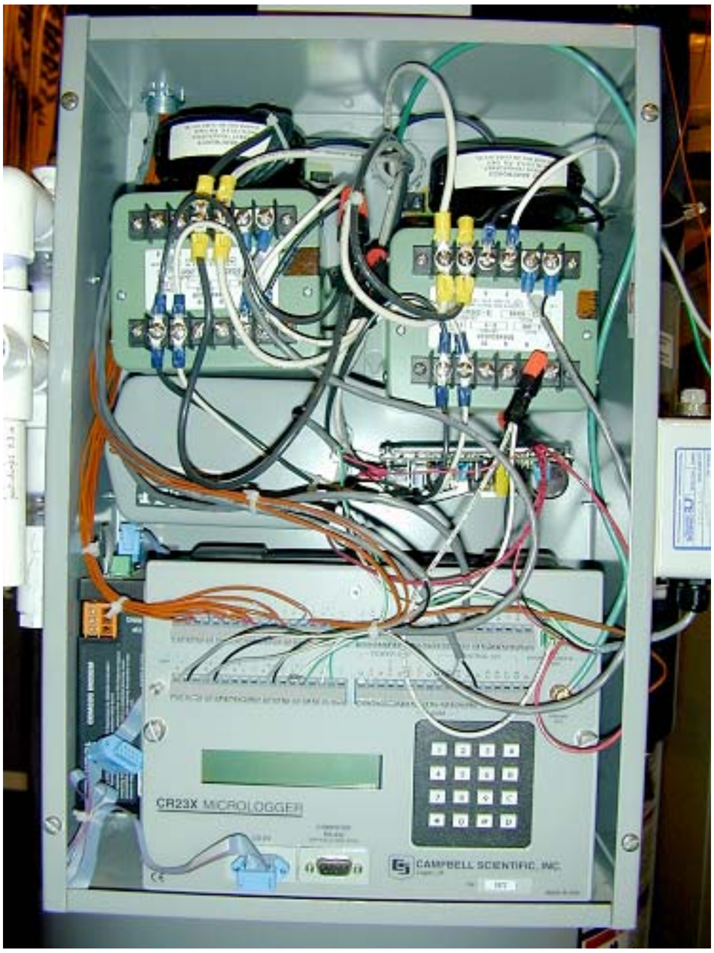

Fig. 3.2 Instrumentation package.

fitting to provide axial tank temperature profiles. The lengths were selected so as to divide the tank into equal volume intervals, facilitating the estimation of average tank temperature. One thermocouple was inserted (and sealed) into the incoming water line to monitor inlet water temperature. Similarly, one thermocouple was inserted (and sealed) into the outlet water line to monitor the hot water supply temperature to the house. Three bare thermocouples were fastened to (but electrically isolated from) the compressor suction, compressor discharge, and evaporator inlet refrigerant lines of the heat pump system. One bare thermocouple was placed in the ambient air away from the evaporator exhaust air stream.

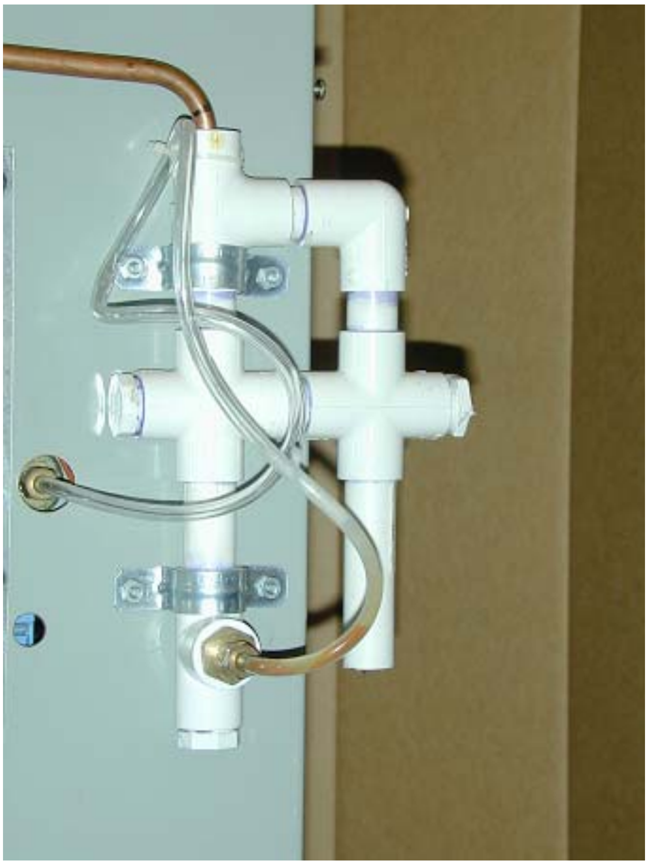

Fig. 3.3 Condensate siphon assembly.

3.2 WATER FLOW

A nutating disc type meter with magnetic coupling to a mechanical dial (Badger Model 25) was installed on the inlet water line to measure water flow. To provide an associated electrical pulse signal for data logging purposes, a reed switch with leads was glued between the nutating disc and the mechanical dial assembly.

3.3 HUMIDITY

A thin-film polymer capacitor type relative humidity transmitter with temperature compensation (Omega Model HX93V) was attached to the side of the instrument box (away from the evaporator exhaust air stream) to measure ambient humidity.

3.4 POWER

Two Ohio Semitronics power transducers with current transformers were used to measure total power to the HPWH and power to the upper element. Connections were implemented so as to exclude power going to the power transducers themselves and to the logger/transmitter power supply. Calibration verification was accomplished in the laboratory by using a reference "revenue" watt-hour meter supplied by a local utility. The total power measurement, from an Ohio Semitronics Model PC5-005X5 power transducer fed from a Model 10424 current transformer with four turns of power lead, was used for all energy performance calculations. The upper element power measurement was used primarily to distinguish between lower and upper element operation.

3.5 THERMOSTAT SETTING

The voltage signal from the thermostat was monitored at the J16 input terminals of the HPWH control board (see Fig. 3.4) to determine the hot water temperature setting selected by the customer.

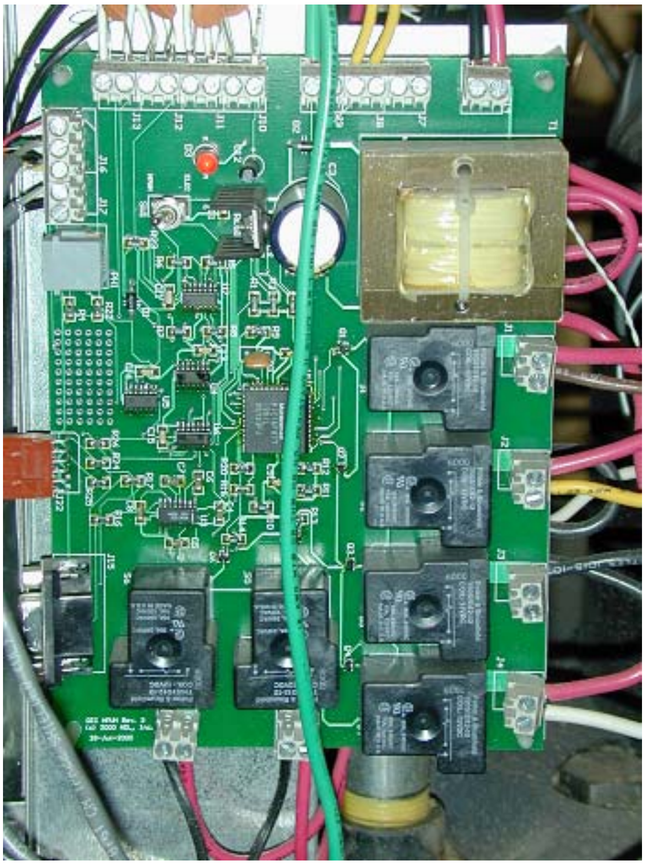

Fig. 3.4 HPWH control board.

3.6 CONDENSATE FLOW

An intermittent siphon system (see Fig. 3.5) was employed for batch measurement of condensate draining from the evaporator. A line from the collection pan beneath the evaporator conducted condensate to the dead-end side of the siphon system. Two separate horizontal sections of PVC pipe connected the dead-end side to the open side of the system. A small-diameter Tygon tube running from the lower portion of the dead-end side through the lower horizontal section of pipe to the bottom end of the open-ended side to form the inverted U required for the intermittent siphoning action. Silicon tub caulk was used to provide a waterproof seal between the outside of the Tygon tube and the inside of the horizontal pipe. The open-ended side of the system was connected to an appropriate drain. The condensate level pressure line was connected to a tee near the bottom of the dead-end side. As condensate gradually drained into the dead-end side, the level (and the pressure at the bottom) of the dead-end side increased until, at some point, liquid filled the cross section of the top of the Tygon U-tube, creating a siphon action which drained the calibrated "batch" (typically 30 ml) from the dead-end side down to the bottom of the Tygon tube. The pressure differential between the bottom of the condensate column in the collector portion of the siphon and the ambient air was measured using a wet/wet low differential pressure transmitter (Omega Model PX154-005DI). Each time the indicated differential pressure dropped rapidly, a condensate dump was indicated.

3.7 MODE SWITCHING

Either an "open" (resistance mode) or a "closed" (heat pump mode) signal was sent from the control input/output channel on the logger to the J17 input terminals of the HPWH control board (see Fig. 3.4) to determine which operational mode was in effect. Successful implementation of this control strategy required placement of the control board switch in the open position and enabled both local and remote switching between operational modes. Remote mode switching was accomplished during phone connection to the data logger by manually changing the value of the associated flag in the custom data logger program.

