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DURABILITY TESTING OF A DROP-IN HEAT PUMP WATER HEATER

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CONTENTS

FIGURES

TABLES

DURABILITY TESTING OF A "DROP-IN" HEAT PUMP WATER HEATER

VAN D. BAXTER
R. L. LINKOUS

ABSTRACT

Ten prototype "drop-in" heat pump water heaters (HPWHs) were placed in an environmentally controlled test facility and run through a durability test program of approximately 7300 compressor duty cycles (the actual cycles accumulated ranged from 6677 to 7950 for the ten units). This durability test was designed to represent approximately 10 years of normal compressor cycling to meet the hot water needs of a residence. The heat pump portion of the HPWHs experienced no compressor, evaporator fan, or power relay failures during the durability test run. One thermal expansion valve had to be replaced because of a blocked bleed port. Two of the test units experienced refrigerant leaks that occurred in pressure transducer fittings added to the circuit as part of the testing program. Since these fittings would not be included in production HPWH units, this problem is not considered a reliability concern.

The first-generation control system proved to be the least reliable component of the prototype HPWH units. Each HPWH controller included four temperature sensors to monitor key control parameters. Out of 40 total sensors, 16 failed during the durability program. These failures were due to problems with spliced joints in the sensor lead wires.

Efficiency measurements on all units showed that the prototype HPWH is at least twice as efficient as conventional electric resistance water heaters. Energy factors measured at the end of the durability run, at ambient conditions close to those prescribed by the U.S. Department of Energy Simulated Use Test, ranged from 1.8 to 2.1. This figure is comparable to average efficiencies achieved by several field test units, but lower than the 2.47 value achieved by an early prototype design in 2000. Based on the data available, there is no evidence that strongly suggests that any of the HPWHs suffered any performance degradation as a result of undergoing more than 7000 compressor cycles in this durability test.

1. INTRODUCTION

"Reliability" is defined as the probability that any product will function without failure under given conditions for a given period of time. In the case of a heat pump water heater (HPWH) or any water heater, the function is the delivery of hot water without falloff in efficiency and without component failure. To define the "given period of time," we examined the lifetime of a typical electric resistance water heater (EWH). *Appliance Magazine* estimated in 1999 that the "initial purchaser" of a conventional residential EWH would retain the appliance for about 7 to 18 years (*Appliance Magazine* 1999); in 2001 this estimate had changed slightly to 7 to 15 years (*Appliance Magazine* 2001). For gas water heaters (GWHs), these time periods were 6 to 13 years for 1999 and 7 to 10 years in 2001. These figures suggest that customers' expectations for the average lifetime of any "drop-in" HPWH that replaces a conventional water heater would be no less than 7 years and probably more in the range of 10–12 years. Therefore we determined to conduct accelerated life, or durability, testing to simulate the equivalent of 10 years of use as representative of life expectancy requirements for an HPWH.

Heat Pump Water Heater Description

The HPWH examined in this study was intended to be a "drop-in" replacement for a conventional EWH. It is pictured in Fig. 1.1 and shown in a schematic cut-away view in Fig. 1.2. As used in this report, "drop-in" means that the HPWH can be installed virtually anywhere an EWH can be installed, using essentially the same installation procedures. It is likely that most HPWH installations will require slightly more time than EWH installations because of the requirement to provide for drainage of condensate from the HPWH evaporator.

Development of this HPWH design was a collaborative effort between Arthur D. Little, Inc. (ADL), EnviroMaster International (EMI, the manufacturing partner), and Oak Ridge National Laboratory (ORNL) with sponsorship from the U.S. Department of Energy (DOE), the New York State Energy Research and Development Authority, and the California Energy Commission (CEC). The design is based on a patented concept originally developed by ADL (Dieckmann et al. 1999a, 1999b). EMI developed an original prototype production design based on the ADL concept in 1999. This design was subsequently refined and considerably improved by ORNL, ultimately achieving an energy factor (EF) rating of 2.47, based on the DOE Simulated Use Test procedure (*Federal Register* 1998), in early 2000 (Tomlinson 2000). Ten units of this final prototype were built by EMI and delivered to ORNL in late summer of 2000 for the durability test program discussed in this report. Another 18 units were built and sent to ORNL for a DOE national field test program that will be described in a separate report (Murphy 2002). In addition, approximately 20 units are to be built for a field test program to be conducted in California by ADL under CEC sponsorship.

Fig. 1.1. Photograph of the prototype "drop-in" heat pump water heater unit that was the subject of the durability testing discussed in this report.

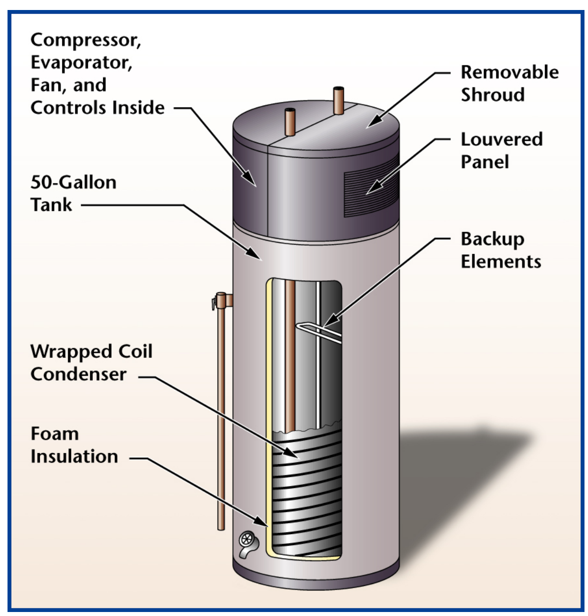

Fig. 1.2. Cutaway schematic view of the HPWH.

The HPWHs in question are about the size of a vertical cylinder 5 ft high and 2 ft in diameter. A small (about 1/4 hp) air-to-water vapor-compression heat pump unit, which uses R-134a as the refrigerant, is located on top of a nominal 50-gal water tank (45.9 gal actual capacity). Heat to the evaporator is provided by ambient air. The unit's condenser coil is wrapped around the bottom third of the water tank to provide heat to the water. Figures 1.3 and 1.4 show the compressor and evaporator sections, respectively. The condenser is composed of a copper tube, slightly flattened on one side and wound tightly around the bottom third of the water tank. A highly conductive heat transfer paste, or mastic, is spread between the condenser tubing and the tank wall during the winding process to ensure good thermal contact along the entire condenser length. Water is heated by the condensing refrigerant via conduction through the tank wall. Blown polyurethane foam insulation between the tank wall and the outer water heater metal jacket helps to hold the condenser in place. The condenser tubes are wrapped with plastic to prevent the insulation from getting between tubes and tank wall during the foam blowing process. Figure 1.5 shows a section of the condenser wrap located at the bottom electric element access cover. By design, the small compressor takes 6–8 hours to heat up a tank of water from a cold start or about 1.5–2 hours to recover a hot tank after a 10.7-gal water draw.

The HPWHs used in the durability tests were each equipped with a solid-state, microprocessor-based control system based on a design that was originally developed by ADL under CEC sponsorship. Figure 1.6 is a photograph of the first-generation version of the control board as used in the durability test units described in this report. A programmable microprocessor chip, which contains the unit control program, can be seen near the middle of the board (partially obscured by some wiring). Along the right and bottom sides of the board can be seen six large black objects. These are the 240-VAC, 30-amp relays that switch power to the compressor, fans (two), electric elements (two), and a condensate re-evaporation heater that is part of an optional condensate management system (not used in this durability test). The control system requires seven inputs (listed in Table 1.1).

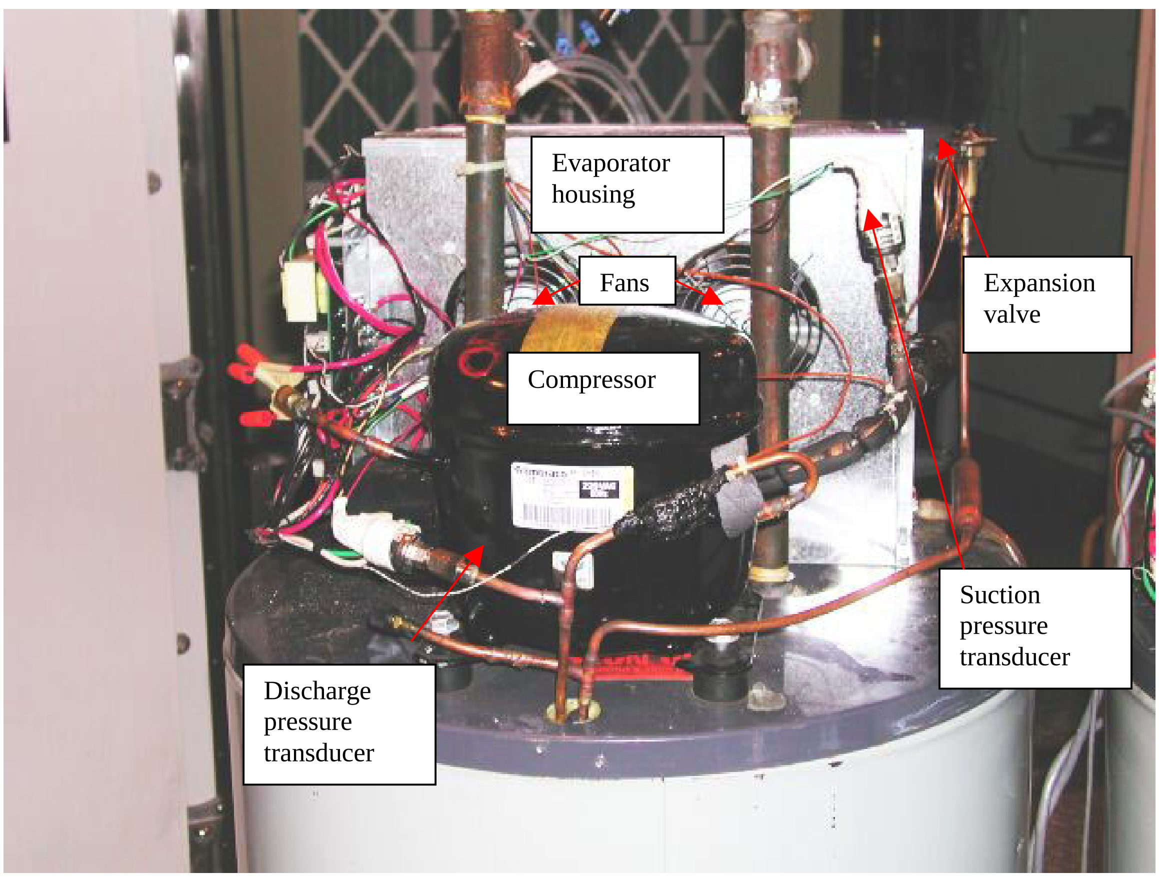

Fig. 1.3. Photo of heat pump section of HPWH showing compressor (foreground), rear of evaporator housing (background), and thermal expansion valve (right-hand side).

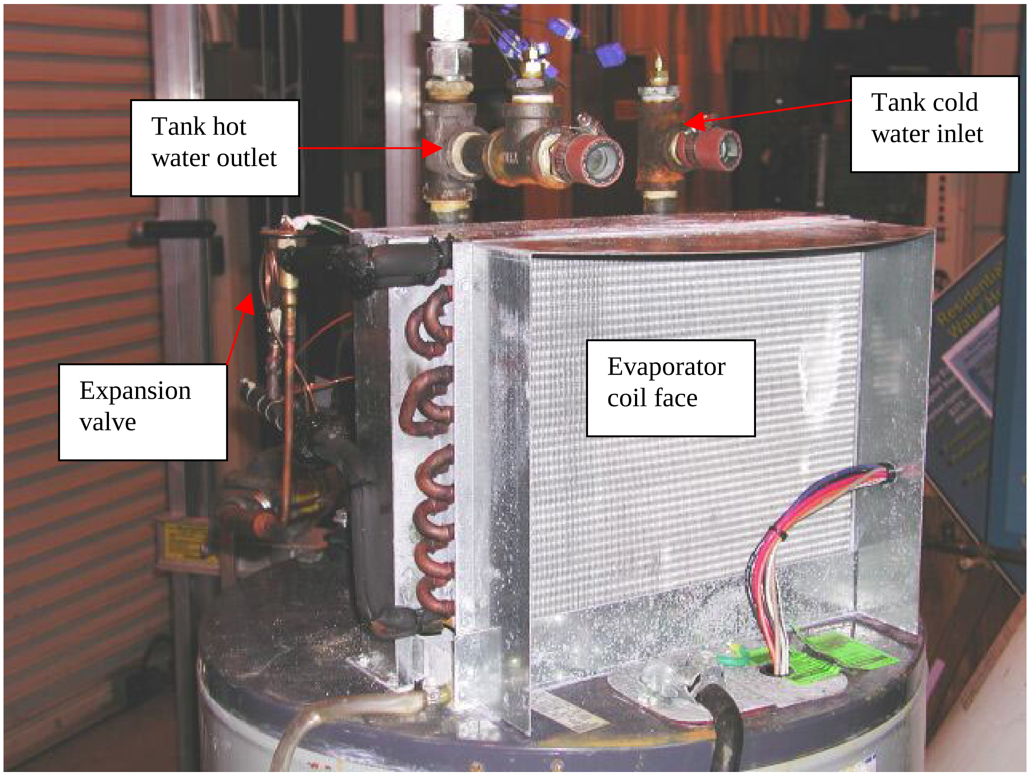

Fig. 1.4. Photo of heat pump section of the HPWH showing front of evaporator with thermal expansion valve to the left.

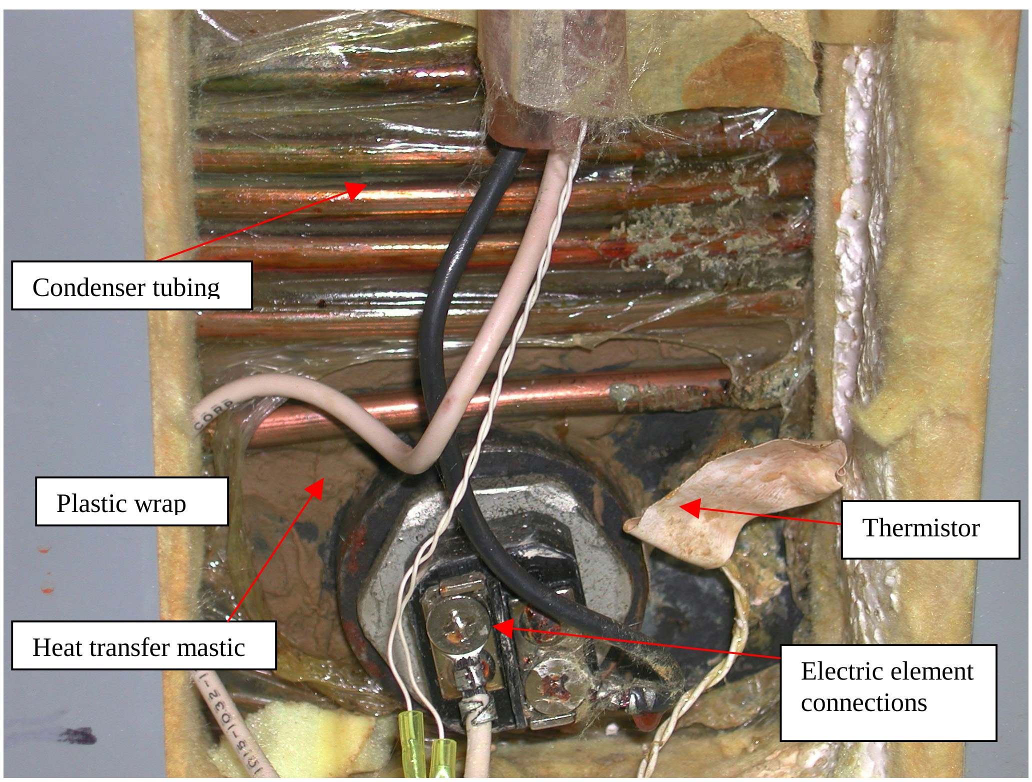

Fig. 1.5. Photo of section of HPWH condenser near bottom of tank in vicinity of lower electric element.