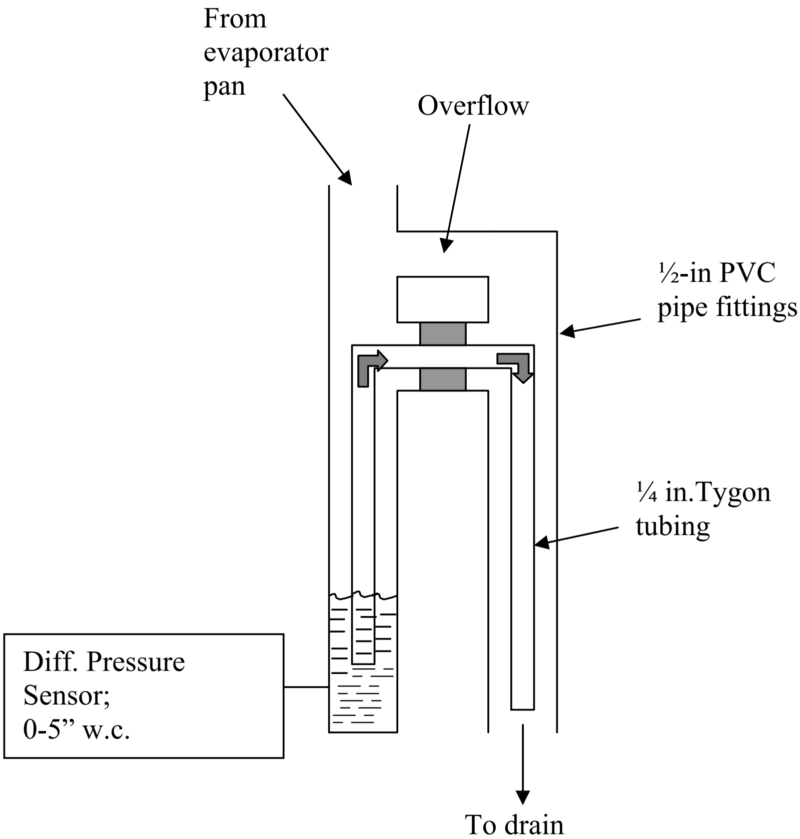

Fig. 3.5 Intermittent siphon arrangement.

4. LABORATORY CHECKOUT

Because the field test units were essentially pre-production prototypes, and because some operational difficulties had been encountered previously with an earlier shipment of 10 units employed in the durability tests (Baxter and Linkous, 2002), intensive checkout exercises were undertaken in the laboratory before shipment to the field test homes. Two tandem laptop computers were used to monitor unit performance. One was connected by means of its serial port to the data logger through its optically isolated RS-232 port to monitor readings from the installed instrumentation package. The second was connected by means of its serial port to the RS-232 port of a Microchip in-circuit debugger module which, in turn, was connected to a Microchip in-circuit debugger header on the HPWH control board through its 6-conductor modular cable jack. With Microchip's MPLAB Integrated Development Environment software, this second link permitted communication with the microcontroller on the HPWH control board, facilitating the loading of software and implementing EEPROM settings. This link was also used to monitor perceived values of sensor inputs and step-by-step operation of the associated on-board C++ control program. When necessary, changes to the EEPROM settings and control program were implemented through this link.

With the monitoring systems in place, an extensive series of water heat-up and draw sequences was undertaken to exercise the various sensors and control systems. Anomalies were detected, causes identified, and solutions implemented until proper operation was verified. As was found previously for the 10 durability test units, the most common problems were associated with unreliable splices in thermistor leads. Other detected anomalies were related to control board defects such as shorted mode switches, loose wiring, a failed light-emitting diode connector, and a broken voltage regulator. In addition, two operational problems were found in the control program software. When all appropriate corrective measures had been completed, a final set of tests was conducted to verify correct remote communication and control operations using the central computer/modem/telephone line/modem/data logger system.

5. DATA PROCESSING

From the weekly downloads of data from the final storage of each logger, the SPLIT portion of the PC208W software was used to prepare weekly summaries. Data from the 2-second arrays were integrated with totalizing algorithms to calculate weekly hot water consumption and associated delivered heat value. Minimum and maximum algorithms were used to estimate source (inlet) and maximum supply (outlet) water temperatures, respectively. Data from the 30-second arrays were integrated with totalizing algorithms to calculate weekly electricity consumption, compressor run time, upper element run time, and lower element run time. Beginning and ending tank temperature data from the 30-second arrays were used to determine average tank temperatures. These values were used to adjust the weekly hot water consumption values to account for additional heat stored in or removed from the tank. The effective weekly coefficient of performance (COP) was calculated (in nondimensional units) by dividing the net heat delivered to the hot water system by the electrical energy consumed. Minimum and maximum algorithms were applied to data from the 10-minute arrays to establish ambient temperature range, thermostat setting range, minimum evaporator temperature, and maximum compressor discharge temperature. An averaging algorithm was used to determine average run-time ambient temperature.

Each set of weekly summary data was entered on a line in a spreadsheet designated specifically for each field test unit. If anomalies in any weekly data were identified, attempts were undertaken to understand and/or remedy the cause. In some cases, this involved a manual call to the unit data logger, the utility contact, or the homeowner.

6. SITE SPECIFICS

The cooperation of nine electric utilities was crucial in identifying potential host residences for the field test units, coordinating their installation, arranging for dedicated data acquisition telephone lines, and assisting with troubleshooting exercises. The testing benefited greatly from the wide range of situations made accessible by this cooperation and the willingness of the homeowners to participate.

As indicated in Table 6.1, 16 municipalities in 9 states, ranging from Connecticut to Washington to Texas to Florida, were represented. Fifteen of the houses had two adults, with the number of children ranging from none to five. One house actually had no residents, but served as a Habitat for Humanities construction office. In this case, scheduled hot water draws were accomplished by a solenoid valve activated by a timer. Two houses were served by well water; the remainder by central (city or county) water systems.

Five units were located in basements, of which one was conditioned (with direct access to the heating/ventilating/air-conditioning systems), three unconditioned, and one semi-conditioned. Four units were installed in (unconditioned) garages. Two units were sited in (conditioned) laundry rooms, and two more in (unconditioned) workshops. An additional two units were installed in utility rooms (one unconditioned and one conditioned). The unit in the unoccupied house was installed in a small closet with two exterior walls, one interior wall, and one shut interior door.

Two of the homes had no previous occupants and, therefore, no previous water heater. Nine residences had previously had conventional (that is, resistance) electric water heaters with the same nominal storage volume (50 gal) as the prototype heat pump water heater field test units. Three of the homes had previously had conventional electric water heaters with larger (two 80-gal, one 52-gal) storage capacity. Two of the residences had previously had add-on electric heat pump water heaters (from other manufacturers).

In another variation with potential performance implications, one installation (South Dakota) was subject to utility load control for a portion of the test period. Field test project personnel were able to observe installation procedures at 13 of the sites and to verify initial proper performance of the field test units. The remaining three sites (Washington, Texas, and South Dakota) were monitored during installation, and remote assistance was provided by field test project personnel as required. Upon installation, for safety reasons the thermostat setpoint temperature was 120°F. However, residents were permitted to adjust this value as they saw fit throughout the test period by means of a knob on the front of each unit.

Table 6.1. Summary of unit installation situations

Unit	State	City	Residents	Water source	Unit location	Conditioning	Previous heater
1	Alabama	Douglas	2 Adults	County	Basement	None	Add-on HPWH
3	Florida	Melbourne	2 Adults, 1 child	City	Garage	None	None
4	Alabama	Verbena	2 Adults, 5 children	County	Laundry	Full	80 Gal
5	Florida	Milton	2 Adults, 2 children	City	Garage	None	50 Gal
6	Tennessee	Knoxville	2 Adults, 2 children	City	Garage	None	80 Gal
7	Connecticut	Cromwell	2 Adults	City	Basement	Semi	Add-On HPWH
8	Washington	Seattle	2 Adults, 2 children	City	Basement	None	52 Gal
9	Connecticut	East Hampton	2 Adults, 1 child	Well	Workshop	None	50 Gal
10	Tennessee	Lenoir City	None	City	Closet	Semi	None
11	Florida	Pensacola	2 Adults, 3 children	City	Garage	None	50 Gal
13	Georgia	Danielsville	2 Adults, 2 children	Well	Utility	Full	50 Gal
14	North Carolina	Wake Forest	2 Adults, 1 child	City	Workshop	None	50 Gal
16	Georgia	Gainesville	2 Adults, 3 children	City	Utility	None	50 Gal
18	Georgia	Conyers	2 Adults, 2 children	City	Basement	None	50 Gal
19	Texas	Smithville	2 Adults	County	Laundry	Full	50 Gal
20	South Dakota	Madison	2 Adults, 2 children	City	Basement	Full	50 Gal

7. DATA ANALYSIS AND INTERPRETATION

Average weekly performance data taken from 16 field test units while operating in the heat pump mode are presented in Table 7.1. Periods with incomplete or problematic data have not been included. A wide range of operating situations and performance is illustrated in this brief summary.