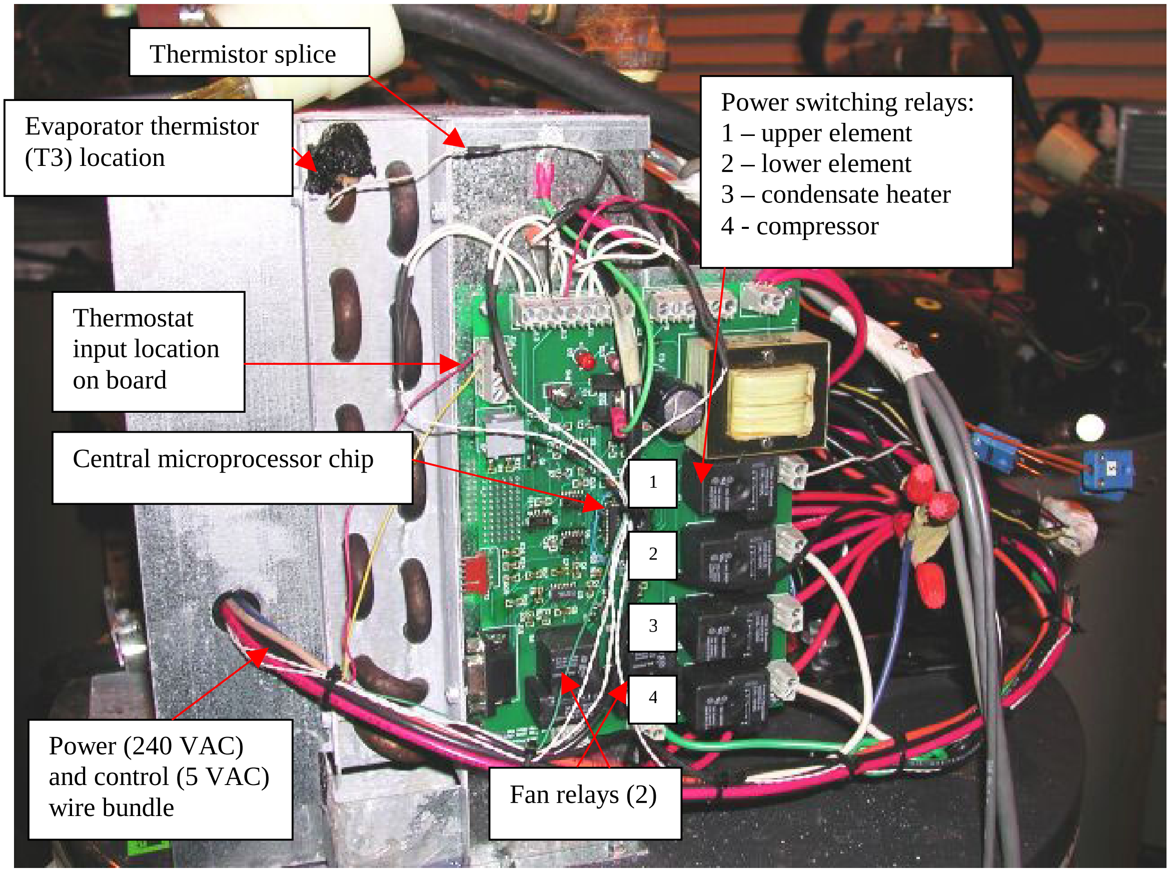

Fig. 1.6. Photo of first-generation HPWH solid-state control board. The microprocessor is in the middle, partially obscured by the wire bundle.

Table 1.1. Analog and digital inputs to heat pump water heater control system

Input	Device	Units
Dial thermostat (user input)	Var. resistor	Ω
Lower tank temperature, T1	Thermistor	Ω
Upper tank temperature, T2	Thermistor	Ω
Evaporator temperature, T3	Thermistor	Ω
Compressor discharge temperature, T4	Thermistor	Ω
Condensate present in pan	Float Switch 1	Open/Closed
Condensate pan overflow	Float Switch 2	Open/Closed

An HPWH mode switch was included on the first-generation control board. It enabled switching between HPWH mode and conventional EWH mode.

Three light-emitting diodes (LEDs) are provided to indicate when the controller is calling for compressor (yellow light), lower element (red), or upper element (green) operation. The durability and field test HPWH control systems were equipped with RS-232 and RJ-45 communication ports for programming, debugging, and setting operating parameters.

The control system is powered from the same 240-V, 60-Hz, single-phase source as the HPWH. The controller does not permit the upper element and lower element to energize simultaneously. It also prevents the compressor and the optional condensate management system's (CMS's) condensate re-evaporation heater from energizing when either electric element is energized.

Heat Pump Water Heater Control Sequence

The software embedded in the microprocessor provides control of the HPWH and determines—based on the values of the inputs (Table 1.1), the system operating parameters (Table 1.2), and the control logic—whether the water heater operates as a

- conventional EWH or
- HPWH

On the first-generation control system used in the durability test units, the software control program sampled each input signal one time and then made control decisions based on those instantaneous values. This feature provided for relatively simple program logic but would render the HPWH vulnerable to erratic operation in the event of random aberrant signal inputs caused by electronic noise, etc. As noted in Sect. 4, random noise events were suspected of causing some operational problems for the durability units.

The operating parameters were embedded into the memory of the microprocessor. Nominal values for the operating parameters are listed in Table 1.2.

Table 1.2. Operating parameter descriptions and nominal values for heat pump water heater (HPWH) control system

Variable	Description	Value
HPWHSet	Set-point temperature of the HPWH	User set
TempLow	The lower evaporating temperature limit for heat-pump operation.	25ºF
TempFan1	The evaporating temperature that shuts off one fan.	60ºF
TempFan2	The evaporating temperature that shuts off both fans.	65°F
MinDisc	Minimum compressor discharge temperature that is required for continued operation of the HPWH after start-up.	100°F
MaxDisc	Maximum allowable compressor discharge temperature.	220°F
HysUpper (HU)	The dead band for the upper tank temperature (+0/–HU)	27ºF
HysLower (HL)	The hysteresis for the lower tank temperature (+0/–HL)	5ºF
HysMode (HM)	The hysteresis for other parameters. Includes TempLow (+HM/–0), TempFan1 (+0/–HM), and TempFan2 (+0/–HM)	10ºF
TempDiff	Lower tank set-point temperature offset from the HPWH set-point temperature.	10ºF
MaxHWT	Maximum lower tank water temperature for heat pump operation	140°F

Heat Pump Water Heater Mode

The control system is designed to maximize the use of the heat pump to promote maximum water heating efficiency. Any time the lower tank temperature falls below a point defined by the relation [HPWHSet – TempDiff – HL], the heat pump is activated (compressor and fans) and the yellow indicator light (LED) is illuminated. Once the lower tank temperature has reached the lower tank set-point temperature [HPWHSet – TempDiff], the compressor and fans will turn off (normal stop). Seven conditions will cause the control system to shut off the heat pump (compressor and fans) prematurely (before the lower tank set-point temperature is reached).

1. The evaporating temperature falls below 25°F [TempLow], at which point frost may begin developing on evaporator coils. The lower electric element is energized to continue heating water. The heat pump will restart once the evaporating temperature reaches 35°F [TempLow + HM] if the lower element has not satisfied the thermostat.
2. The upper tank temperature falls below the point defined by [HWPHSet – HU]. The upper element is energized to heat the upper portion of the tank. The heat pump restarts once the upper tank temperature (T2) reaches the HPWH set point.
3. The lower tank temperature has reached 140°F [MaxHWT], but the HPWH set point has not been reached. In order to limit the discharge pressure and temperature, the heat pump is shut off. The lower element is energized until the HPWH set-point temperature is satisfied.
4. If the CMS option is active and the CMS float switch 2 is at an overflow condition, the heat pump is shut off. The lower element is energized to heat the tank water. The heat pump will not be reactivated until the condensate in the evaporator drain pan is removed (see CMS description later in this section).
5. The HPWH mode select switch is in the EWH position. The unit will operate as a conventional EWH until the switch position is changed.
6. Anytime the compressor discharge temperature is not between 100°F [MinDisc] and 220°F [MaxDisc] following a 3–6 min period after start-up, the heat pump will be switched to operate in the EWH mode. Very cold water entering the tank is the most likely cause of this situation (compressor failure is another). The lower element will energize to begin heating the bottom of the tank. After a short time delay (about 3 min in the first-generation control design), the control system will attempt to restart the heat pump. This cycle will repeat until the tank water temperature is high enough to sustain heat pump operation (or until the lower set point is reached in the event of compressor failure).
7. If the compressor discharge temperature exceeds 220°F [MaxDisc], the compressor will be shut off to avoid operation with excessive refrigerant temperatures. The lower element will be energized to continue heating the tank water. After a delay of 3–6 min, the compressor will be restarted if the lower set point has not yet been reached.

If the evaporating temperature becomes too high, the compressor is at risk of becoming overloaded (because the density of the suction gas increases). In order to prevent high evaporating temperatures, the control system employs the following sequence.

1. If the evaporating temperature exceeds 60°F [TempFan1], one of the two evaporator fans is turned off. Normally, this will cause the evaporating temperature to drop. The fan will be turned on again once the evaporating temperature drops below 50°F [TempFan1 – HM].
2. If, after one fan is turned off, the evaporating temperature continues to rise and exceeds 65°F [TempFan2], the second fan is also turned off. Natural convection then becomes the only mode of heat transfer (on the airside) of the evaporator. The second fan will be turned on again once the evaporating temperature drops below 55°F [TempFan2 – HM].

An algorithm in the control system delays compressor restart for 3–6 min after any heat pump shutdown. This may create a situation in which the lower element energizes (and remains on for the remaining delay period) if a water draw occurs immediately following the shutdown.

Electric Water Heater Mode

Any time the lower tank temperature falls below the temperature [HPWHSet – TempDiff – HL], the lower element is energized and remains on until set-point temperature is reached. If the upper tank temperature falls below the HPWH set-point temperature [HPWHSet – HU], the lower element is turned off and the upper element is energized. The upper element remains on until the upper tank temperature equals the HPWH set-point temperature [HPWHSet]. The lower element is then reenergized and continues heating the lower portion of the tank until the lower set-point temperature is reached.

Condensate Management System Option

The control system includes an optional provision to control the level of condensate in the heat pump evaporator drain pan. This option would allow the HPWH to be installed without requiring plumbing to an external drain to remove evaporator condensate.

There are two condensate levels that the CMS responds to—a condensate-present level and a condensate-overflow level. When the CMS indicates there is condensate present (float switch 1 is closed), the control system will energize the CMS heater (unless either of the electric elements is on). The heater will remain energized until condensate float switch 1 opens, indicating no more appreciable condensate. The CMS heater can operate while the heat pump is operating. However, if the heat pump is off and CMS is still required to evaporate condensate, one fan will be turned on while the heater is on.

If the condensate reaches the overflow level (float switch 2 is closed), the HPWH will switch to EWH mode and continue heating the tank until the lower tank set-point temperature is reached. Then the control system will energize the CMS heater and one fan. The heater and fan will remain energized until float switch 1 opens, indicating that there is no longer any appreciable condensate.

The CMS option was not included in the controls for the units examined in this durability test run.

2. TEST PLAN DEVELOPMENT

A key measure of reliability is the ability of the system to survive accelerated life tests that compress a number of real-world cycles into a manageable set of laboratory tests. Prior testing of an early prototype of the subject drop-in HPWH according to the DOE 24-h Simulated Use Test (a reasonable representation of real-world hot water consumption patterns) indicated that the heat pump would cycle on twice during a 24-hour test period. The lower thermostat would turn the heat pump on after the first hot water draw of the test, and it would remain on for an extended period to meet the remaining five hot water draws prescribed by the DOE protocol. During the subsequent recovery period, testing showed that the heat pump turned on once more as internal tank temperatures equilibrated and to replace tank heat losses over this period. Therefore, assuming the HPWH will undergo an average of two cycles per day in a representative residential application, over a 10-year lifetime, the total number of compressor cycles will be (10 years)(365 day/year)(2 cycles/day) = 7300 cycles. We therefore prepared a plan to conduct accelerated testing so that in about 200 days of testing, the HPWH compressor could be subjected to 7300 cycles.

This estimate of 7300 cycles for a 10-year life was prepared in the absence of any significant data from actual residential applications of the HPWH. In the interim, however, some information has become available from the DOE national field test program that ran more or less concurrently with the durability testing effort (Murphy 2002). An examination of the cycling history for three of these units indicates that they experienced anywhere from 14 to 26 compressor cycles per week, with a slight dependence upon hot water usage (i.e., the greater the usage the greater the number of cycles) (see Table 2.1). For units with high water use, there can be a number of large hot water demand instances each week when the upper element is used for quick recovery of the top portion of the tank. When this happens, the compressor will shut off until upper tank recovery is complete; then it turns back on to finish heating the remainder of the tank, leading to a greater number of compressor cycles. Over a 10-year service life, this range of weekly cycling rates would equate to a total of about 7,300 to 13,500 compressor cycles. For sites with low to moderate hot water demand, the 7300-cycle estimate used for the durability testing seems to reasonably represent about 8–10 years of compressor duty cycles. For sites with heavy usage, 7300 cycles may represent more like 5–9 years of compressor cycles.

Table 2.1. Average weekly compressor duty cycles for three field-test HPWHs compared with weekly hot water demand

Approximate weekly water use (gallons)	Weekly compressor cycles	Total compressor cycles for 10 years of operation at average weekly rate
260	14–18	7,280–9,360
490	15–19	7,800–9,880
630	15–26	7,800–13,520

Source: Murphy 2002.

A number of components of conventional EWHs are retained in the “drop-in” HPWH design. These include the tank itself, upper and lower elements, anode rod, high-temperature cutout switch, pressure/temperature (P/T) relief valve, and tank fittings. Based on the fact that the HPWH design does not affect these components, we anticipate that the failure rates of these components will be the same as for conventional resistance water heaters. In some cases, the lifetime of a component may actually be longer when it is used in the HPWH design. The lower element, for example, may last longer for the HPWH since it is not expected to operate very often barring compressor failure. The upper element may operate more or less frequently in the HPWH design than it would in the conventional water heater design, but this depends on the hot water draw usage pattern (total water use and time of use) for any given installation.

As a starting point for the performance and durability test protocols, we characterized the reliability of each of the major components of the HPWH as being either (1) an issue (needing to be addressed in the durability testing protocol) (2) a possible issue, or (3) or a non-issue (component's performance would not be changed if used in the HPWH design). These characterizations are shown in Table 2.2.

Table 2.2. Heat pump water heater performance issues

Significance	Component	Comments
Non issue	P/T valve	Same function as for electric water heater (EWH)
	High-temperature cutout switch	Same function as for EWH
	Evaporator coil	No moving parts
	Electric heating elements	Operating hours not expected to be significantly greater than for EWH for typical applications
Possible issue	Tank corrosion	Possible breaching of tank lining during brazing process to attach ends of condenser wrap to tank wall
	Tank scaling	Potential for scale formation on heated tank wall
	Refrigerant degradation	Elevated compressor discharge temperatures; possible R-134a breakdown into fluorine-containing compounds (hydrofluoric acid); coking
	Tank insulation degradation	Tank insulation sees hotter temperatures than for EWH
Issue	Condenser bonding	Possible separation from tank; higher condensing temperatures are produced
	Compressor reliability	Reduced voltage starting; 10-year equivalent cycling
	T-stat reliability	Reliability indicated through cycling tests
	Evaporator fan lifetime	10-years of continuous operation = 87,600 h; advertised fan lifetime is 65,000 h
	Controls	Tests for consistent operation

Expected Ambient Operating Condition Envelope

The drop-in HPWH is designed to perform over a specified range of ambient temperature and humidity, incoming cold water temperature, and hot water delivery temperature. Although exact operating conditions may differ somewhat from the following, we anticipate that these conditions will be close to those experienced in most HPWH locations.

Ambient dry bulb temperature:	40 to 120°F
Ambient RH:	50 to 90%
Incoming cold water:	45 to 70°F
Hot water delivery temperatures:	up to 140°F (although unit should be able to function with thermostat at highest setting of 158°F)

The extremes of the ambient temperature range cover temperatures that could be found in garage installations during the summer and winter in various locations in the United States. The 1% design dry bulb temperature (DBT) for Las Vegas, for example, is 108°F; temperatures in a south- or west-facing closed garage may be 10–12°F, or more, higher during solar noon. Design DBTs are lower for other U.S. locations. A water heater should not be located where the ambient temperature could fall much below 40°F. Therefore, the 40–120°F DBT operating range appears reasonable and clearly covers HPWH installations in basements and in conditioned spaces within a house.

The amount of moisture contained in the ambient air will affect the average evaporating pressure in the HPWH evaporator. If the inlet air moisture level is high enough to form condensate on the coil, the evaporator pressure will be higher than in dry air conditions. The highest evaporating pressures would occur at high inlet wet bulb temperatures (WBTs) and DBTs. Since the enthalpy of the inlet air is a function of DBT and WBT, we selected total enthalpy of the ambient air as one parameter for determining the evaporation load of the HPWH. We arbitrarily selected several locations in the United States and calculated the total enthalpy of the outside air for design days (1% design dry bulb and wet bulb) during the summer. These results are presented in Table 2.3. We also estimated the minimum air temperature and relative humidity (RH) that a HPWH would experience if located in a relatively confined space inside the home or in an unconditioned basement. These results are also shown in Table 2.3.

Table 2.3. Design ambient air conditions for the heat pump water heater

Garage Installations	Design DBT/WBT (°F)/RH (%)	Total enthalpy (Btu/lb)
Jacksonville	96/79/47	42.3
Las Vegas	108/71/15	34.4
Austin, Texas	100/78/37	41.2
Birmingham	96/78/45	41.3
Boston	91/75/48	38.3
Fresno, CA	101/70/21	33.8
San Jose, CA	89/66/29	30.6
Los Angeles, CA	81/64/39	29.1
Cool locations (winter)	40	
Closet locations	50/48.5/90	19.5
Basement locations	80/75/80	38.7

The highest ambient temperature is 108°F and, as mentioned earlier, an HPWH in an enclosed garage could experience 120°F. Significant production of evaporator condensate would not be expected at such a high ambient temperature condition (barring blockage of air flow). A somewhat lower ambient DBT and higher RH (e.g., similar to conditions in Jacksonville) would encourage rapid condensate formation since the capacity of the HPWH is high and the WBT reasonably close to the DBT.

Durability Test Protocol

From DOE 24-h Simulated Use Tests conducted on a conventional 50-gal EWH, we confirmed that the lower heating element provided all of the recovery energy for the six hot water draws. For prototypes of the HPWH, it was observed that the heat pump unit would similarly provide all of the required recovery energy. For either type of water heater, the controls would be designed so that if the water at the top of the tank drops below the upper temperature set point, the lower element (or compressor, in the case of the drop-in HPWH) turns off and the upper element provides the recovery. If the tank of any water heater were filled with cold water and the system turned on, initial heating would be provided by the upper element; when the upper thermostat became satisfied, the upper element would turn off and the compressor or lower element would turn on to complete full heating of the tank. It is unlikely that in normal operation an HPWH or EWH would repeatedly operate starting from a completely cold tank. Accordingly, our durability test protocol was designed to test the system with the compressor coming on after a water draw from a fully heated tank. Since electric element reliability was not considered an issue relative to HPWH reliability, both top and bottom elements in the ten test HPWHs were disabled for the durability test sequence.

The durability testing consisted of operating the ten HPWH units in an environmental chamber under a set of ambient conditions, representative of the envelope described earlier, that grew progressively harsher with time and number of cycles. Unit components that failed during these tests were to be repaired or replaced so that, as far as possible, all ten units would complete the entire set of tests. Each unit was instrumented as described in Fig. 2.1 and Table 2.4 so that changes in the performance of components, as well as of the unit as a whole, as the number of cycles increased could be determined. The instruments for collecting temperature data included three thermocouples on the refrigeration system (compressor discharge, compressor suction, and top of compressor shell) and eight water thermocouples (water in and out, plus six temperatures at different levels inside the tank). In addition, the lower tank thermistor (control sensor T1) was monitored. Data were collected continuously beginning in mid-December of 2000 and continuing until October 7, 2001.

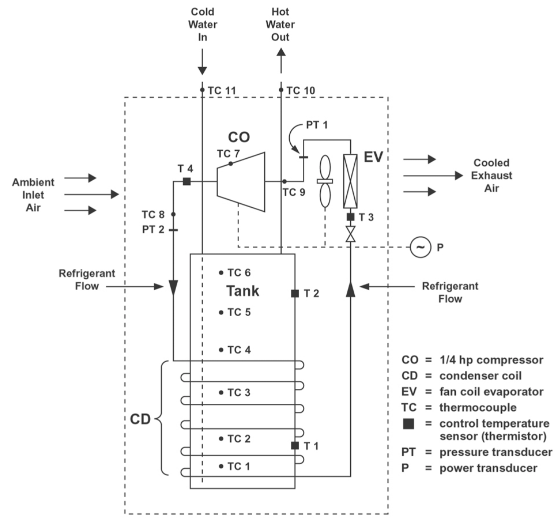

Fig. 2.1. Schematic of heat pump water heater test setup showing data instrumentation locations and control thermistor (T1–T4) locations.

Table 2.4. Heat pump water heater test unit data inputs

HPWH test units	Water flow	Pressure	Power	Temperature[a]
Per unit	1 (flow meter, pulse input type)	2 (compressor discharge and suction pressure)	1 watts, compressor plus fans	12
Total for ten units	10	20	10	120

[a] Eleven thermocouples plus indicated temperature of lower tank control thermistor (T1).

The goal was to identify design and component weaknesses that could impact the reliability and performance of the HPWH over 10 years of simulated residential use. Doing so required a laboratory “cycle rate” of less than one cycle each hour. Cycling was designed to be continual, thereby minimizing compressor off time and maximizing compressor starts.

To accomplish this acceleration and to retain real-world tank conditions over each cycle, hot water was introduced into the tank at a rate of 0.2–0.4 gpm to speed tank temperature recovery (see the test loop discussion in Sect. 3). This arrangement provided the acceleration needed to complete the 7300 cycles in under a year while allowing the condenser and tank to operate through the same temperature changes as in a real-world application. The test facility used for accomplishing this task and for conducting the durability tests is described in Sect. 3 of this report. In addition to accumulating the 7300 duty cycles, the test protocol was designed to cycle the HPWHs under increasingly severe conditions. The protocol was divided into four periods or stages characterized by different ambient conditions and supply voltages. There was also a brief fifth stage at the end during which post-durability simulated use tests, or EF tests, were performed. The operating conditions for these protocol stages are summarized in Table 2.5.

Table 2.5. Operating conditions for each stage of durability test protocol

Stage	Ambient air conditions	HPWH power supply voltage
1	75–80°F dry bulb temperature 50% relative humidity	240 volts AC
2	75–80°F dry bulb temperature 80% relative humidity	240 volts AC
3	100°F dry bulb temperature 50% relative humidity	240 volts AC
4	100°F dry bulb temperature 50% relative humidity	192-204 volts AC[a]
5	67.5°F dry bulb temperature 50% relative humidity	240 volts AC

[a]Five units ran with 192 V supply, one with 196 V, and four with 204 V. HPWH = heat pump water heater

The following sequence of steps was used to accumulate the 7300 compressor duty cycles necessary to simulate 10 years of HPWH operation.

1. Set ambient conditions for stage 1—75–80°F DBT and 50% RH. Set the voltage to the HPWHs at 240 VAC.
2. Set the thermostat of each HPWH to 140°F.
3. Initiate a hot water draw (10.7 gal at 1.0–3.0 gpm). Return approximately 58°F water to the tank during this and all draws. After the draw and with the heat pump operating, introduce hot water at about 150°F into the top of the tank to supplement condenser heat from the compressor (supplemental hot water feedback). The flow rate for the hot water re-introduced into the top of the tank is controllable so that the thermostat (compressor control) is satisfied in about 40 minutes. This required a flow rate of 0.2–0.4 gpm for the HPWHs tested in this study.
4. As soon as the compressor turns off, immediately initiate another draw, repeating Step 3.
5. After every 13 cycles, initiate a “heavy draw” cycle. The heavy draw is designed to deplete the tank to a point just before the upper element would normally turn on. For our tests (since the electric element was disabled), we estimated this point to be when the tank thermocouple second from the top of the tank (see Fig. 2.1) reached 95°F. The heavy draw simulates the impact of coincident operation of a clothes washer, showers, and dishwashers that would reduce the tank temperature so that the upper element would be needed to assist recovery. When the tank water temperature cools to the point of incipient operation of the upper element, the compressor stays on and heats the tank from a cooler temperature than from the preceding 13 cycles. Supplemental hot water feedback is used during “heavy draw” cycle recovery as well to limit its duration.

   We chose the 13-cycle interval for the following reason: Earlier 24-h HPWH tests indicated that the compressor cycled about twice each day. We assumed that in the real world, a heavy water draw from an HPWH would occur about once each week. This suggests that the HPWH would need to recover from a low tank temperature produced by a heavy water draw after every 13 cycles.

6. Continue the stage 1 cycling tests (normal draws and the less frequent heavy draws) around the clock for approximately 2000 cycles.
7. Shift to stage 2 (increase the chamber RH to 80%) and continue the testing regimen described in steps 3–5 for 2000 more cycles. This stage cycles the compressor under higher latent load conditions.
8. At approximately 4000 cycles, shift to stage 3 (increase ambient temperature to 100°F with RH = 50%). The conditions at this stage, somewhat representative of garage locations, further increase the load on the compressor. Continue the cycle schedule (steps 3–5) as before for an additional 2000 cycles.
9. At approximately 6000 cycles, shift to stage 4 (decrease the power supply voltage by 15–20% while maintaining the chamber temperature at 100°F and 50% RH). Continue cycles as described in Steps 3–5 for 1300 more cycles. The reduced voltage conditions in this stage increase the compressor stator winding currents and temperatures.

At the conclusion of the durability tests, ambient and power supply conditions were shifted to stage 5 levels and the EF of each unit was measured according to the DOE Simulated Use Test. Supplemental hot water feedback flow was disabled for these EF tests. Subsequently the compressors were removed and checked for wear in cooperation with the compressor manufacturer. The tanks were returned to the HPWH manufacturer for examination of the condenser/tank adhesion quality and insulation condition and a visual check of other components. A flowchart of the durability testing protocol is shown in Fig. 2.2.

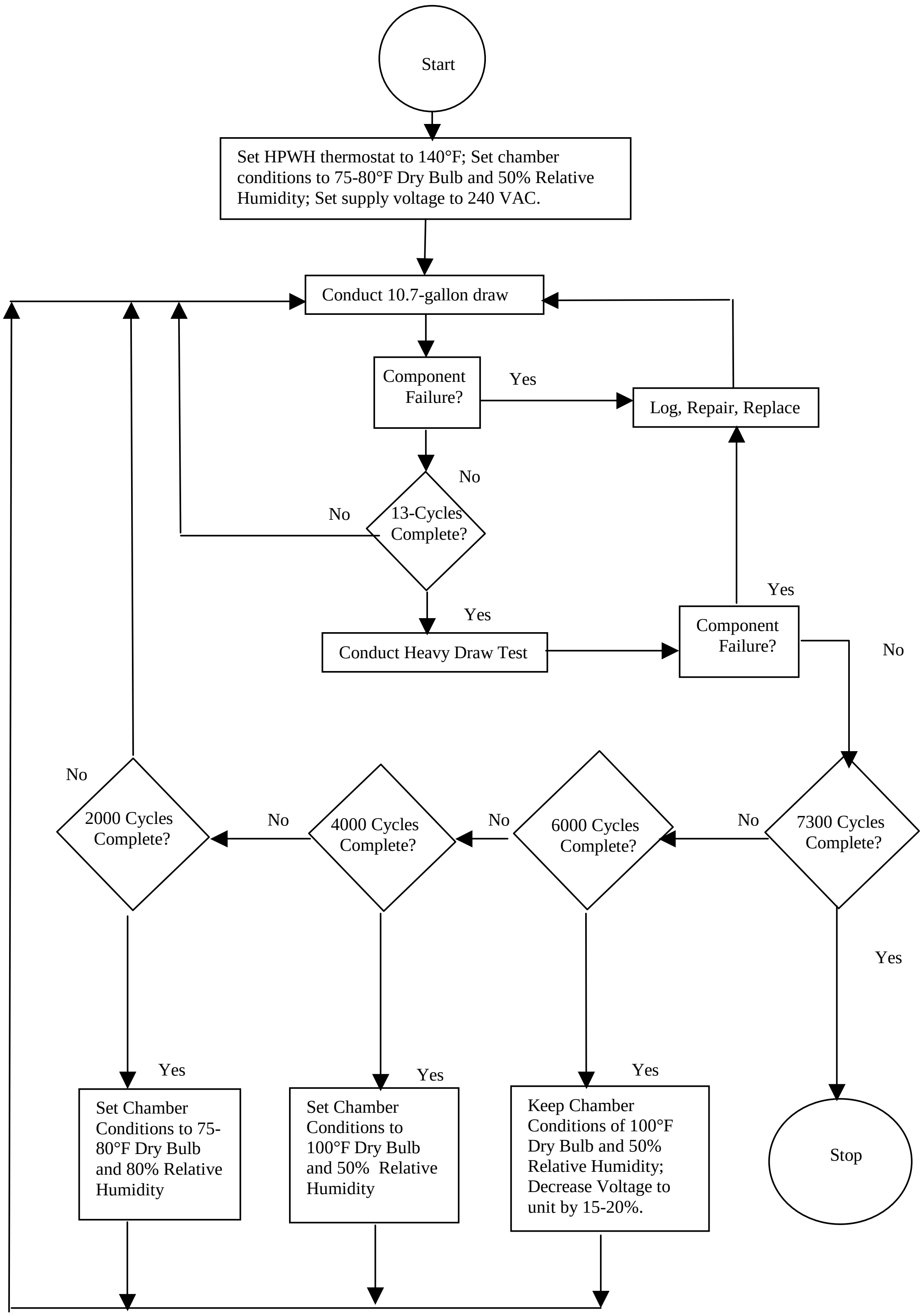

Fig. 2.2. HPWH durability test protocol.