Table 7.1. Summary of average weekly HPWH field test performance

City	State	Cents per kWh	Cond (A)	Gal	kBtu	kWh	COP	kWh Savings	% Savings	$ Savings
Knoxville	TN	5.81	N	648	346	55.2	1.84	62.7	53.2	3.64
Verbena	AL	5.29	F	794	388	60.8	1.87	60.2	49.7	3.18
East Hampton	CT	9.64	N	511	310	44.3	2.05	56.7	56.1	5.46
Seattle	WA	8.65	N	778	397	74.3	1.57	52.2	41.3	4.51
Pensacola	FL	6.40	N	611	280	39.2	2.09	51.0	56.5	3.26
Wake Forest	NC	8.98	N	455	231	33.5	2.02	46.1	57.9	4.14
Gainesville	GA	6.89	N	411	210	30.4	2.02	43.8	59.0	3.01
Danielsville	GA	6.89	F	430	235	28.2	2.44	43.5	60.7	3.00
Melbourne	FL	7.00	N	499	179	23.8	2.21	38.8	62.0	2.72
Douglas	AL	5.90	N	443	228	35.7	1.87	38.5	51.9	2.27
Milton	FL	6.40	N	451	216	33.2	1.91	37.1	52.8	2.38
Madison	SD	4.84	F	309	180	24.7	2.14	35.9	59.3	1.74
Lenoir City	TN	5.81	S	502	219	39.1	1.64	32.2	45.2	1.87
Smithville	TX	7.00	F	375	148	22.0	1.97	29.6	57.4	2.07
Conyers	GA	6.89	N	305	126	17.1	2.16	26.9	61.1	1.85
Cromwell	CT	9.64	S	270	135	18.4	2.15	25.6	58.1	2.46
simple averages		7.00		487	239	36.2	2.00	42.6	55.1	2.97

(A) Location of Unit: N = non-conditioned, F = fully-conditioned, S = semi-conditioned

A major factor that contributed to reduced COP in some units was a relatively large amount of resistive element (upper and/or lower) operating time. The primary cause of increased upper element operation was increased concentration (patterned combination of amount and time) of household hot water draws. Increased lower element time was caused by items that kept the control system from allowing compressor operation. In the case of Unit 10 (see Table 6.1), the substantial cooling of air in the very small installation closet (especially during cold weather periods) caused the evaporator temperature to reach its lower limit, shutting down the compressor and activating the lower element. In the case of Unit 8, concentrated hot water draws combined with low inlet water temperature and low ambient temperature (again encountered during cold weather periods) prevented the compressor discharge temperature from reaching the required value in the allowed time, shutting down the compressor and activating the lower element. Less frequent causes of lower element activation were excessive compressor discharge temperatures (especially in high ambient and high thermostat setpoint temperature situations) and on/off compressor timer violations. Decreased ambient temperature and/or increased thermostat setpoint temperature increased standby losses (conduction of heat from the tank to the surrounding air) and also decreased ideal (that is, Carnot potential) performance because the system was required to lift heat over a larger temperature range. Given standby losses also had a larger performance reduction effect in situations with less heat demand. These factors are summarized in Table 7.2.

Table 7.2. Factors that can reduce COP

Large draws
Closely spaced draws
Small installation volume
Low inlet water temperature
Low ambient temperature
High ambient temperature
High thermostat setting
Low heat demand

Figure 7.1 illustrates daily hot water consumption for one week of Unit 9 operation in the resistance mode. It is clear from these data that the heaviest usage occurred during the two

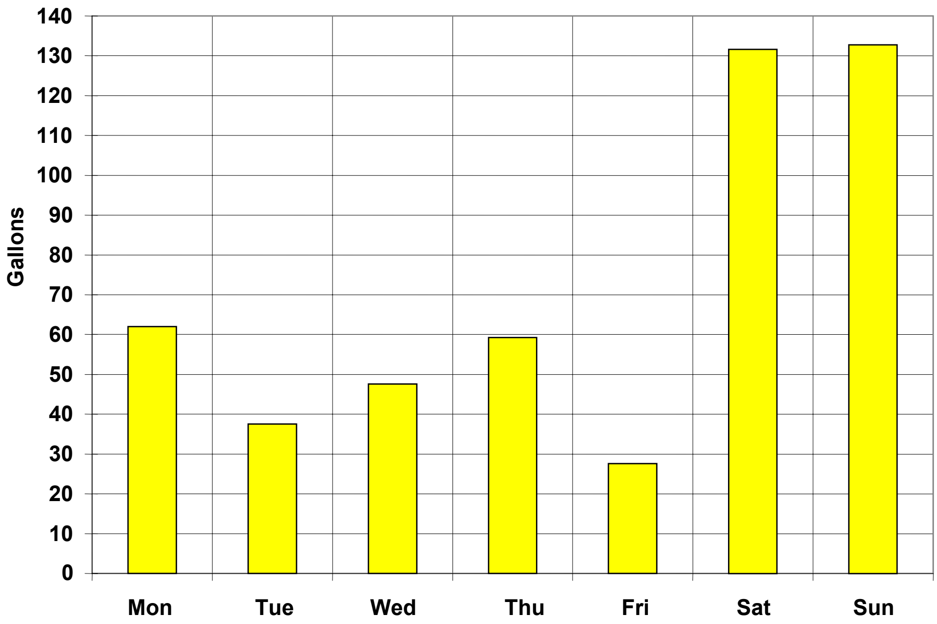

Fig. 7.1. Sample daily hot water consumption pattern for Unit 9 in resistance mode.

weekend days. In Fig. 7.2, the dramatic variation of hourly hot water consumption for Saturday of that week is demonstrated. This pattern of variation is followed closely by the hourly electricity consumption given in Fig. 7.3. Data from the same unit operating in heat pump mode two weeks later shows a similarly uneven pattern of hot water usage in Fig. 7.4. However, the corresponding electricity consumption data presented in Fig. 7.5 shows a decidedly more uniform demand pattern, with consistent, low power draws (from the compressor and fans).

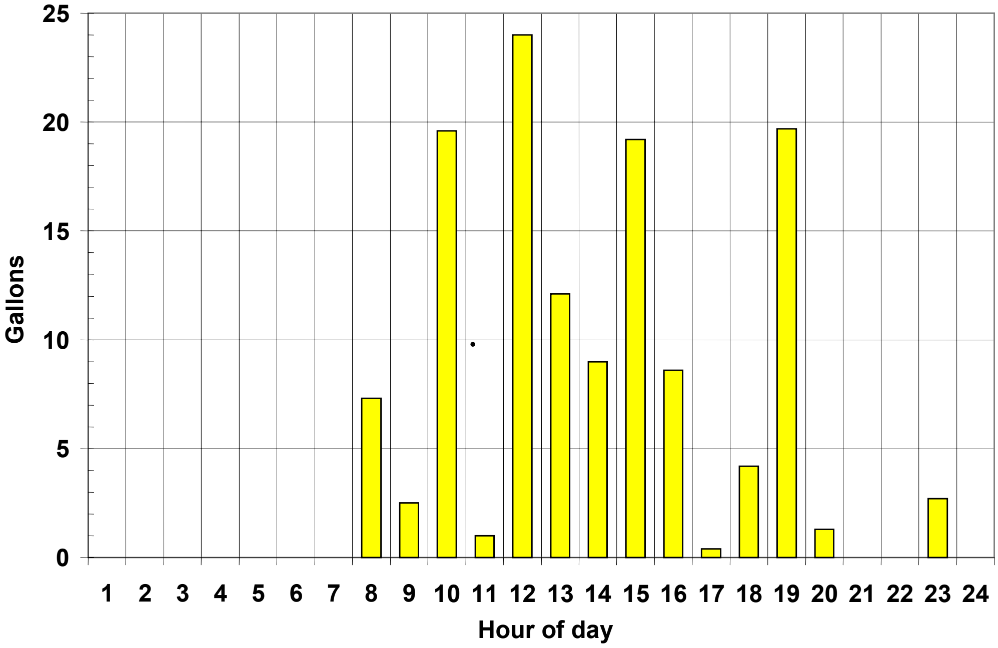

Fig. 7.2. Sample hourly hot water consumption for first Saturday; Unit 9; resistance mode.

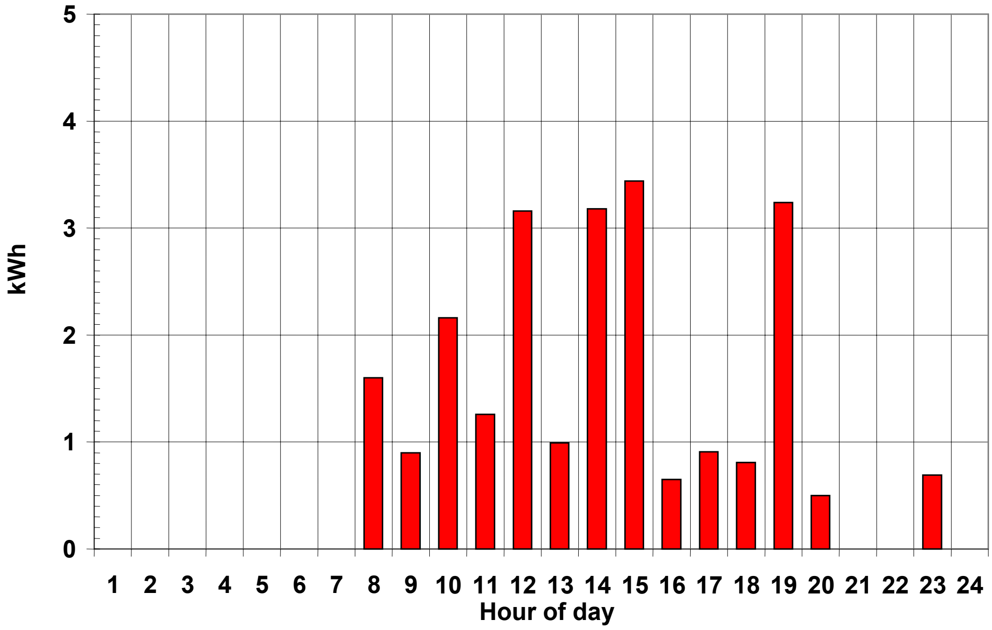

Fig. 7.3. Sample hourly electrical energy consumption for pattern in Fig. 7.2.