3. THE TEST FACILITY

To conduct the durability test, a test facility was required with the capability to provide a continuous supply of hot and cold water for the ten test HPWHs and control the cycling schedule as outlined in the test protocol. A search for available facilities revealed that none were known to exist that would be capable of executing the protocol described in Sect. 2; therefore, we had to construct a new test stand at ORNL. An overall view of the facility is pictured in Fig. 3.1. The environmental chamber pictured in Fig. 3.1 was used to establish ambient conditions needed for the durability tests. This chamber already existed and has been used in testing a variety of gas-fired and electrically driven commercial and residential heat pump systems and refrigeration systems. The ten units were installed in the rack shown in Fig. 3.2, five on the rack shelf and five on the chamber floor underneath the shelf. One of the units is pictured on the chamber floor to illustrate the hose arrangements used to connect them to the main water loops. To complete the test facility, a system of hot and cold piping loops, together with water-temperature–conditioning systems (described in more detail later in this section), was constructed and connected to hot and cold water supply and return headers in the chamber. Each durability test HPWH unit was connected to these headers through a flow-control module described later in this section.

As noted in the Introduction, each of the test HPWHs is about 5 ft high and 2 ft in diameter. The small heat pump unit located on top of the tank provides water heating. The testing protocol discussed in Sect. 2 was developed to accelerate the cycling rate of the HPWH so that 7300 cycles could be

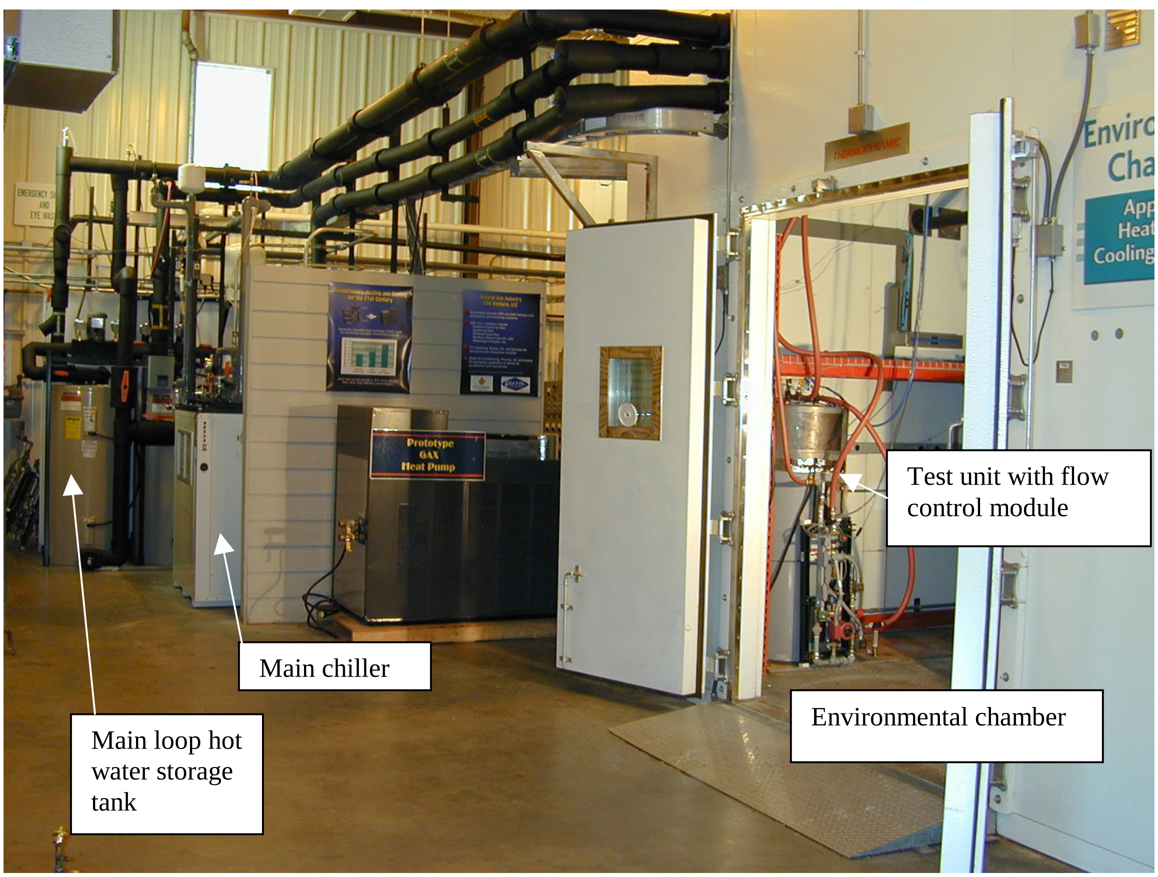

Fig. 3.1. Overall view of heat pump water heater durability test facility.

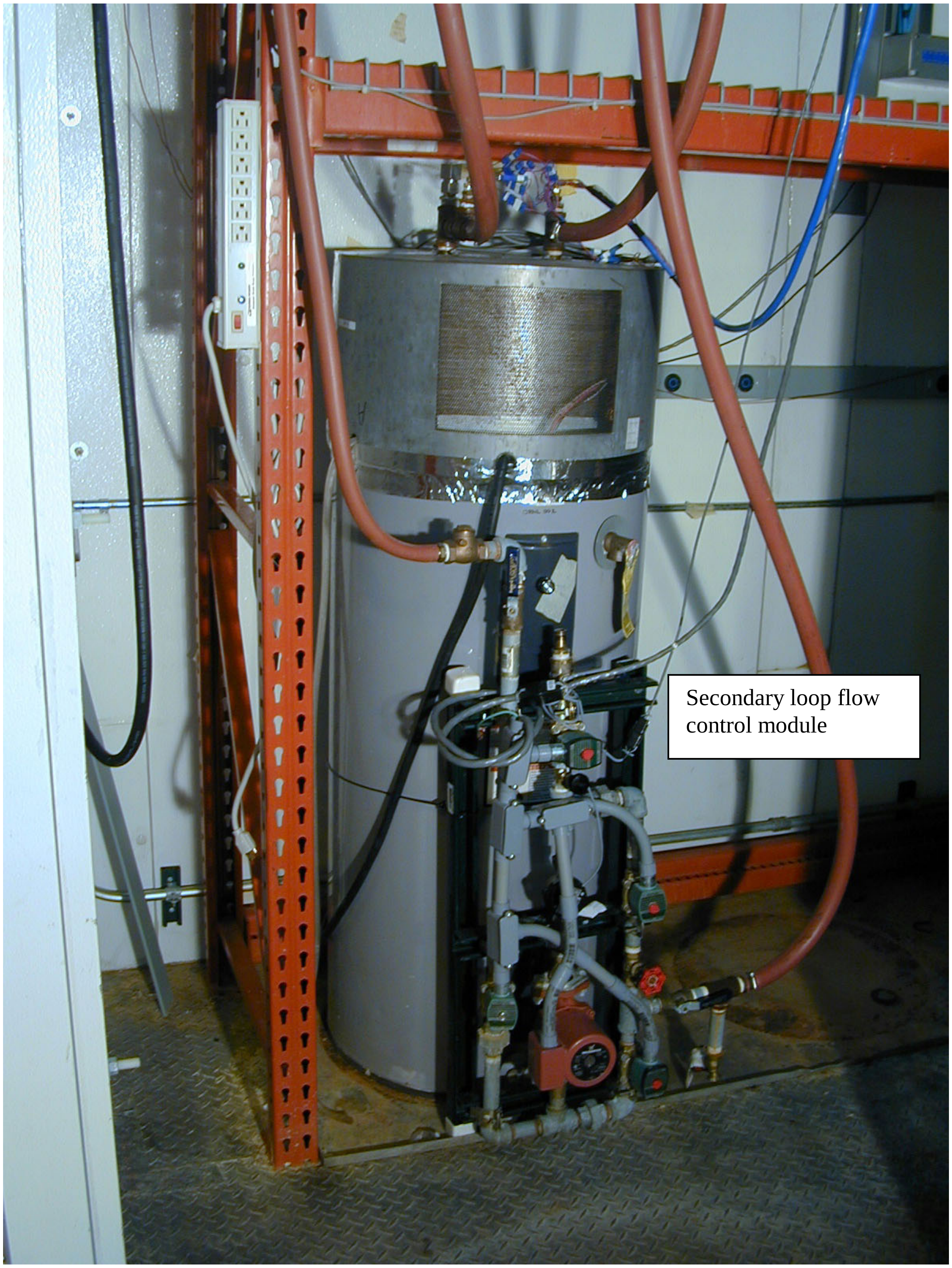

Fig. 3.2. Sample installation of test heat pump water heater inside environmental test chamber. Five heat pump water heaters were placed on the chamber floor like the unit shown. Five more were placed on the shelf above the floor units.

completed in a reasonable time frame. To help the compressor reheat the tank, a small supplemental hot water feedback flow from the main loop system was injected back into the tank outlet (i.e., reversed flow through the water tank). Controlling the flow of this injected hot water can make the time needed to restore the tank to its initial heated condition much shorter than for a normal recovery provided by the compressor alone.

Figure 3.3 shows a schematic of the test stand used for the durability tests. A primary/secondary loop approach was used for providing cold and hot water to each test unit according to its individual demand. For clarity, Fig. 3.3 shows only two HPWH units each with secondary piping. However, the two primary loops (hot and cold) were connected to each of the ten secondary loops.

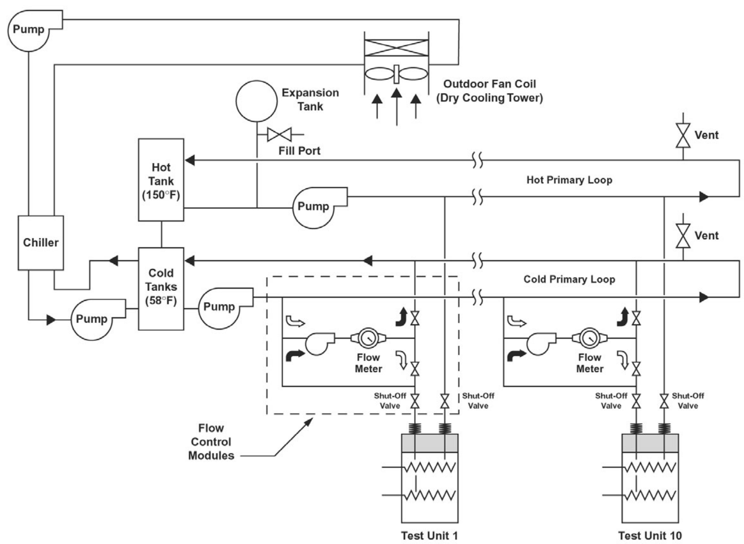

Fig. 3.3. Schematic of HPWH test facility plumbing connections.

Primary Loop System

Figures 3.4 and 3.5 are a photograph and schematic, respectively, of the primary loop system. This system consists of two water circulation loops (one hot and one cold), four storage tanks, and associated conditioning equipment. These loops provide water at approximately 58°F to the HPWH tanks during the ‘draw’ modes and at approximately 150°F to the tanks for supplemental hot water feedback (‘supplemental heat’ mode). There are two 120-gal cold water storage tanks, one 120-gal hot water storage tank, one 120-gal buffer tank, two primary circulation pumps (one for each loop), a chiller for primary water temperature conditioning, an outdoor fan coil for chiller heat rejection (pictured in Fig. 3.6), and an expansion tank to maintain water system pressure.

The hot tank with electrical connections to the two heating element access covers can be seen at the left-hand side of the photo in Fig. 3.4. Just to its right is the buffer tank, and the chiller unit is pictured in the foreground. The two cold water tanks are located between the chiller and the buffer tank but are obscured in the photo by the chiller. All of the system components, except the outdoor fan coil, are located within the Buildings Technology Center’s laboratory facilities in Building 3144A at ORNL. Standard, residential 120-gal electric water heater tanks were used for primary system hot, cold, and buffer water storage. Two 5500-W electric heating elements in the hot tank were used to keep the hot loop at operating temperature conditions (150°F). The controls supplied with this water heater were modified so that both heating elements could be operated simultaneously. The buffer tank was used to provide for flow and pressure balancing between the two primary loops when water

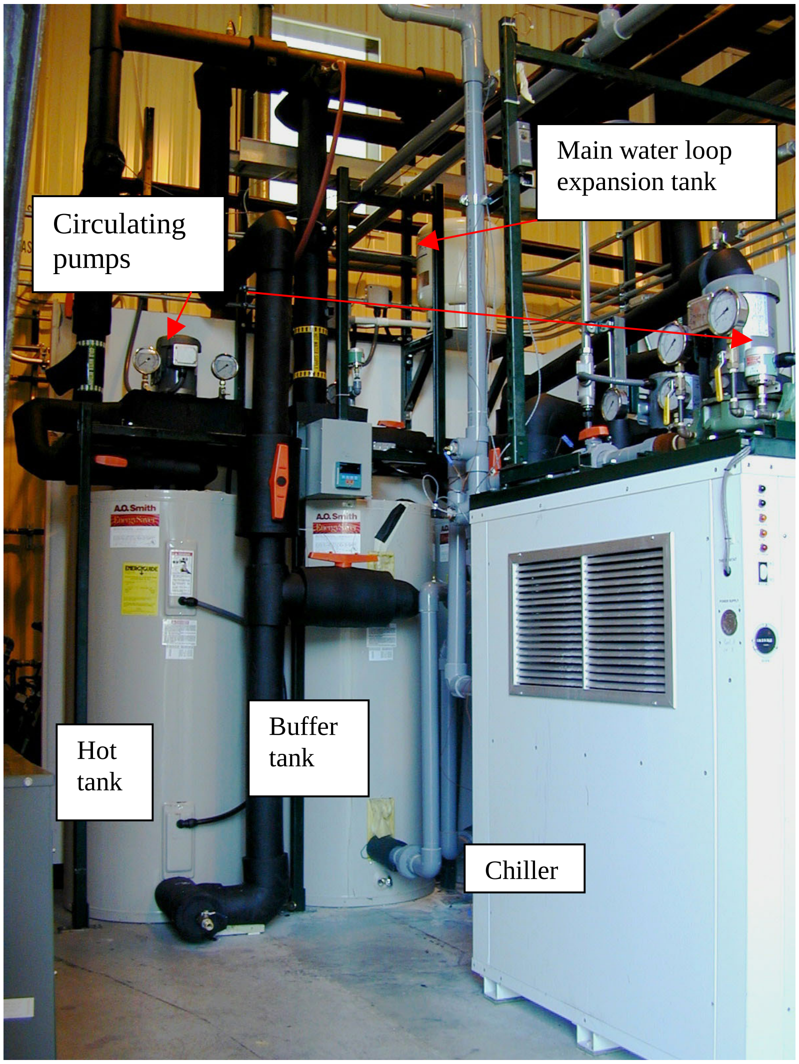

Fig. 3.4. Photo of heat pump water heater test facility chiller and primary water storage tank section.

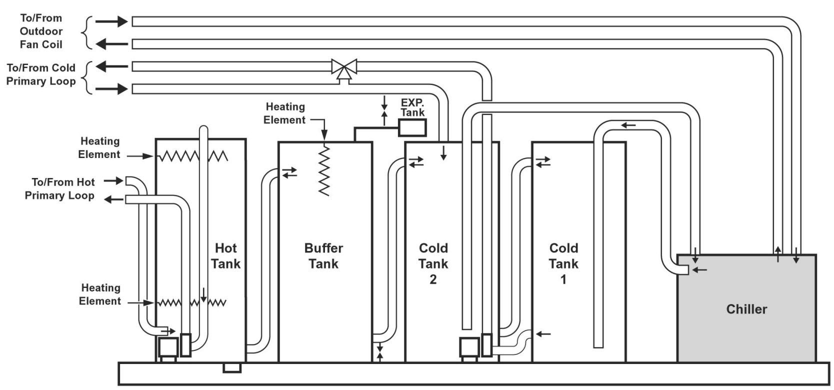

Fig. 3.5. Schematic of heat pump water heater test facility chiller and primary water storage tank section.

Fig. 3.6. Photo of outdoor fan coil used for chiller heat rejection.

removals from the cold loop were greater or less than those from the hot loop. One 4500-W electric element was used in the buffer tank to keep the top half at approximately 130°F.

The hot and cold piping loops served to provide cold and hot water for the test HPWH draw and supplemental heat modes, respectively. Each consists of large-diameter pipe and a circulating pump, and each maintained a constant flow rate of about 30 gpm during the durability testing. The fittings normally used for electric elements were used for loop piping connections for the two cold tanks and the buffer tank. The cold loop was plumbed, as illustrated in Fig. 3.5, so that water was circulated from the bottom of tank 1 to the test units and then returned to the top of tank 2 after picking up supplemental heating water flows.

The system chiller was actually an E-Tech commercial water-to-water heat pump (WWHP) model WW-120 NTD (82,000 BTU/h rated cooling capacity and 105,000 BTU/h heat rejection rate with 70°F water entering the evaporator and 120°F water/glycol entering the condenser). The chiller capacity requirement was estimated as follows.

A control volume was assumed around the primary water system encompassing the hot and cold primary loops and the four storage tanks. It was assumed that over a 40-min period, nine of the HPWH test units will each undergo a 10.7-gal "typical" hot water draw at 3 gpm (for 3.57 min.). It was further assumed that the remaining test unit will undergo a "heavy draw" of 40.7 gal (at 3 gpm for 13.57 min). Thus a total of 137 gal of water (at 58°F) is removed from the cold primary loop and tanks and an equal amount of water (at about 150°F) is injected into the hot primary loop and tank. This results in a total amount of heat energy added to the primary system of about 105,500 Btu.

$$\{137 \text{ gal} \times 8.33 \text{ lb/gal} \times (150\text{-}58)\ °\text{F} \times 1 \text{ Btu/lb-°F} \approx 105{,}500 \text{ Btu}\}$$

All of the test units were assumed to be in "supplemental" flow mode (control valves in position B, Fig. 8) for the balance of the period at the expected minimum flow rate of 0.2 gpm—nine of the units for 36.43 min and the other for 26.43 min. Thus a total of 70.87 gal of water at approximately 150°F will be injected into the HPWH tanks from the hot loop during the period and an equal amount of cooler water (assumed to be at 58°F) injected into cold primary loop. Thus about 54,300 Btu is removed from the primary system over the 40-min period.

$$\{70.87 \times 8.33 \times (150\text{-}58) \times 1 \approx 54{,}300 \text{ Btu}\}$$

To balance the energy flows into and out of the primary system, the chiller has to remove about 51,200 Btu from the primary system cold tanks during the 40-min period. This is equivalent to a rate of about 77,000 Btu/h, or about 6.5 tons cooling capacity.

The selected WWHP had the required capacity capability. A user-adjustable condenser pressure-regulating valve was provided with the WWHP to modulate condenser water/glycol flow to maintain a minimum condenser pressure level.

In operation, the cold loop return water temperature to Cold Tank 2 and the chiller evaporator ranged from 65 to 75°F most of the time, and the average Cold Tank 1 storage temperature ranged from 48 to 58°F. A three-way mixing valve was included in the cold loop to mix the cold loop return and cold tank storage water streams to maintain the 58°F supply temperature to the cold primary loop and the test HPWHs.

An existing outdoor fan coil located outside Building 3144A (Fig. 3.6) was determined to be of adequate size to handle the heat rejection load from the system chiller. The coil has a rated capacity of 120,000 Btu/h at 95°F outdoor ambient temperature and 125°F entering water/glycol fluid temperature.

Expansion tanks were included in both the primary water system and the heat rejection loop to maintain a static fluid pressure of at least 20 psig. Pressure switches were included in both loops as well to provide an alarm signal to the control system in the event of loss of fluid pressure (system leak).

Secondary Loop Systems

Each test HPWH operated independently of all others in its own separate secondary flow loop. The secondary piping circuits were used to control the draw and supplemental water flows for the test HPWH units. Each had a flow control module, as shown schematically in Fig. 3.7, containing a small circulating pump, a water flow meter, four solenoid valves, a check valve, and metering valves for fine control of the water flow. One of these flow modules can be seen in Fig. 3.2 as attached to its test HPWH. Module controls were configured so that in flow position A ('draw mode') all solenoid valve coils are energized, and in position B ('supplemental heat mode') the coils are deenergized. Metering valve A was used to adjust the draw flow rate to the desired 1–3 gpm level. Metering valve B was used to adjust the supplemental hot water flow to approximately 0.2–0.4 gpm to achieve "typical" HPWH cycle times of about 40 min (see the following discussion). The controller for the HPWH test loop was set up to operate each secondary circuit flow control module as follows.

1. For typical cycles:
    A. With the solenoid valves in position A (Fig. 3.7), 1–3 gpm was introduced from the cold primary loop to the bottom of the HPWH tank (the draw mode). In this process, hot water from the top of the tank was discharged into the hot primary loop.
    B. After 10.7 gal of cold water were delivered to the HPWH tank (approximately 3 min and 34 s at 3 gpm), the control system switched the solenoid valves to position B and supplemental heating flow began. Flow rate for this mode (0.2–0.4 gpm) was set using metering valve B to achieve a total typical cycle length (from start of one water draw to the start of the next water draw) of about 40 min.
    C. Supplemental mode flow continued until the HPWH compressor shut off (lower tank thermostat satisfied). Immediately, the solenoid valves were switched by the loop controller to position A and the typical cycle was repeated 12 more times until 13 typical cycles were completed.

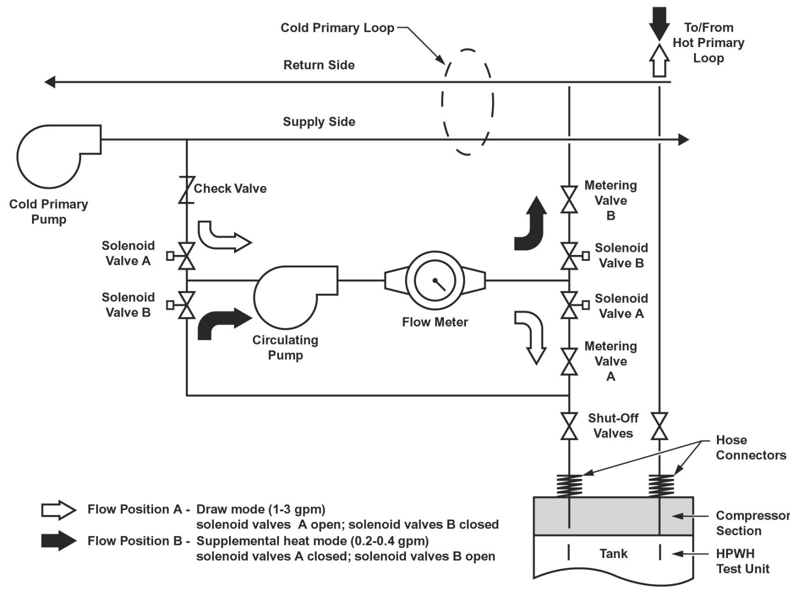

Fig. 3.7. Schematic of secondary loop flow control module.

2. For “heavy draw” cycles (every 14th cycle):
   A. After 13 typical cycles, the draw mode was controlled to discharge a much larger volume of hot water from the HPWH tank. Water was removed from the HPWH tank until the tank water temperature at position 5 (second thermocouple from top of tank, Fig. 2.1) reached 95°F.
   B. When the indicated temperature reached 95°F, the solenoid valves were switched to position B and supplemental heat water flow commenced and continued until the HPWH compressor shut off (lower tank thermostat set point satisfied).
   C. Immediately, the solenoid valves were switched to position A and a new series of 13 typical and 1 heavy draw cycles began.

This cycle of 13 typical and 1 heavy draws continued until each test HPWH completed about 7300 water heat cycles representing about 10 years of normal operation.

Data Acquisition and Control System

An Intellution® Version 7.0 software system was chosen to provide automatic control of the secondary flow modules and the primary water loop. Analog and digital input hardware from MTL® was used to gather temperature, pressure, water flow, and power draw data from the primary instruments, and the Intellution software logged and stored the data. Type T, copper-constantan thermocouples were used for all temperature sensors. Omega® model PX120 transducers were used to measure compressor suction and discharge pressure on each test HPWH. The suction sensors were of 0–100 psig range, and the discharge sensors had a 0–500 psig range. Ohio Semitronics model PC5-002X5 power transducers were used to measure the power draw of the compressors and fans of the HPWHs. One transducer was used for each HPWH; the transducers had a range of 0–1000 W. Each power circuit included a power stat to control the power supply voltage to the HPWH test unit. An Omega model FTB-4605 water flow meter was used to measure the draw and supplemental water flows on each flow control module. This meter provided a pulse output of 151.4 pulses/gal. The Intellution software program was configured to count each typical and heavy draw cycle and maintained a running total of cycles throughout the test period.

4. TEST RESULTS AND DISCUSSION

The durability test program commenced in mid-December 2000 and continued until October 7, 2001. Except for two major outages due to failures of the primary loop system during Stage 1 of the protocol (totaling about 3 weeks), the test loop and the HPWHs operated more or less continuously throughout this period. Figure 4.1 shows power, average tank water temperature, and secondary loop water flow for a representative 6-hour period for test unit 1. The cycling pattern shown in Fig. 4.1 is fairly representative of that undergone by all ten test units during the durability run. Table 4.1 summarizes total cycles accumulated and the approximate operating hours for each unit. Cycle count ranged from a high of 7950 to a low of 6677. Operating hours ranged from about 6000 to about 5200. Unit 6 was off-line for about 2 weeks in late January with a failed evaporator thermistor (T3). Unit 10 was off-line for about 2 weeks in June and July until problems with its lower tank thermistor (T1) were diagnosed. Unit 3 was off-line for a total of 33 days in June, July, and August with a variety of thermistor and control board problems (discussed more fully in Sect. 4.1).

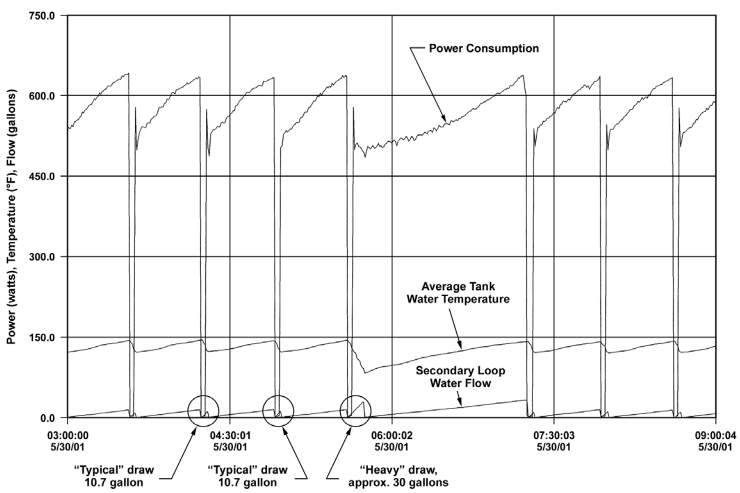

Fig. 4.1. Representative cycling behavior of test heat pump water heater 1 over a 6-h period. Power consumption, average tank water temperature, and secondary loop total flow vs. time shown. Cycling rate was approximately 32 cycles/day at this point in time.

Table 4.1. Total cycles and approximate operating hours accumulated by each unit during durability test

Unit	Total cycles accumulated	Estimated total operating hours
1	7950	5930
2	7696	5940
3	6677	5200
4	7748	5940
5	7213	5970
6	7736	5640
7	7804	5940
8	7349	5930
9	7534	5950
10	7398	5660

The following sections discuss various component problems and failures for the test units; results of post-test examinations of the compressors, condenser wrap, and tanks; and operating characteristics of the units (e.g., efficiency, refrigerant operating conditions).

4.1 System Reliability Discussion

Component Failures

Table 4.2 provides a summary of the various component failures experienced by the ten test units over the course of the durability run. The most immediate thing to note is what is *not* in this table. There were no compressor failures, fan failures, or compressor or fan relay failures on any of the units during the durability test sequence. (Compressor 3 later failed to start during calorimeter testing at the compressor manufacturer's laboratory after the durability run. The most likely reason for this failure, according to the manufacturer's engineer, is that an open circuit failure occurred within the connector between the motor and the internal power terminal.) The only problem arising from any component of the vapor compression heat pump systems was one thermal expansion valve (TXV) failure on unit 2. The TXV used was equipped with a bleed port that enabled rapid equalization of high- to low-side system pressure. This particular TXV had a blocked bleed port and would not equalize quickly. The problem was found and the TXV replaced during pre-test checkout of the HPWHs. No TXV failures occurred during the durability test run.