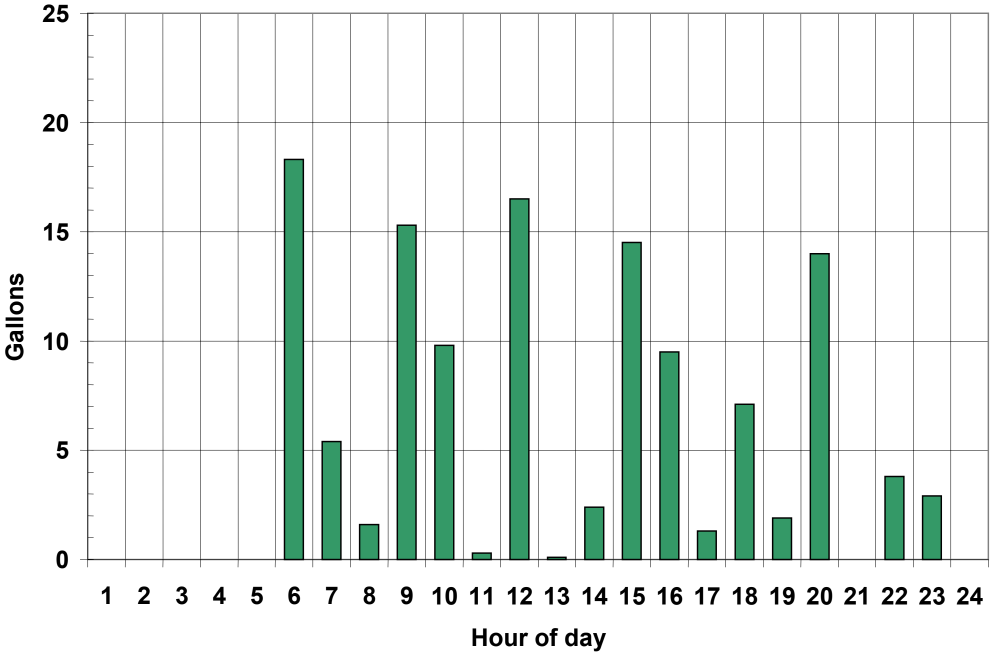

Fig. 7.4. Sample hourly hot water consumption for second Saturday; Unit 9; heat pump mode.

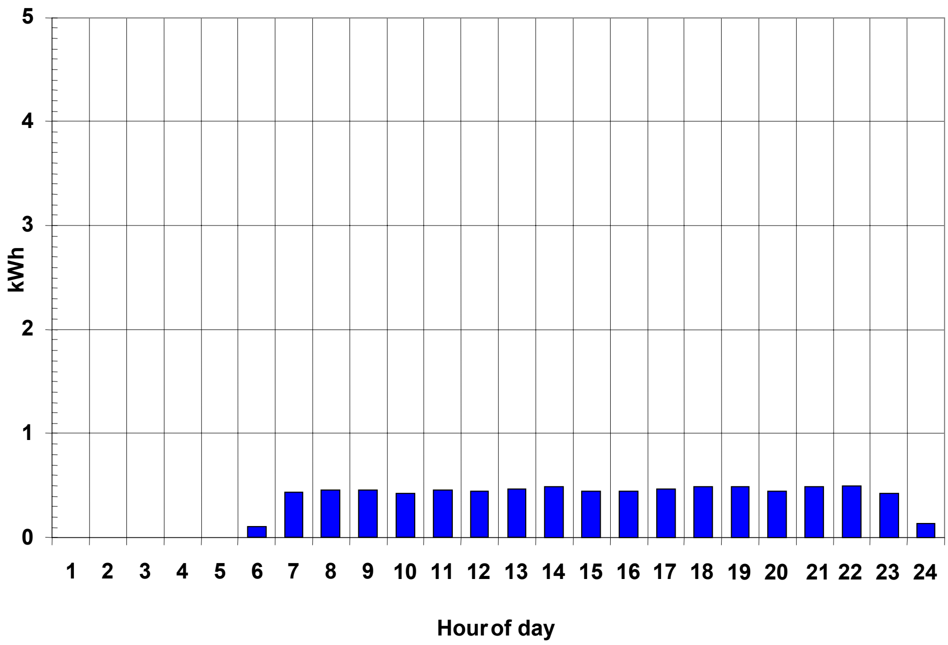

Fig. 7.5. Sample hourly electrical energy consumption for pattern shown in Fig. 7.4.

A limited amount of resistance mode data was acquired from 14 of the field test units. As expected, the vast majority of the heating in this “conventional” mode was accomplished by the lower resistive element. Of course, the size of the deviation in the COP from unity in this case was an indication of the relative size of standby losses, as mentioned earlier.

Data for the two modes are plotted as electrical energy consumption versus delivered heat energy in Fig. 7.6. For any given amount of delivered heat energy, the energy savings that could be achieved by the use of heat pump mode as compared to resistance (conventional) mode can be estimated as the vertical distance between data for the two modes.

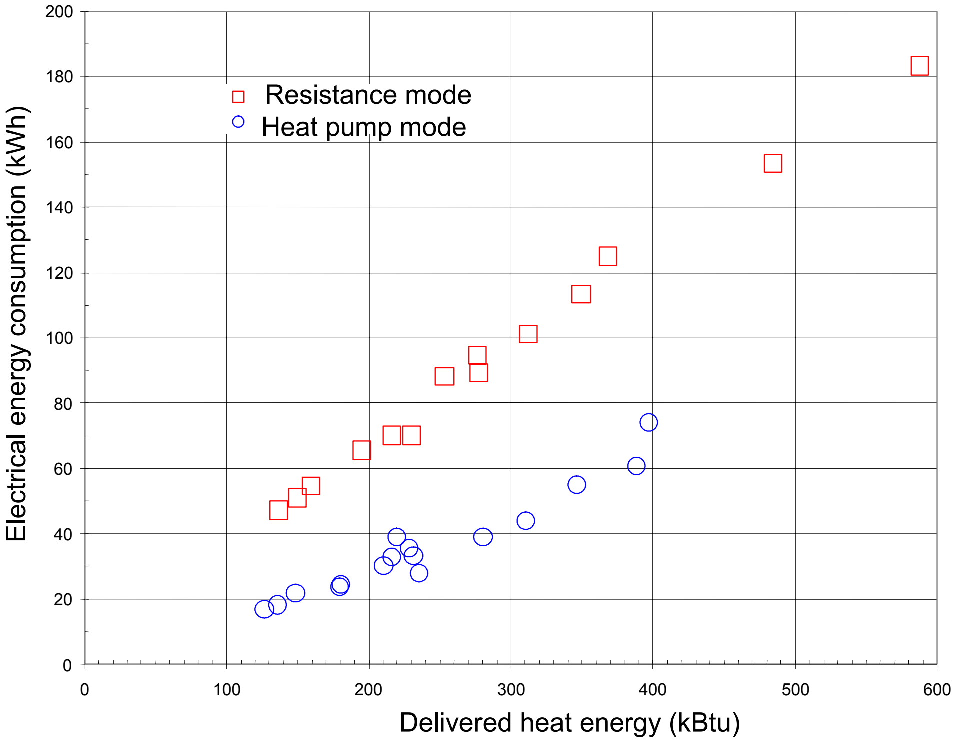

Fig. 7.6. Average performance of units in heat pump and resistance modes.

7.1 COMPARISON OF ELECTRIC DEMAND OF HPWH AND RESISTANCE MODES

The average diversified electric demand for a subset of units operated in the two modes was determined and used to evaluate the impact of a changeover from resistance storage water heaters to the HPWH. Six of the sites were operated in the resistance heating mode for 6 weeks during the winter to provide some of the data needed. We chose a 15-minute demand interval as representative of the interests of most utilities. Diversified demand is a term that represents the average demand of a large number of appliances all operating as designed and installed in a single climate (and utility region), and while the study had only a few sites, the estimate for diversified demand was made using data gathered for each of the sites over the 6-week period.

This represents a total of 180 house-weekdays of data. Following the 6-week period of data collection in the resistance heating mode, we switched the unit over to the heat pump mode and continued to collect data as before. Data from each of these periods was used and plotted as shown in Fig. 7.7. The "double hump" demand for the six units in the resistance mode shows that the diversified demand peak of the resistance water heaters varies from about 2.6 kW in the morning to 1.6 kW in the evening. There was relatively little demand in the early morning hours as expected due to the rapid recovery of the units in the resistance mode.

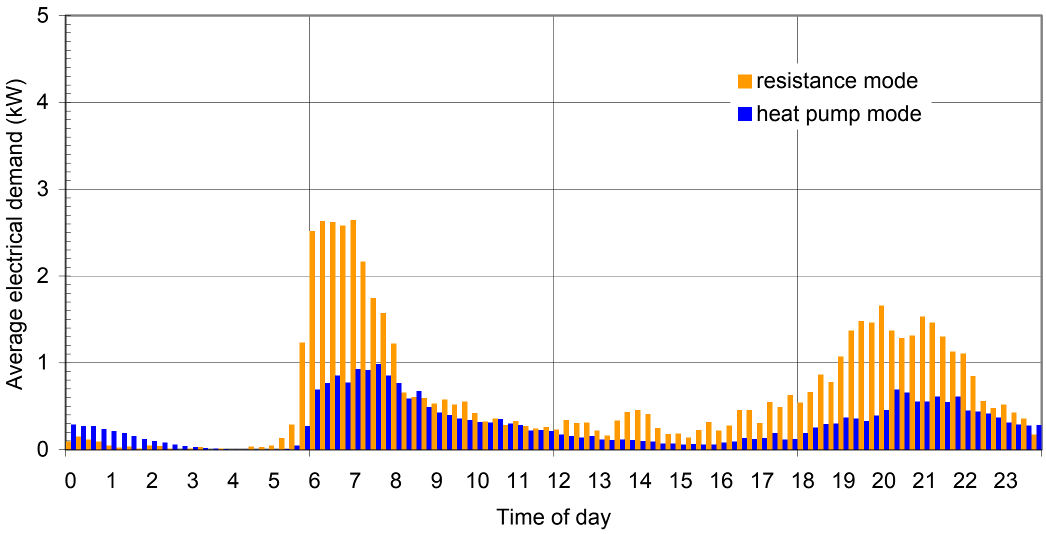

Fig. 7.7. Diversified winter weekday demand of six units in heat pump and resistance modes.

Fig. 7.7 also shows the diversified demand data for the units in the heat pump mode (a mode that includes heat pump as well as supplementary resistance energy). The morning peak is about 1 kW and the evening peak is 0.7 kW – a savings of 1.6 kW in the morning and 0.9 kW in the evening as compared to the resistance mode. Interestingly, the diversified demand of the unit in the heat pump mode exceeds that in the resistance mode in the early morning. This is due to the slow recovery of the HPWH to restore the tank temperature from draws of the previous evening. Although small, the extra demand imposed by the HPWH in the early morning hours is a utility benefit as most utilities are interested in shifting load to these hours. The HPWH daytime profile shown in Fig. 7.7 is generally much flatter than the profile for the resistance water heater, and considerably lower than the demand of a resistance water heater at any time of day.