There were instances of refrigerant leakage on two of the units (7 and 9) during the testing. In both cases, these were determined to have occurred in the solder joints of discharge pressure transducer fittings. These fittings were added to the test units to facilitate our data acquisition needs. Such fittings would not normally be a part of the system; therefore, this would not be reliability concern for production units.

By far the greatest source of problems was the control system temperature input sensors (thermistors T1–T4). Figure 4.2 is a photo of one of the thermistors as used in the first-generation controls. The design featured very fine 28-gauge lead wires and included a spliced connection (the black banded area in Fig. 4.2) to provide connecting leads from the thermistor location to the control board terminal points. All of the thermistor failures noted in Table 4.1 were due to failures of these splices as either shorts or open circuits. The control system software embedded in the central microprocessor interpreted a shorted connection as a temperature reading of 100°F and an open circuit as a temperature of 179°F. Either of these values can be real inputs from any of the sensors during heat pump operation (although not likely for the evaporator sensor, T3). A short in the lower tank (T1) thermistor circuit would be interpreted by the controller as cold water in the bottom of the tank. This would cause either the heat pump or the lower element to operate continuously until the high-temperature-limit switch shut off the unit power or a failure occurred. An open T1 circuit would be interpreted as an excessively hot tank, and the controller would not permit the heat pump (or lower element) to operate at all. Either type of failure in the evaporator thermistor circuit (T3) would indicate a high evaporator temperature and the controller would not permit the fans to operate. The result would be very low suction pressure with concomitant evaporator frosting (if the local humidity is high enough) and significantly reduced heat pump water heating capacity.

A short in the upper tank thermistor circuit (T2) would be interpreted as cold water in the top of the tank, locking out the heat pump or lower element and causing the upper element to run continuously until the high-temperature switch cuts off unit power. An open T2 circuit will not allow the upper element to operate for quick hot water recovery in the event of high water usage. A short in the discharge thermistor (T4) circuit (indicating a discharge temperature of 100°F to the controller) would cause the water heater to operate in EWH mode until the thermistor was replaced. An open circuit failure of T4 (indicating a temperature of 179°F) would nullify the high-discharge-temperature cutoff safety feature built into the control system. Ultimately, any failure in circuits T1–T3 would be manifested by no or low hot water supply to the residence. A short-circuit failure of T4 would result

Table 4.2. Heat pump water heater component failure summary

Unit #	Lower tank thermistor, T1	Upper tank thermistor, T2	Evaporator thermistor, T3	Discharge thermistor, T4	Thermostat potentiometer	Mode select switch	Control board[a]	High temperature limit switch	Expansion valve	Compressor	Other
1			9/25/01								
2		8/00[b]							9/00[b,c]		
3	7/10/01	7/26/01 9/6/01	11/00[b]	7/10/01 Not replaced			7/30/01 8/7/01			12/4/01[d]	8/7/01[e]
4		8/00[b]									
5	9/00[b] 7/26/01	7/26/01 8/17/01			7/25/01[f]		8/00 Shorted because of water leak				
6	7/23/01		2/1/01[g]			9/00[b]					
7											2/5/01[h]
8											
9	10/00[b]							7/24/01[i]			10/23/01[j]
10	7/25/01[k]										

[a]All units exhibited 60-cycle noise on thermistor and thermostat inputs during pre-test checkout. Grounding control boards to the HPWH tanks corrected this problem.
[b]Discovered and replaced during pre-test checkout period.
[c]Valve bleed port evidently blocked; discharge and suction pressures would not equalize quickly.
[d]Compressor motor failed during post-test calorimeter evaluation by compressor manufacturer (Embraco).
[e]An intermittent short developed between the new control board and heat pump shroud. Adding insulation between the board and shroud eliminated this problem.
[f]A discontinuous jump was discovered in the thermostat (it would jump from 135 to 141°F). Thermostat was left at 135-°F setting and not replaced.
[g]One incident of evaporator frosting/fan shutdown was noted in December 2000. Thermistor failed in late January 2001 but was not replaced until February 1.
[h]Unit developed a refrigerant leak at the discharge line pressure transducer fitting solder joint and lost about 80% of its charge over a one-month period.
[i]Switch failure most likely due to moisture-induced short caused by condensate overflow from evaporator drain pan. The overflow was due to poor drainage system in our test facility and is not inherent in the HPWH design.
[j]Unit developed a refrigerant leak at discharge line pressure transducer fitting solder joint and lost about 60% of its charge between November 2000 and October 2001.
[k]Unit began cycling erratically in late June and was out of service for a total of 13 days until thermistor problem was diagnosed and thermistor replaced.

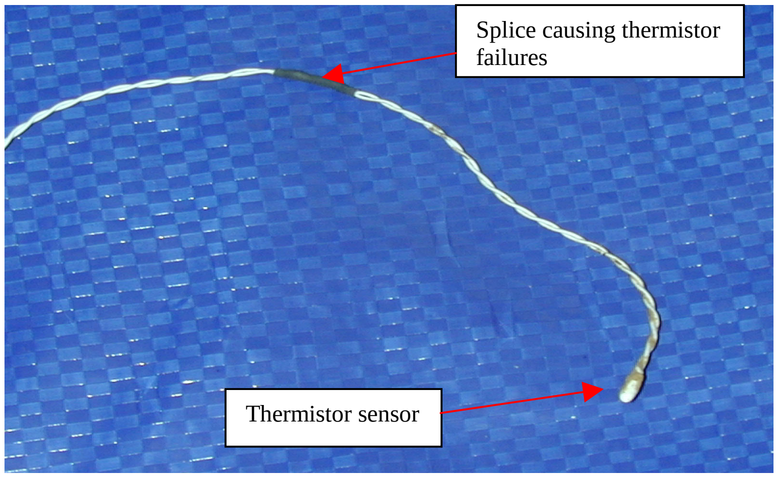

Fig. 4.2. Close-up photo of thermistor used as temperature sensor in first-generation control design.

in EWH operation (and no energy savings) until the problem was corrected. An open circuit failure of T4 would not immediately affect hot water production and might go unnoticed for some time. However, it could result in a premature compressor failure, particularly if the user had the unit thermostat set at a very high level (>140°F) for long periods.

Unit 3 was plagued by the most control system component failures and problems among the test HPWHs. Besides losing five thermistors at various times, it also experienced failures of two control boards. Most of these problems occurred during the June–August period (Stage 3 and 4 operation). The combined effect of these various problems caused the unit to be out of service for almost 5 weeks during that period. The unit first began experiencing erratic cycling behavior (rapid compressor cycling) in mid-June, but this behavior tended either to self-correct or to stop when power was turned off and then back on. In late June, however, the problem became permanent; and the unit was shut off until mid-July, when we were able to attempt to diagnose the problems. Both thermistors T1 and T4 were found to be sending erratic signals to the controller—bouncing rapidly from high to low values. Thermistor T1 was replaced, and thermistor T4 was disconnected so that the unit would run. Replacing T4 would have required removing the unit from the test chamber, so it was decided to run without T4 so that cycles could continue to accumulate.

Two weeks later, thermistor T2 began exhibiting the same erratic behavior, so it was replaced. The original control board failed shortly afterward. The first replacement board also proved to be inoperable, so we had to install a second replacement. During replacement of the heat pump cover shroud, it became evident that there was an intermittent short between the shroud and the new board—the unit shut off and thermistor T1 showed an open circuit. The shrouds have a thin foam insulation lining on the inside, but this one apparently had enough wear in the vicinity of the control board that a short was possible. Adding a layer of fiberglass insulation eliminated this short. About a month later, the new T2 thermistor began giving fluctuating readings, so we disconnected it from the board terminal and completed the durability test run without this sensor in place.

Unit 5 experienced a failure of its control board early in the pre-test checkout period. The failure was caused by a small leak in a hose connection on the cold water inlet. The water dripped onto the shroud and penetrated the joint between the shroud halves. Water subsequently dripped onto the power connection terminal located on the HPWH frame, causing an arc that destroyed the control board.

Test unit 9 experienced a failure of its high-temperature cutoff switch, but this failure is not considered to be due to any problem inherent in the HPWH design. Unit 9, along with about six of the others, had condensate overflow problems during the high-ambient-humidity test stages (2 and 3). The overflow was determined to be due to a poorly designed drainage scheme employed in the test chamber, and not to the HPWH drain pan design itself. The original design directed condensate from the units to the main chamber drain port (up to 20 ft from some of the test units) using ¼-in.-inner-diameter (ID) plastic tubing. This design did not work reliably at all, so it was replaced with a system of short drain hoses with large IDs and float-controlled condensate pumps, which effectively eliminated the condensate drainage problems. In some of the units, including #9, the overflow from the drain pans leaked into the electrical connection and down to the upper element access area. This leakage caused the failure of the high-temperature switch in unit 9. While this particular failure instance was not due to any flaw inherent in the HPWH design, it does point out a potential problem that could occur in field installations should the HPWH drain pan become blocked. Some care should be taken in future production designs to ensure that any condensate overflow incident will not lead to water leakage into the electrical power connection box on the water heater.

Electrical Noise Problems

During the pre-test checkout of the test units for proper operation, we observed that the control system was subject to unsteady fluctuation of the input signals (thermistors and thermostat). Using an oscilloscope, we discovered that all the signals exhibited a 60-Hz (60-cycle) wave pattern. Grounding each unit's control board to its frame eliminated this problem. Random episodes of noise spikes were observed on the control inputs of several of the units throughout the checkout period and during the durability testing itself. The root causes of these spikes could not be determined with absolute certainty; however, it was thought that they probably were due to the close proximity of the control lead wires (low-voltage) to the power wiring (high-voltage) in the first-generation system design. Figures 1.5 and 1.6 show that the thermistor wires (small white wires) were bundled with the 240-V power wires. Physical separation of the low-voltage and high-voltage wiring and shielding of the low-voltage wires could help eliminate many of the random noise spikes seen in our testing program.

Another factor that may have played a role in some of the "noise" problems was the way the control logic was set up. In the first-generation version of the control program, the microprocessor would sample each control input (thermistors and thermostat) only one time before making control decisions. With this setup, a single anomalous signal input could cause the controller to sense a problem when, in fact, none actually existed. Averaging several control input readings (or oversampling) could help ameliorate the effect of isolated aberrant thermistor readings. A change was made to the control program so that it averaged 32 values of each control reading before taking control actions for the units used in the field test program (Murphy 2002).

Unit Noise Problems

No increase in fan or compressor noise level was observed for any of the test units during the durability run. There were a few instances of relay chatter (a buzzing type of noise) related to some of the thermistor failures where the thermistor signal fluctuated rapidly, causing the controller to rapidly cycle the relay.

Miscellaneous Unit Quality Issues

When the test units were received from the manufacturer, a number of assembly problems were noted and had to be corrected before checkout and durability testing. The major issues were that several of the unit thermostats were not firmly attached to the upper access covers—they turned freely in the mounting sockets. When tightened, they worked as designed. In addition, on several of the units, the control boards were not secured well to the HPWH frame—mounting screws or studs were missing. A careful manufacturing quality assurance program will eliminate instances such as these.

4.2 Unit Performance

Performance Characteristics Over Time

Figures 4.3 and 4.4 show end-of-cycle discharge pressure and temperature trends for HPWHs 2 and 3 compared with their average tank water temperature and lower tank thermistor (T1) reading. The thermostats for each test unit were all set at about 140°F, as is reflected by the T1 curves in Figs. 4.3 and 4.4. However, because of differences between units (e.g., in compressor performance, refrigerant pressure drop, air flow, thermistor precision and variability), there was some variation in heat pump operating conditions from unit to unit. The differences in end-of-cycle discharge pressure and temperature between units 2 and 3 illustrate the approximate ranges of these operating parameters among the ten units (Figs. 4.3 and 4.4, respectively). There was a range of approximately 50–60 psi in discharge pressure and a range of 15–20°F in discharge temperature. The impact of thermostat variation on unit operating conditions is clearly seen in Figs. 4.3 and 4.4. The drop in pressure and temperature for unit 3 during days 100–110 was caused by a gradual (and unintended) decrease in the thermostat setting to its minimum position. Also, an inadvertent increase in unit 3's thermostat setting between days 30 to 55 is reflected in a discharge pressure increase, as shown in Fig. 4.3. The discharge temperature also shows an increase in this period; however, the temperature instrumentation on all the test units was malfunctioning between days 35 to 80, so temperature data during this period are somewhat suspect.

Individual end-of-cycle trend plots for all ten units are provided in Appendix A.

Low-Voltage Impacts

When stage 4 of the test protocol started, the voltage to all ten test HPWHs was reduced to 192 V, a 20% reduction from 240 V. However, it soon became apparent that some of the units (2, 3, 5, 8, and 9) had difficulty in restarting their compressors after normal cycle shutoff. Two would not restart for

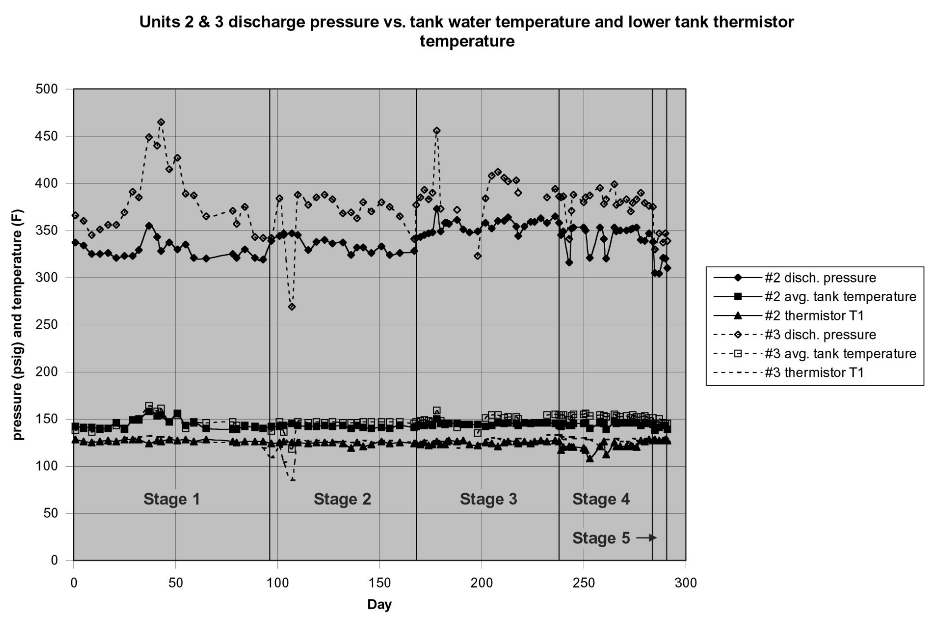

Fig. 4.3. Representative end-of-cycle discharge pressure, average tank water temperature, and lower tank (T1) thermistor readings for units 2 and 3 throughout the durability run.