Data in the same format are presented for the summer period in Fig. 7.8. As expected, the summer peaks were lower than the respective winter peaks, but substantial peak reductions were still indicated for the heat pump mode.

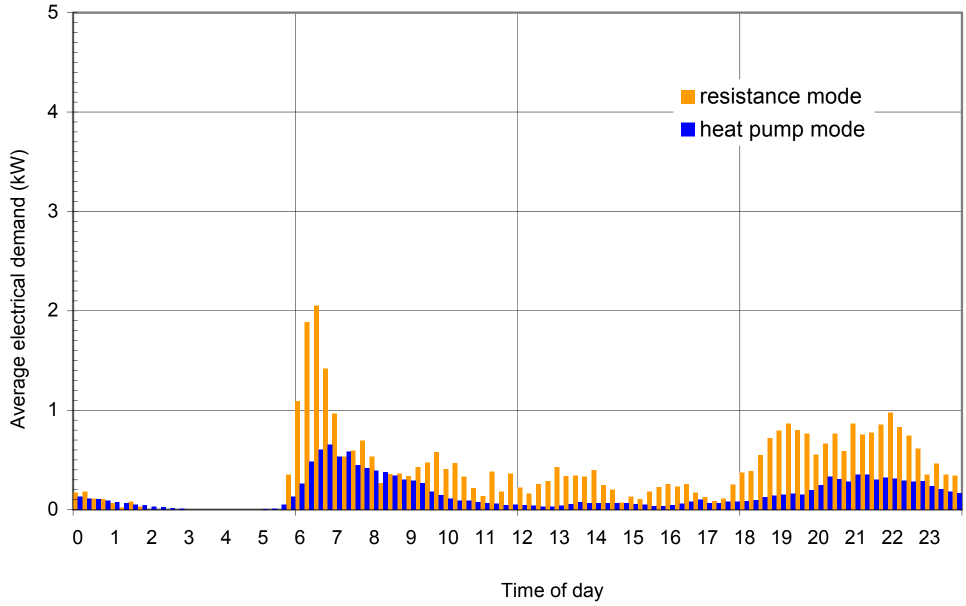

Fig. 7.8. Diversified summer weekday demand of six units in heat pump and resistance modes.

7.2 ENERGY SAVINGS OF THE HPWH

The overall weekly performance of the HPWH field test units is shown in Table 7.1. Savings in kWh (column 9) were calculated based on the measured COP of the units in the resistance mode and normalized to the hot water heat consumption shown in column 6. As can be seen in this table, the average savings in electric energy consumption was approximately 55%. Table 7.1 also shows the marginal electricity costs (column 3), the conditioning of the HPWH location in the home (None, Semi, and Full), and the delivered efficiency (COP) which is the energy contained in the hot water delivered to the user divided by the electrical input to the water heater. The last column on the right shows the average weekly savings delivered by the HPWH as compared to the resistance water heater. Savings range from \$1.74 to \$5.46/week based on utility rates for each site.

The performance of the field test units depended on their location within the dwelling. Where overall ambient temperatures were low as in the case of a unit located in a small closet or a unit in an unheated basement, delivered efficiency (ratio of thermal heat delivered to electrical input) was lower than for locations where the HPWH was in warmer ambients (garage in Florida, e.g.). Fig. 7.9 shows weekly data for all of the field test units, disaggregated to show trends for performance versus average ambient temperatures. The data were filtered and fitted with trend lines for the ambient temperature ranges indicated. The data show the effect of average ambient temperature on HPWH performance and the relation between the two is as expected.

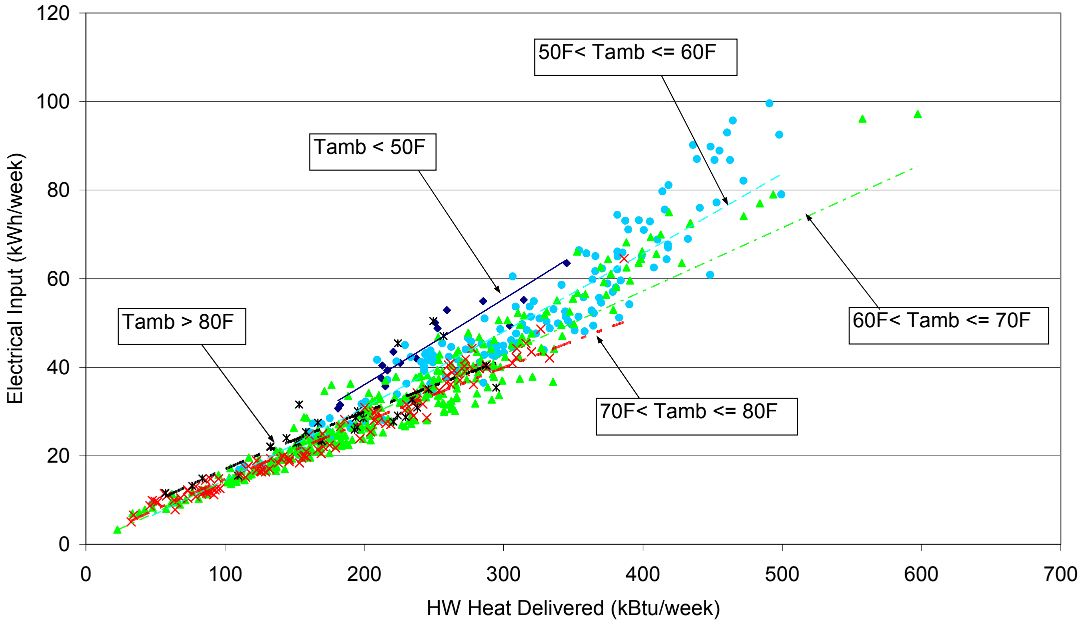

Fig. 7.9. Ambient temperature chart

We also summarized the time that each of the heating sources in the HPWH (the compressor, the upper element and the lower element) was operating for a typical week. As shown in Fig. 7.10, compressor weekly run times varied from 80 to 30 hours per week depending on the hot water consumption. Upper element use tended to be greater with long compressor run times (more hot water consumption) and little to none for units that provided smaller water demands. As described earlier in the report, the lower element ran only when the compressor tried to operate in conditions outside of an acceptable operation envelope. One case is the unit in Seattle where low ground water temperatures (low entering water temperatures to the HPWH) combined with heavy and concentrated hot water demands were sufficient to engage the lower element for recovery periods particularly in winter. This led to greater use of the lower element than might have otherwise occurred.

Finally, we mapped the electrical energy consumption for the compressor, upper element and lower element. As shown in Fig. 7.11, in all but one case, the electric energy consumed by the compressor greatly exceeded the energy consumed by either element. In most cases, over 80% of all electrical energy was consumed by the compressor. In the one case mentioned earlier (Seattle), the electrical energy consumed by the elements exceeded that used by the compressor. However, even in this “worst case,” the HPWH saved more than 41% of the electrical energy that would have been required by a conventional resistance water heater.

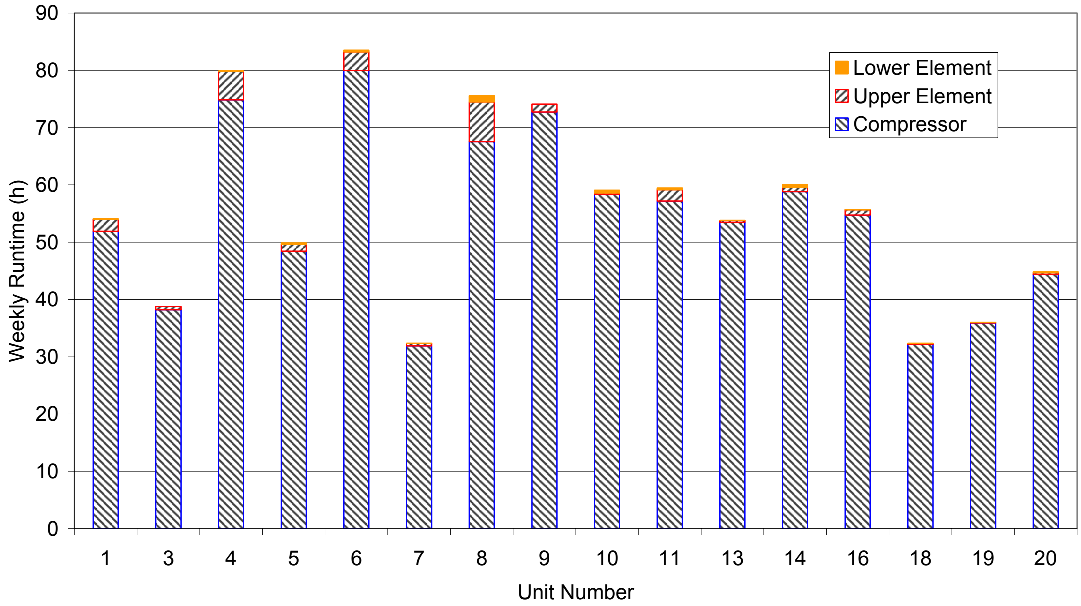

Fig. 7.10. Average runtimes of HPWH heating source (heat pump mode).