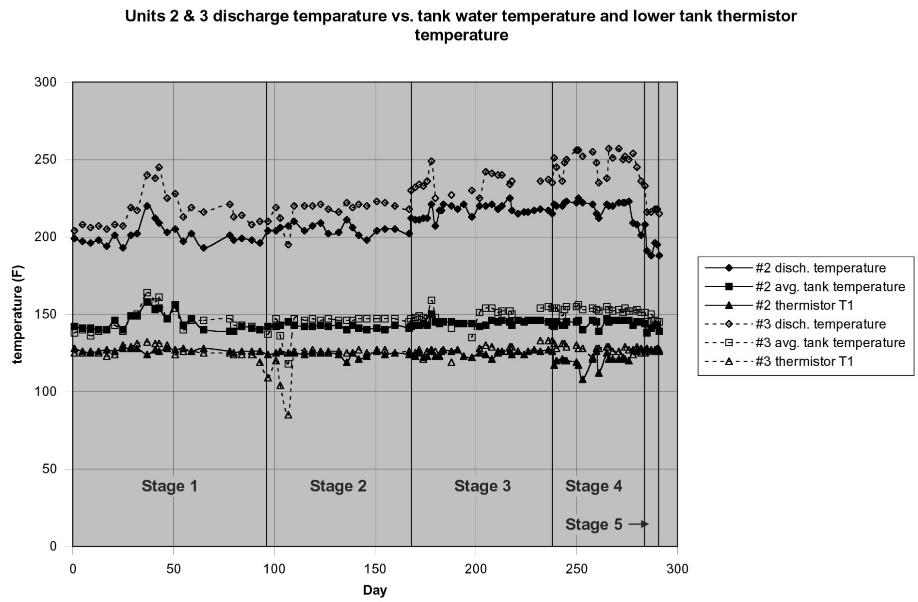

Fig. 4.4. Representative end-of-cycle discharge temperature, average tank water temperature, and lower tank (T1) thermistor readings for units 2 and 3 throughout the durability run.

more than 40 min, and three others were experiencing 20–30 min delays in restarting. We increased the voltage on units 2, 5, 8, and 9 to 204 VAC (15% below nominal 240 VAC) and on unit 3 to 196 VAC (18% below nominal). All ten of the units would then cycle reliably, though compressor restart generally took a few minutes longer than it did at 240 VAC supply. The reasons for this behavior are not fully identifiable, but it is suspected to be largely due to reduced starting torque available at the lower voltage. The reduced torque would require that suction and discharge pressures must equalize to a greater extent before the compressor could start.

The principal impact of reduced supply voltage was expected to be increased compressor operating temperature. This impact was manifested as measurable increases in compressor discharge temperature in some of the test units, including 2 and 3 (Fig. 4.4). Unit 2 experienced a slight (5–10°) rise in discharge temperature in Stage 4 of the protocol compared with Stage 3. Unit 3 experienced a rise of 15–20° in discharge temperature and a slight increase in average tank temperature as well. The greater rise in temperature for unit 3 can be attributed to the fact that its discharge thermistor (T4) was inoperative, and thus its control system was unable to limit the discharge temperature.

Other units experienced little change in discharge conditions, and some showed a decrease in pressure. Figure 4.5, for instance, shows that the end-of-cycle discharge pressure of unit 1 dropped significantly at the beginning of Stage 3 (high temperature) and underwent an even greater drop in Stage 4 (high temperature and low voltage). The pressure recovered to approximately Stage 1 levels in Stage 5. Discharge temperature increased by about 10° near the beginning of Stage 3 and remained nearly constant until Stage 5. The end-of-cycle T1 thermistor level for unit 1 dropped from 125–130°F in Stages 1 and 2 to 110–115°F in Stage 3 and to about 95°F in Stage 4. End-of-cycle discharge pressure and temperature for unit 7 showed similar behavior in Stage 4, as shown in Figs. 4.6 and 4.7. The T1 thermistor values for unit 7 dropped from 125–130°F to 100–110°F at the beginning of Stage 2 (humidity increases) and to about 95°F in Stage 4. The thermostat settings did not change in either case, so that was not likely to have been the reason for the lower T1 end-of-cycle readings. By observation of the units, it was surmised that the discharge thermistor (T4) began shutting off the compressor before the lower tank set point temperature was reached, thus causing

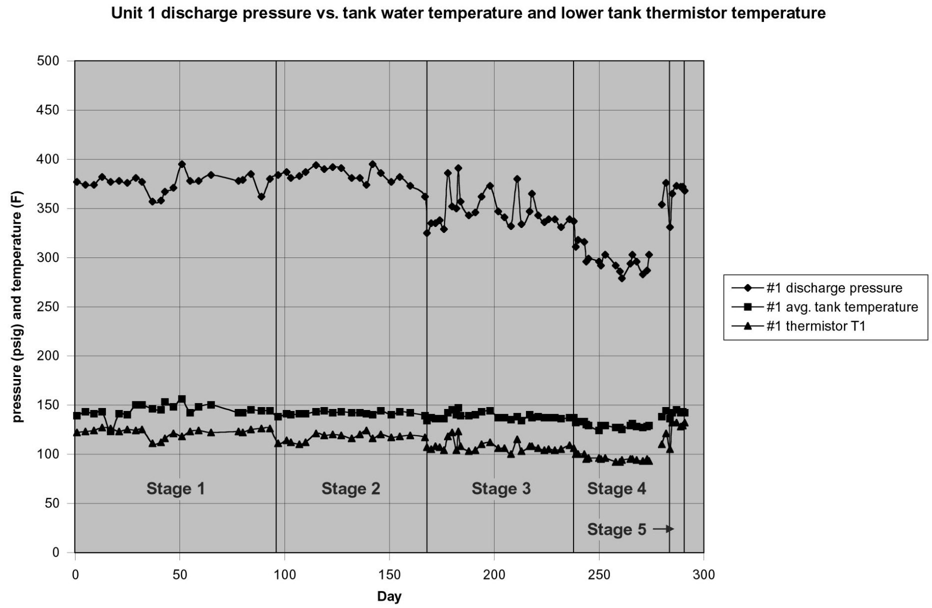

Fig. 4.5. Representative end-of-cycle discharge temperature, average tank water temperature, and lower tank (T1) thermistor readings for unit 1 throughout the durability run.

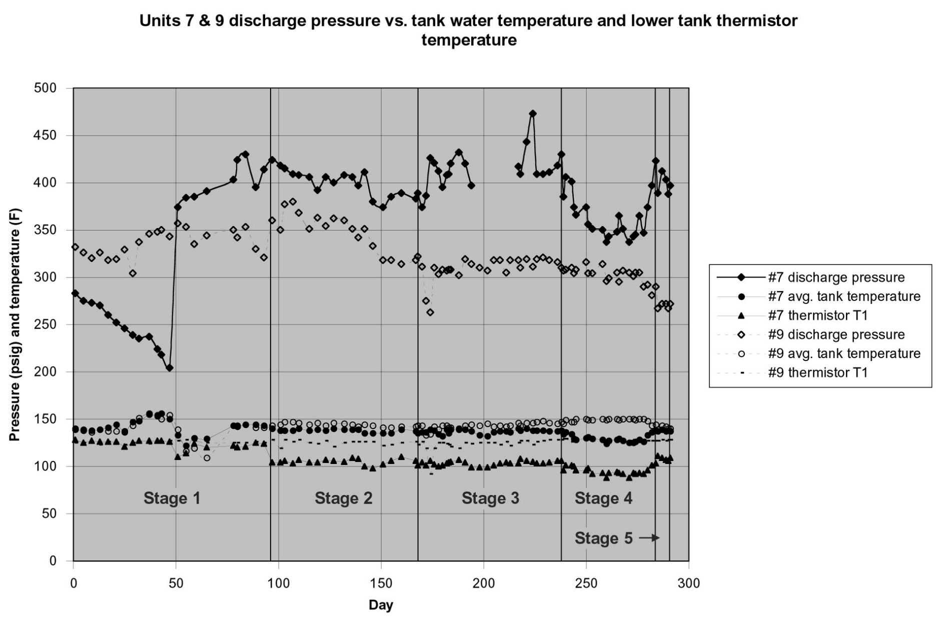

Fig. 4.6. Representative end-of-cycle discharge pressure, average tank water temperature, and lower tank (T1) thermistor readings for units 7 and 9 throughout the durability run.

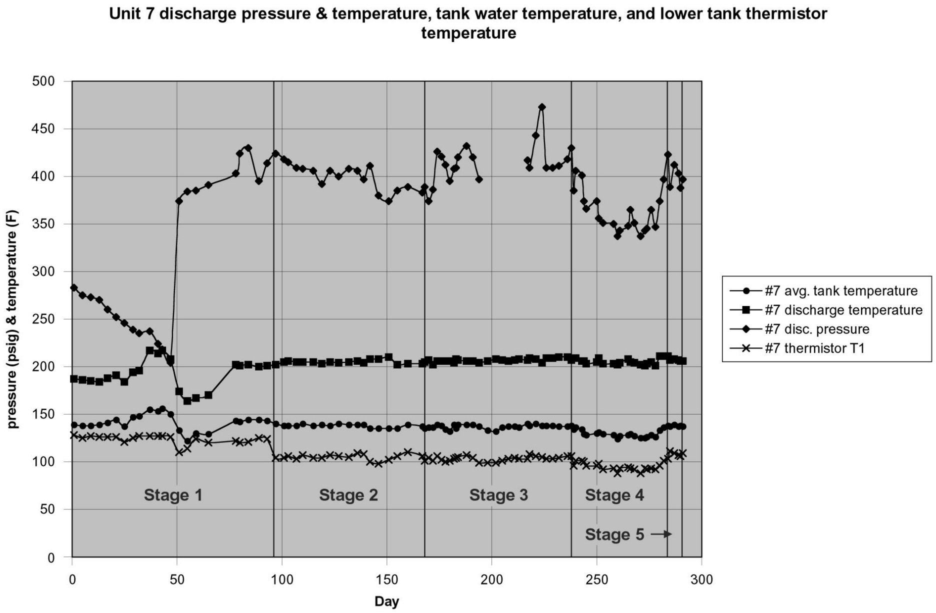

Fig. 4.7. Representative end-of-cycle discharge pressure and temperature, average tank water temperature, and lower tank (T1) thermistor readings for unit 7 throughout the durability run.

end-of-cycle T1 values to be lower than in prior stages. Recall that the controls include a yellow, red, and green LED to indicate compressor, lower element, and upper element operation, respectively. At normal cycle shutoff, when the thermostat is satisfied, none of the LEDs is on. Beginning in Stage 2, at cycle shutoff of unit 7, it was observed that the red LED would illuminate, indicating a call for lower element operation. Since we had disconnected both electric elements for the durability tests, the test loop controls simply initiated another hot water draw from unit 7 when the call for lower element operation occurred.

Several conditions could cause the HPWH control to shut off the compressor and call for lower element operation (see the discussion of control logic in the Introduction section). The two most likely are (1) the lower tank temperature exceeded 140°F, which the data clearly indicate was not the case; or (2) the discharge thermistor (T4) indicated a discharge temperature in excess of 220°F. Apparently, HPWH 7's T4 thermistor began indicating higher discharge temperature readings after the ambient humidity was raised to 80%, causing compressor duty cycles to end earlier (before the T1 thermistor reached 125–130°F). When the voltage was reduced in Stage 4, higher compressor operating temperature likely caused the discharge temperature to reach the level even earlier in the duty cycle at which T4 triggered the control to turn off the compressor (at even lower T1 values and lower discharge pressure and tank temperature). This scenario also explains the reduced discharge pressure, T1 readings, and tank temperature in Stages 3 and 4 for unit 1. Observation of end-of-cycle LED condition for unit 1 confirmed that unit 1 operated similarly to unit 7 during Stage 3 and 4 and intermittently at the beginning of Stage 2 as well.

Loss-of-Charge Impacts

As noted in Sect. 4.1, units 7 and 9 both experienced refrigerant leaks during the durability run. Figure 4.6 illustrates end-of-cycle discharge pressure, average water temperature, and T1 thermistor values for these two units. Loss of charge (from about 16 oz as filled to about 3 oz) in unit 7 is evident in the continuously decreasing discharge pressure over the first 6–7 weeks of the test period.

When the leak was repaired and the unit recharged with 16 oz of R-134a, discharge pressure levels recovered to the 380–420 psig range (except for Stage 4, as discussed above). End-of-cycle discharge temperature for unit 7 ranged from 200–210°F after recharging and remained at this level until the end of the durability run. The refrigerant loss in unit 9 occurred much more gradually and did not become apparent until near the end of the testing, when we noticed the efficiency of unit 9 was significantly lower than that of the others (see the next section, “Unit Efficiencies”). Unit 9 lost a total of about 60% of its charge (from 16 oz down to about 6 oz). The discharge pressure data in Fig. 4.5 suggest that most of the loss happened in two shorter time periods (between days 140–150 and days 275–285), judging from the relatively rapid drop in pressure levels at those times.

Unit Efficiencies

Approximate EF values were measured for each of the test HPWHs at several points during the durability test run. These tests were run according to the DOE Simulated Energy Use Test procedure (*Federal Register* 1998); however, the ambient temperature and supply voltage conditions varied from the standard values specified in the procedure. The supplemental hot water feedback flow was inactive during these EF tests. Summary results are given in Table 4.3. Tests were run for each unit in Stages 2–5 of the protocol. We intended to run a test during Stage 1 as well, but unfortunately we did not get the EF test control routine operable soon enough to do so.

Table 4.3. Energy factor test results at various points in the durability test sequence

Unit	Stage 2 April 2001 75–80°F 80% RH 240 V	Stage 3 June 2001 100°F 50% RH 240 V	Stage 4 August 2001 100°F 50% RH low voltage	Stage 5 October 2001 67.5°F 50% RH 240 V
1	2.19	2.49	2.14	1.95
2	2.38	2.25	2.16	1.80
3	2.36	2.52	2.44	1.99
4	1.90	2.01	2.16	2.08
5	2.37	2.28	2.53	2.14
6	2.06	2.41	2.36	2.05
7	2.35	2.36	2.10	2.08
8	2.15	2.31	2.23	1.87
9	2.10	1.67	1.85	1.48
10	2.18	2.47	2.19	1.91

Apart from unit 9, whose EF performance in Stages 3–5 was degraded as a result of refrigerant loss, the EF results vary little from unit to unit. Unit 5 seemed to perform a little better than the rest, on average, and unit 4 a little less well, but not significantly so. The percentage difference between high EF and low EF in each of the tests (excluding unit 9) ranged from 16% to 20%. Results of calorimeter tests of several of the compressors are given in Table 4.4 along with the manufacturer’s reference values (for new compressors). None of the compressors tested shows any capacity or EER loss compared with the “as new” performance levels. In the case of compressor 9, this fact provides further evidence that the degradation in performance seen in HPWH unit 9 was due to its refrigerant loss and not to any system or component degradation.