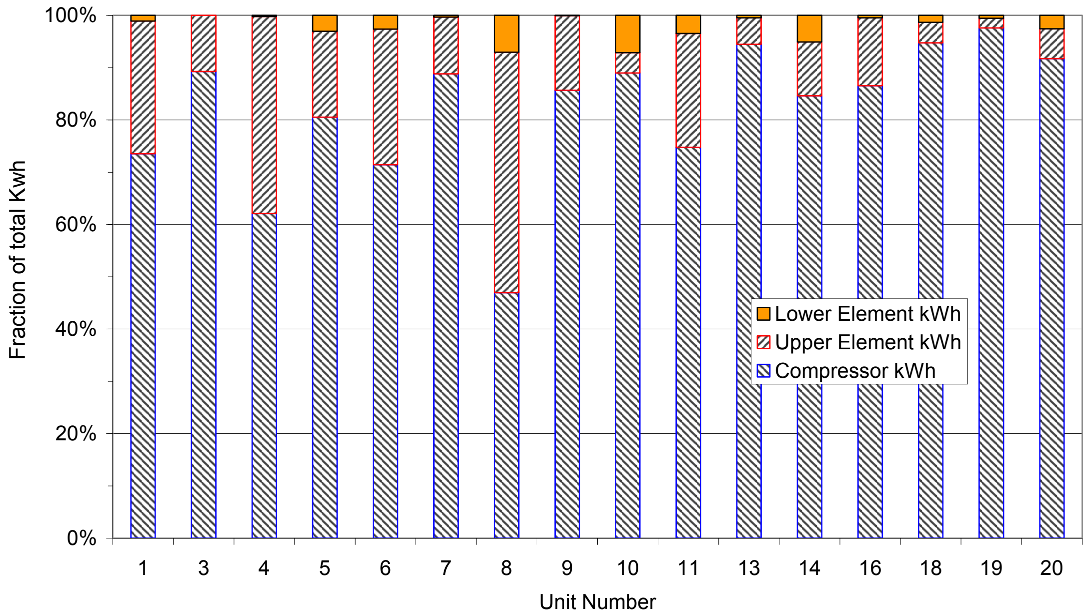

Fig. 7.11. Source of heat (weekly).

7.3 IMPACTS ON SPACE CONDITIONING AND CUSTOMERS

The field data collection system and approach was optimized so that the frequency with which sensors were scanned depended on how quickly the data might change. Ambient temperature, for example, would not change appreciably over a 2-second period data collection and storage; however, flow from the water heater to the house would change over a 2-second period due to small hot water draws. However, to understand how draws from the tank (2-second data) affect heat losses from the tank (based on ambient temperature, e.g. 10-minute data), we needed to be able to estimate ambient temperatures for all times based on measured 10-minute data. This was done by interpolation between scan intervals. Based on the data collected, we were able to evaluate a number of parameters including: Tank heat losses, hot water draw characteristics, evaporator cooling, and dehumidification.

Tank heat losses. Tank heat losses from the skin of the water heater were determined by analyzing the average tank temperature and ambient temperature data from several field test sites during times when there were no hot water draws and no heat input to the tank. These periods of time occurred generally after midnight. Based on analyses from several sites, we determined the average UA of the HPWH tank to be 3.2 Btu/h/ΔT where ΔT is the difference between the average tank temperature and ambient temperature. The average tank temperature was determined from measurements at six locations inside the tank, and the ambient temperature was logged from a thermocouple located on the instrument package and away from the influence of the water heater tank wall.

Hot water draw characteristics. We were able to determine the nature of all hot water draws from the 2-second data and the time that the scan was initiated or terminated. We defined the initiation of a draw event as a record occurring more than two seconds after the previous record in the datafile. We also defined the ending of a draw by looking to the subsequent record to determine if it occurred more than two seconds ahead in time. This process allowed us to define all draws including those as brief as two seconds in duration. Associated with each draw were temperatures into and leaving the hot water tank so that the thermal energy contained in each draw could be determined. Of perhaps greater importance, by pinning down the end time of each draw, the delivery temperature at the end of the draw could be determined. This was used to address the issue of differences in hot water runouts between the HPWH and resistance modes of operation.

Cooling impacts. The influence of HPWH operation on a dwelling's heating/cooling system depends on cooling at the evaporator of the HPWH and heat that is lost from the compressor shell as well as from other components of the vapor compressor system. We calculated the net cooling from the HPWH from first principles:

$$Q_c = \Delta U + \Sigma Q_d + Q_{loss} - E_{in}$$

Where,

Q_c = Net cooling (cooling at the evaporator – shell heat loss),
ΣQ_d = Total thermal energy in all draws from the tank,
Q_{loss} = Heat loss from the tank to ambient, and
E_{in} = Electrical energy to the tank.
ΔU = Internal energy change

Each of the parameters above (with appropriate dimensions) were calculated for every data download, and data downloads were done approximately weekly. Knowing the time logged at the beginning and end of a data download allowed us to calculate daily values for each of the parameters.

From the list of field test sites, we selected four sites each of which represented a different "type" of HPWH installation in order to examine evaporator cooling effects and draw characteristics. Each of the four sites represented a "typical" water heater installation (basement, utility room, etc.) for the climate and custom for that region. We chose the following sites:

Seattle, with the HPWH (Unit 8) located in an unconditioned basement;
Gainesville, with the HPWH (Unit 16) located in an unconditioned utility room adjacent to the garage;
Pensacola, with the HPWH (Unit 5) located in an unconditioned garage;
Danielsville, with the HPWH (Unit 13) located in a conditioned utility room (i.e. inside the house).

Results from analyses of data and calculations of tank heat loss and average ambient temperature for each of these sites are shown in Fig. 7.12 covering a time of 20 months. Each data point shown represents approximately a week of data. As the data show in this figure, the Seattle unit was installed and made operational in February 2001, the Pensacola unit came on line in April 2001, and the Gainesville and Danielsville units were installed at the end of May 2001. Although the Seattle basement HPWH is somewhat protected from outside temperature changes, the average ambient temperature in the Seattle basement dropped to 50°F in the winter. This fact combined with cold ground water temperatures in Seattle tended to reduce the capacity of the HPWH and therefore to increase the opportunity for backup resistance heating. In Pensacola during winter, the average ambient temperature dropped to near 50°F, but warmer ground water temperatures year-round there helped to reduce the need for backup heating. Ambient temperatures for the Danielsville HPWH located in the conditioned space remained in the comfort range throughout the experiment.

Tank heat losses in each of the four locations hovered around 4500 Btu/day and changed inversely with ambient temperatures. If a resistance water heater were manufactured using this tank, the heat loss would be about the same. Moreover, if that resistance water heater were to be tested according to the DOE 24-h Simulated Use Test Procedure, the 4500 Btu/day heat loss would represent 10% of the total energy needed to heat 64.3 gallons of water per day from 58° to 135°F and would produce an Energy Factor of about 0.9. The impact of heat losses from the tank depends on the location of the tank in the dwelling (garage, basement, etc.) as well as climate.

Cooling and dehumidification provided by the HPWHs at the four sites is shown in Fig. 7.13. The horizontal "resistance mode" line shows the periods of time when the HPWHs were operated as resistance water heaters, and in those intervals, cooling and dehumidification would be zero as shown.

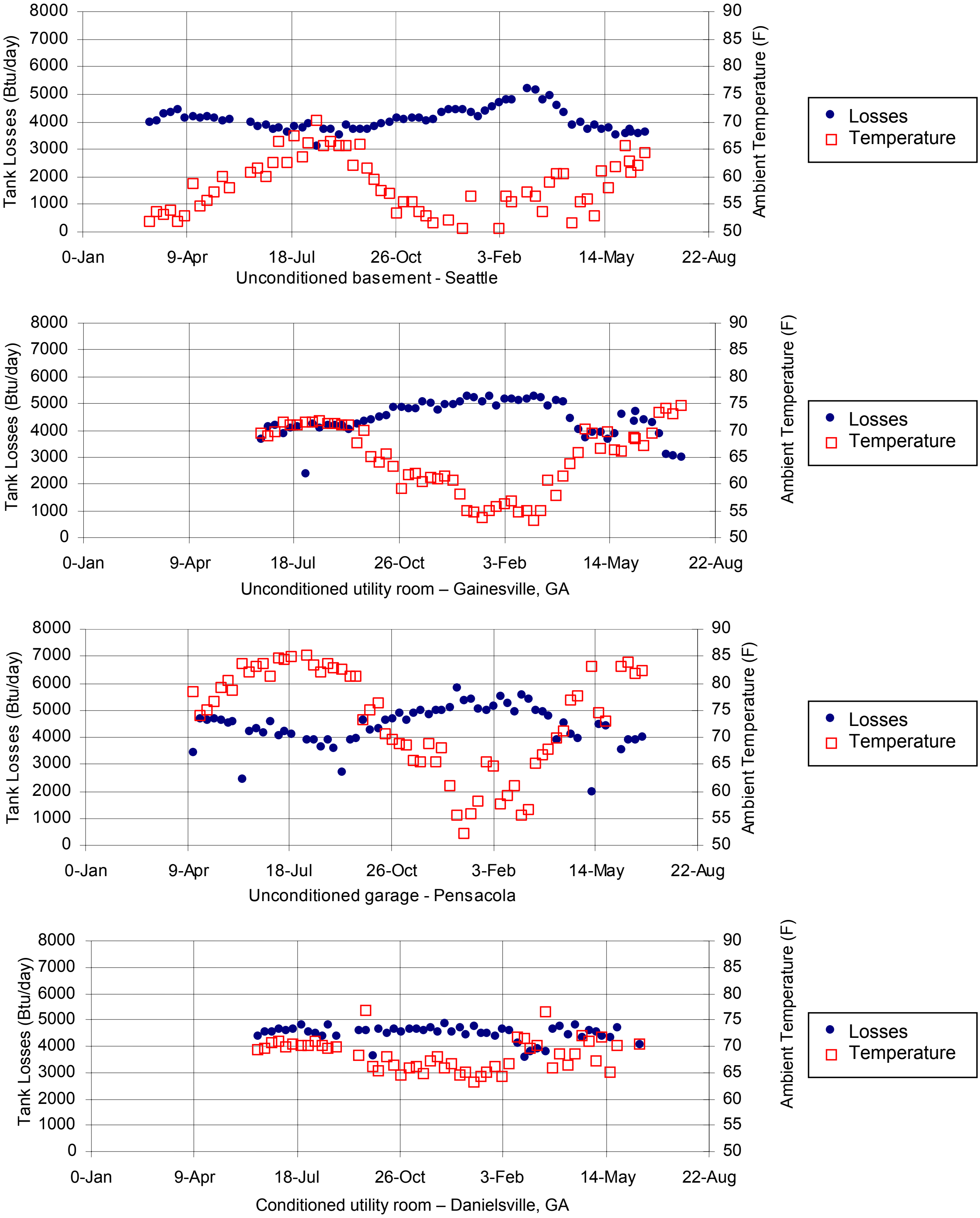

Fig. 7.12 Ambient temperatures and tank heat losses at four sites.

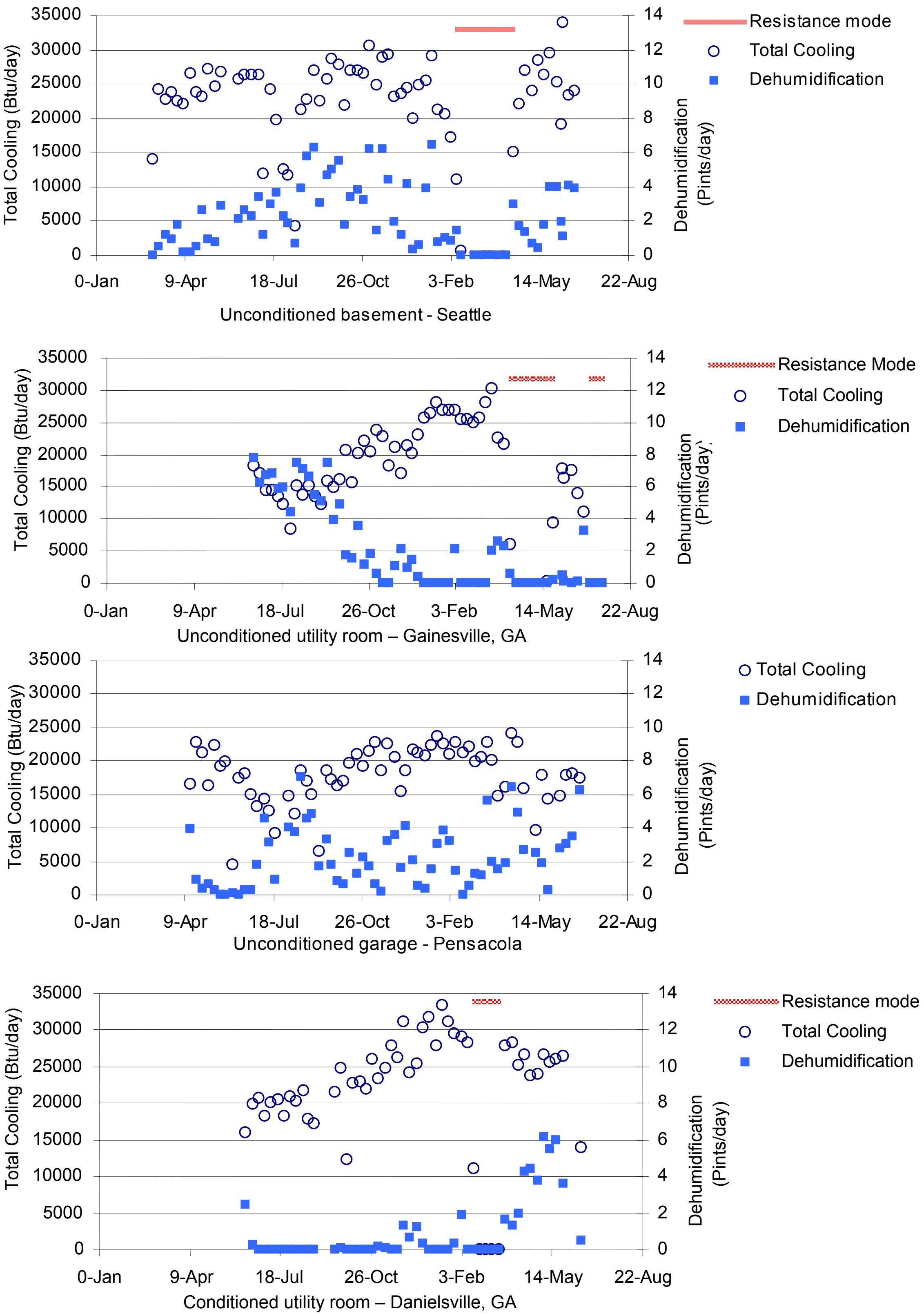

Fig. 7.13 Cooling and dehumification at four sites.

In Seattle, the HPWH provided basement dehumidification for the entire duration of the study as shown in Fig. 7.13. Dehumidification (in terms of pints of water removed per day) tended to be greater during the winter than in the summer. Total cooling provided by the Seattle HPWH evaporator tended to remain steady at about 25000 Btu/day. This constant cooling rate was the result of several factors that tended to offset one another: (1) the COP of the HPWH was lower in the winter than in summer due to cooler source temperatures for the HPWH evaporator, and (2) a significantly larger water heating load caused by the combination of increased hot water demand (i.e. more gallons of hot water produced) and the need to raise the water though a much higher temperature interval (much cooler incoming water temperatures during winter). The low COP tends to reduce the cooling produced at the evaporator; however the increased hot water energy demand tends to increase cooling at the evaporator. The two trends are offsetting in Seattle as shown in Fig. 7.13. We should note that the total cooling calculation includes both sensible and latent components. The latent component results in dehumidification of the surrounding air. In Seattle with greater dehumidification appearing in winter, there is a corresponding drop in sensible cooling. Consequently, a wintertime drop in basement temperature due to sensible cooling would be ameliorated.

In Gainesville (unconditioned location), lower winter ambients in the unconditioned utility room tended to reduce the COP of the HPWH in the same manner as for the Seattle example. The reduction in COP, however, was not as great as for Seattle. The increase in hot water energy (product of hot water volume and temperature increase between cold water entering the HPWH and hot water leaving the HPWH) from summer to winter was greater than in Seattle. Consequently, the tradeoff between COP reduction and additional hot water energy production was not balanced. The HPWH produced more total cooling during winter than in summer as shown in Fig. 7.13.

In the Danielsville site where the HPWH was in conditioned space, the COP of the HPWH actually increased from summer to winter although the evaporator source temperature (room temperature) remained the same for summer and winter. The increase in COP was attributed to lower average condensing temperatures caused by cooler water to the tank during winter than during summer. As in the case of the unconditioned Gainesville site, the hot water energy delivery increased from summer to winter due to higher hot water use (gallons) as well as more energy supplied in each draw by the HPWH (higher temperature lift to reach the thermostat setpoint). An increase in both COP and heating energy worked together to increase the cooling load on the house during the winter as shown at the bottom of Fig. 7.13. Since the HPWH is in the conditioned space, this cooling load must be met with the heating system of the house. A heat pump provided all of the space conditioning in this Danielsville house.

We examined the impact of the HPWH evaporator load on the heating system of the house over a range of average heating season performance factors (HSPF) for the space conditioning heat pump. We assumed that heat losses from the tank were 13% of the heat energy delivered in hot water by the HPWH; the 13% figure was based on calculations from measured data at the Danielsville site and can be determined from Figs. 7.12 and 7.13. We further assumed that the COP of the HPWH during winter was 2.5. This figure was based on measured data from the study for the Danielsville site. The overall impact of having the HPWH in the heated utility room is shown in Fig. 7.14.

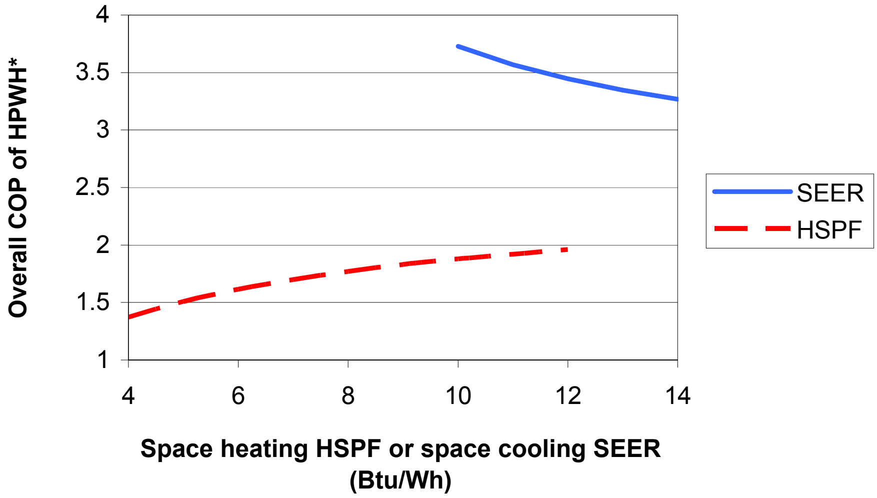

Fig. 7.14. Efficiency impact of HPWH (* includes space impacts) in conditioned (heated and cooled) utility room – Danielsville

As shown, the COP of the HPWH depends on the HSPF of the space heating heat pump. Although the HSPF of the heat pump in the Danielsville home was not measured, Fig. 7.14 indicates how the COP of the HPWH depends on the efficiency of the space heating system. During the cooling season, the HPWH located in conditioned space provides cooling as well as water heating. To account for this summer benefit, we evaluated this cooling benefit as a function of the efficiency of the space cooling system. The results of this analysis are shown in Fig. 7.14. The value of the cooling benefit provided by the HPWH depends on the efficiency of the space cooling system. We assumed that cooling from the HPWH evaporator was useful, and that if it were not for the HPWH, that amount of cooling would need to be provided by the home's air conditioning system. We conclude from this analysis that were it not for the cooling provided by the HPWH evaporator, cooling would be provided by the space cooling system at the seasonal energy efficiency ratio (SEER) levels shown.

7.4 DEHUMIDIFICATION

The total cooling shown in Fig. 7.13, is made up of sensible cooling and latent cooling or dehumidification. Throughout the field study, we measured condensate produced at the evaporator of each field test unit. Figure 7.13 shows the measured dehumidification from the four field test sites described earlier. In Seattle, basement dehumidification occurred for most of the winter and summer. The data show that the dehumidification rate was generally higher during the winter and less in the summer months. The only periods when dehumidification was not present was during times when the HPWH was switched to the resistance mode. For the unconditioned utility room in Gainesville, dehumidification occurred during the summer following the installation. Relatively dry winters, along with the HPWH in a closed off utility room, tended to eliminate the production of condensate during the winter. One can see from the figure that following Feb-4, condensate production began once more. Switching the HPWH to the resistance mode for much of that spring eliminated further production of condensate.

In Pensacola, condensate was produced for most of the year. However, since this unit was located in a garage (unconditioned), there would likely be little benefit ascribed to dehumidification there.

The conditioned utility room HPWH in Danielsville presents an interesting dehumidification pattern. Dehumidification tended to occur only during the swing seasons. Apparently interior conditions were dry enough in winter so that condensate did not form. The lack of condensate production in summer was due to the air conditioning system of the dwelling that dropped the dew point of the air inside the house below the temperature of the HPWH evaporator coil. This eliminated any opportunity for the HPWH evaporator to produce condensate. All of the cooling produced by the evaporator went into sensible (and beneficial) cooling of the dwelling.

7.5 HOT WATER DEFICITS

Hot water deficits (runouts) are draw events when at the end of a draw, the hot water temperature has become too low to be practical. We defined a runout as a draw event having an ending temperature 105°F or less. To be sure, there are many draws, particularly short ones in which the ending temperature never reaches 105°F, and a customer would not term them a runout. However, by being able to conduct the experiment with the HPWH in the resistance mode as well as in the heat pump mode, changes in runouts could be determined. The seemingly slow recovery time of the HPWH suggests that hot water runouts could increase in a switchover from resistance to heat pump water heating.

To test this hypothesis, we used the four test sites described earlier, and we examined all of the data for the study to determine (1) the total number of draws for each period, and (2) the number of potential draw runouts. The results from these analyses are shown in Fig. 7.15. In three of the test sites (Seattle, Gainesville, and Danielsville, we operated the HPWH for some time in resistance mode. As can be seen in Fig. 7.15, the number of hot water runouts did not appear to be associated with the operation of the HPWHs. The fact that the upper resistance element was retained in the design of the HPWH and used in the same way as in a conventional resistance water heater for quick recovery seemed to eliminate the potential for increased hot water runouts. The only trend evident in some cases was an overall increase in hot water runouts during the winter, especially as seen in Pensacola. This seasonal trend was not unexpected.

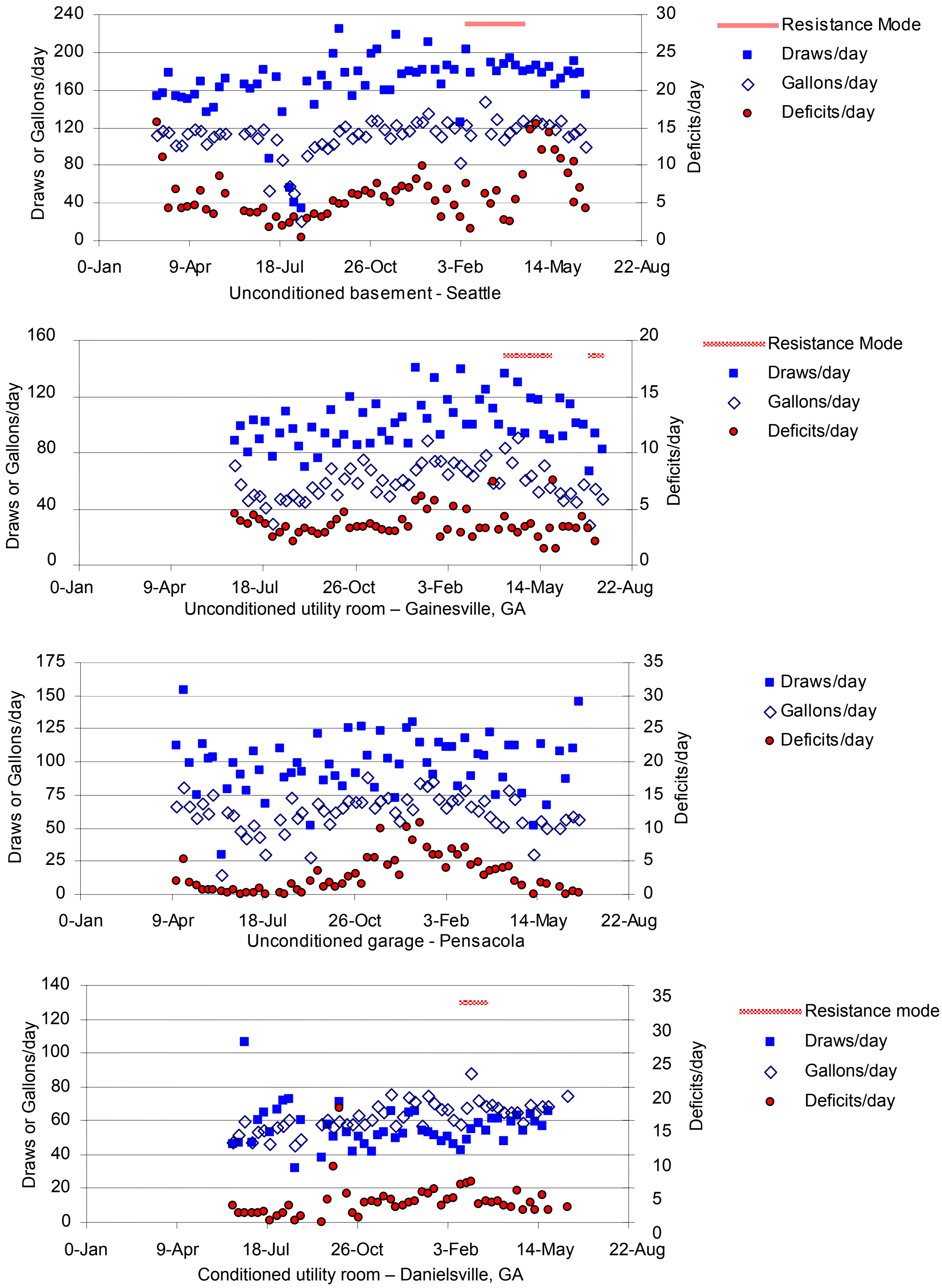

Fig. 7.15. Hot water draw patterns at four sites.

SUMMARY AND CONCLUSIONS

The first year of a national field study of the “drop-in” residential heat pump water heater was completed. In most cases, residential sites for the study were existing single family homes where the test HPWH replaced a resistance water heater. Rather than doing a “before/after” field study, we opted to replace the existing electric water heater with a HPWH that had not only been fully instrumented, but that also had the capability of being operated as a conventional resistance water heater. Measuring the performance of the unit in the heat pump mode and in the conventional resistance water heating mode allowed us to evaluate changes in performance of the unit as well as changes that the switchover would have on hot water consumption patterns, electric energy use patterns and loads placed on the space conditioning systems for both summer and winter.

No effort was made to “preselect” locations within each house for the HPWH. We made site selections based on typical locations for the water heater in the various climates. For example, in the Deep South, electric water heaters are typically located in garages, whereas in the northeast, electric water heaters are often located in basements. In some locations, electric water heaters are located in closets or in utility rooms in the conditioned space. Sites were selected to cover the range for electric water heater locations.

Over the first year of the study, we found that, on average, the HPWH reduced the electric energy needed for water heating by 55%. The electric energy savings ranged from as high as 62% for a unit located in a Florida garage to as low as 41% for a unit located in a basement in Washington State. These are reductions in the total electric energy consumption of the water heater, and in the case of the HPWH, include energy used by the compressor, controls and any use of backup resistance heating elements.

We operated a number of units alternatively as resistance water heaters then in the heat pump mode for periods in the summer and in the winter to determine impacts on diversified peak demand. The morning peak demand reduction was found to be about 1.5 kW and the evening peak demand reduction averaged 0.8 kW with the heat pump mode. In the summer, this mode also shaved about 0.4 kW from the afternoon peak of a conventional resistance water heater while at the same time, providing cooling (and possible dehumidification) from the evaporator. If the HPWH were located in the conditioned space, this summertime cooling would offset a portion of the cooling demand of a home’s air conditioning system thereby producing additional savings and demand reduction.

We mined the data from the field study to estimate cooling at the evaporator. This is energy that is removed from the surrounding space as the compressor operates. Evaporator cooling increased with ambient temperature, with hot water production and with lower condensing temperatures. In four of the test sites analyzed in detail, evaporator cooling amounted to about 30,000 Btu/day. Assuming that the HPWH has a 50% duty cycle, the cooling rate is equivalent to 0.2 tons of air conditioning and dehumidification.

We measured the amount of condensate removed from the evaporator and found dehumidification rates from 0 to 6 pints/day of water. While not significant in some settings, this amount of moisture removal taking place over the long runtime of the HWPH is notable. As expected, condensate production was high where humidities were high (e.g. Pensacola garages, Seattle basements and with the HPWH located in other unconditioned spaces. Interestingly, we found for the unit located inside the conditioned space in Danielsville, there was no dehumidification removed by the HPWH due to operation of the air conditioning system. In this case, however, sensible cooling would constitute all of the total cooling and be beneficial.

Finally, we examined changes in hot water draw patterns, frequency and any increase for runouts with the heat pump mode. We found that switching to this mode did not measurably affect the hot water draw volume or patterns. Based on the data, we found no differences in the frequency of hot water runouts for the two modes.

In summary, the field tests have shown that the "drop-in" design can work effectively and efficiently in all of the locations that were tested including garages, basements, and utility rooms. The energy impact of the HPWH located in a conditioned space depends on the climate (heating vs. cooling demand) as well as the efficiency of the space conditioning systems. We hope that this study conducted in real-world settings and occupied homes has answered questions and provided useful data needed to see a meaningful market materialize for the HPWH.

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