Table 4.4. Compressor calorimeter test results (130°F saturated condensing temperature, 45°F saturated evaporating temperature, 220 V, 60-Hz power supply)

Compressor	Test date	Capacity (Btu/h)	Power (Watts)	EER (Btu/W·h)
1	2/15/02	3560.5	443.8	8.02
2	12/4/01	3509.6	445.5	7.88
4	2/15/02	3691.2	444.6	8.29
7	12/4/01	3573.4	461.2	7.75
8	2/15/02	3511.3	451.5	7.78
9	2/15/02	3621.6	450.4	8.05
10	2/15/02	3587.4	450.0	7.99
Reference[a]	–	3530.0	452.6	7.80

[a]Manufacturer's reference performance in "as new" condition.

The ambient conditions for the Stage 5 tests are closest to those prescribed for the standard DOE test procedure. EFs for that series ranged from 1.80 to 2.14 for the nine good units. These values are low compared with the value reported for a prototype laboratory unit that achieved an EF of 2.47 in early 2000. (Tomlinson 2000) However, they compare fairly well with the performance values being achieved by several units that are undergoing field testing in residences in several locations throughout the United States. Average overall efficiency values of the field units for periods with little or no resistance element usage have ranged from about 1.8 to 2.4 (Murphy 2002). These field-measured efficiencies are subject to many variables, including thermostat setting, water use patterns, ambient temperature conditions, and entering water temperature, to name a few. The ambient temperature for the field test units was fairly close to the conditions of Stage 5 in the durability testing, while the field thermostat settings were generally somewhat lower than those maintained for the durability test units.

It is difficult to determine from Table 4.3 data alone whether any of the units suffered any degradation in performance, other than unit 9, which lost part of its refrigerant charge. However, we were also able to examine steady-state tank heat-up performance during each EF test. Table 4.5 provides 15-min snapshots of the steady-state performance of each unit during the Stage 2 and Stage 5 EF tests with COP, power consumption, refrigerant operating conditions, and performance change from Stage 2 to Stage 5. These data were taken for time periods during which the average tank water temperature was approximately the same during each test. The loss of charge in unit 9 is quite evident from its elevated suction temperature compared with the other units in the Stage 5 tests.

The "measured" COP values in Table 4.5 were computed from the data as follows:

$$COP = (m \times c_p \times \Delta T)/(Power \times 0.25)/3.412 \quad (1)$$

where:

m = mass of water in tank = 45.9 gal × 8.25 lb/gal,
c_p = specific heat = 1 Btu/lb/°F,
ΔT = change in average tank water temperature over 15-min period, °F,
Power = average unit power draw in Watts during 15-min period,
0.25 = duration of test period in hours,
3.412 = conversion from Btu/h to Watts

The drop in these measured steady-state COP values from Stage 2 tests to Stage 5 tests (exclusive of unit 9) ranges from a low of 9% to a high of 29%. Thermodynamic cycle calculations, using the

Table 4.5. Steady-state unit efficiencies and heat pump operating conditions, 15-minute data averages

Unit	Stage 2 April 2001: 75–80°F dry bulb, 80% RH							Stage 5 October 2001: 67.5°F dry bulb, 50% RH							% COP drop Stage 2 to Stage 5 measured	%COP drop Stage 2 to Stage 5 Calc'd[b]
	COP meas.[a]	Power (watts)	Avg tank temp. (°F)	Pdisc (psia)	Tdisc (°F)	Psuct (psia)	Tsuct (°F)	COP meas.[a]	Power (Watts)	Avg tank temp. (°F)	Pdisc (psia)	Tdisc (°F)	Psuct (psia)	Tsuct (°F)		
1	2.60	579	130.3	302	180	69	63	1.93	530	130.3	330	190	57	54	25.8	16.2
2	2.53	580	131.7	280	193	67	64	1.80	543	131.9	304	194	57	58	28.9	17.2
3	2.26	590	131.9	285	200	69	69	2.05	520	131.9	290	211	54	54	9.3	11.6
4	2.33	590	132.3	275	200	69	61	1.84	535	132.6	310	213	53	53	21.0	23.9
5	2.70	575	132.4	305	205	64	60	2.07	535	133.0	305	215	55	54	23.3	7.4
6	2.69	545	124.0	255	210	65	63	2.04	523	124.1	290	212	53	55	24.2	24.7
7	2.34	585	129.3	325	192	66	63	1.97	540	129.3	358	206	54	53	15.6	17.7
8	2.57	585	132.6	290	180	70	66	1.83	550	132.1	320	188	58	58	28.8	19.3
9	2.64	570	132.4	268	200	74	65	1.15	579	132.6	280	208	74	69	56.4	3.2
10	2.69	595	134.6	335	214	67	62	1.91	556	134.8	360	220	55	55	29.0	19.6
Average refrigerant conditions[c]				295	197	67	63				319	206	55	55		

[a]Measured from tank temperature increase and average power consumption during 15-minute test period.

[b]Change in thermodynamic cycle COPs from Stage 2 to Stage 5; cycle COPs calculated from R-134a properties using measured pressures and temperatures and assuming 20°F subcooling at the condenser exit.

[c]Excluding unit 9 values.

refrigerant conditions measured and assuming 20°F of subcooling at the condenser exit, show calculated COP drops from Stages 2 to 5 ranging from about 7% to 25% (excluding unit 9). The range and the individual values for the thermodynamic cycle COP drops between stages are comparable with the measured COP decreases. Considering this steady-state performance comparison, the comparison with measured field test unit efficiencies, and the compressor test results, we conclude the likelihood is small that there was any degradation in performance for units 1–8 and 10 during the 10-month durability test period.

The degradation in unit 9 performance is considered to be largely (most likely entirely) due to its refrigerant loss. As noted in Sect. 4.1, the refrigerant leak was located in a fitting we had added to enable discharge pressure measurement and will not be part of the final production design.

4.3 Post-Test Compressor and HPWH Tank Examinations

Tank and Condenser Wrap Examination

The information presented in this section came primarily from John Hoyt (2001) of EMI (the HPWH manufacturer). Nine of the test HPWHs were cut apart as shown in Figs. 4.8, 4.9, and 4.10 to examine the post-test condition of the condenser and tank. A full report is included in Appendix B, and highlights are listed here.

Fig. 4.8. Heat pump water heater tank with insulation stripped away from the lower half to reveal condenser wrap.

Fig. 4.9. Top end cap of tank with dip tube (cold water inlet) and anode rod exposed.

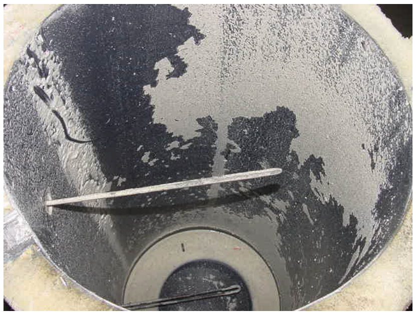

Fig. 4.10. Interior view of heat pump water heater tank.

These examinations have revealed that the condensers on all of the units remain tightly attached to the tank walls. The condenser wrap is anchored at each end by spot-welding or brazing the copper tubing to the tank wall. All of these braze points appeared tight and in good condition. The heat transfer mastic was of a creamy, peanut-butter-like consistency when first applied. After the durability run, the mastic was observed to be somewhat darker in color and to have hardened to a wax-like consistency. There were no signs of crumbling or other breakdown.

The glass lining on the inside surface of the tanks appeared to be in very good shape. There was a slight change in the texture of the lining inside the tank opposite the two condenser braze points. However, the lining did not become separated from the tank wall and did not show evidence of corrosion at the braze points. There was a light, uniform coating of scale adhering to the tank inside wall [most likely $CaCO_3$ (calcium carbonate) or some other precipitate related to hard water]. However, since the durability testing was conducted using a closed test loop in which the water was repeatedly recirculated, this result is not considered to be indicative of what may happen in real residential installations where the water has a high degree of dissolved mineral content.

There was evidence of moisture (rust) inside the electric junction boxes in front of the evaporator, and around the high-temperature cutout switch (inside the upper element access panel) on several of the units. As noted previously, this was most likely due to the condensate drainage problems we experienced with our test setup. We do not think these problems were caused by inadequacies in the HPWH evaporator drain pan design; however, the design is being changed to provide a larger drain opening and a backup overflow drain to channel water away from the HPWH in event of any overflow situation.

The foam insulation showed no evidence of degradation apart from some discoloration of the insulation that was in direct contact with the top of the condenser coil and the discharge line.

The anode rods experienced different levels of degradation ranging from being partially to completely dissolved.

Compressor Examination

The information in this section was obtained from teardown tests and oil analyses conducted by the compressor manufacturer (Embraco) after the durability run was completed. (Doering 2001, 2002a, and 2002b) Nine compressors were sent to Embraco. Eight of these were sent to the headquarters in Brazil for examination by company experts in compressor life testing and wear. These were also calorimeter-tested to check for any performance deterioration (see Table 4.4). The ninth compressor (unit 5) was cut apart in the presence of ORNL staff at Embraco facilities near Atlanta.

Figures 4.11 through 4.22 illustrate the condition of compressor 5 after the durability run. Figure 4.11 is a photo of used oil taken from compressor 5. Analyses done on used oil samples from the eight units sent to Brazil indicated no change in color compared with new oil and no appreciable oil degradation apart from an increase in acid levels in some of the samples (see Table 4.6). However, the acidity increase was not great enough to indicate oil degradation, according to Embraco’s analyst.

Figures 4.12 and 4.13 illustrate the suction side and discharge side, respectively, of the valve plate. The suction side shows some discoloration around the discharge port, most likely from oil residue deposited during high-temperature operation. Our test data show that the discharge temperature consistently peaked at around 231°F for this unit in the latter half of the durability run (Fig. A.5). There was no visual indication of excessive wear on the piston (Fig. 4.14), cylinder walls (Figs. 4.15 and 4.16), or bearing surfaces (Fig. 4.17). The suction and discharge reed valves (Fig. 4.18) were in good shape with no indication of deformation. There was some discoloration on the discharge valve similar to that observed around the discharge port in Fig. 4.12.

The motor and windings were clean, with no evidence of insulation degradation or scorching (Fig. 4.19). The contact points from the start relay (Fig. 4.20) and over-temperature protection relay (Fig. 4.21) showed some carbon buildup, as might be expected after 7300+ starts, but still functioned adequately (the motor was started up a few times before disassembly).

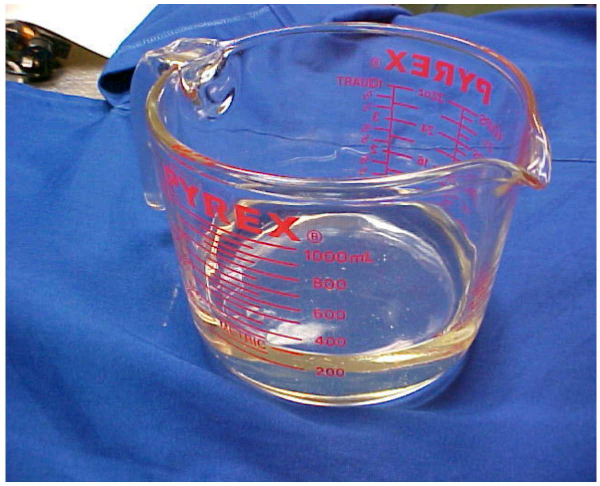

Fig. 4.11. Oil charge removed from compressor 5.

Table 4.6. Oil analysis results

Oil sample	Acidity (mgKOH/g)	Color[a] index	FTIR[b]	GC-MS[c]	Miscibility
Used oil – compressor 1	0.010	0.5	Ester oil	Freol Alpha 22E oil[d]	Transparent solution
Used oil – compressor 2	0.017	0.5	Ester oil	Freol Alpha 22E oil	Transparent solution
Used oil – compressor 3	0.020	0.5	Ester oil	Freol Alpha 22E oil	Transparent solution
Used oil – compressor 4	0.030	0.5	Ester oil	Freol Alpha 22E oil	Transparent solution
Used oil – compressor 7	0.015	0.5	Ester oil	Freol Alpha 22E oil	Transparent solution
Used oil – compressor 8	0.010	0.5	Ester oil	Freol Alpha 22E oil	Turbid solution
Used oil – compressor 9	0.010	0.5	Ester oil	Freol Alpha 22E oil	Transparent solution
Used oil – compressor 10	0.020	0.5	Ester oil	Freol Alpha 22E oil	Transparent solution
New oil	0.010	0.5	Ester oil	Freol Alpha 22E oil	Transparent solution

[a]The color is the same as that of virgin oil.

[b,c]The FTIR (Fourier transform infrared) spectra showed the characteristic band of ester oil and the CG-MS (gas chromatography) identified the oil as Freol (brand) Alpha 22E.

[d]Alpha 22E oil properties

density at 20°C, g/cm^3:	0.950
viscosity at 40°C, cP:	20.88
viscosity at 40°C, cSt:	22.32
viscosity at 100°C, cP:	3.80
viscosity at 100°C, cSt:	4.27
flash point, °C:	205.0
fire point, °C:	221.0
dielectric strength, KV:	60.00
total acid number, mgKOH/g:	0.010
Hydroxyl value, mgKOH/g:	3.44
Color index:	0.50
IR spectrometry analysis:	polyolester oil
Miscibility test, HFC-134a, °:	–37.00

Source: Personal communications from R. Doering of Embraco NA to Van Baxter of Oak Ridge National Laboratory, December 14–18, 2001, and March 19, 2002.

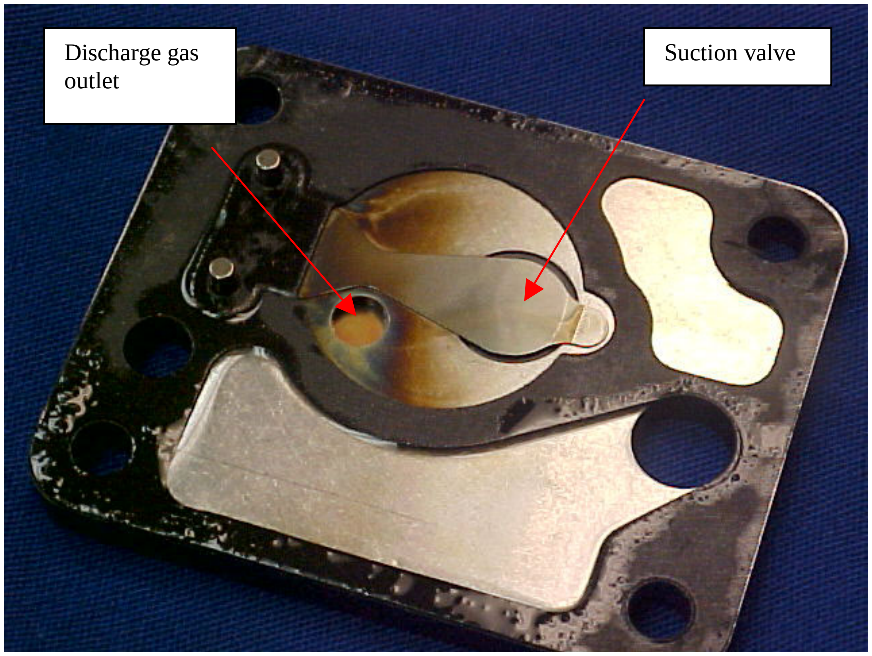

Fig. 4.12. Valve plate from compressor 5—bottom.

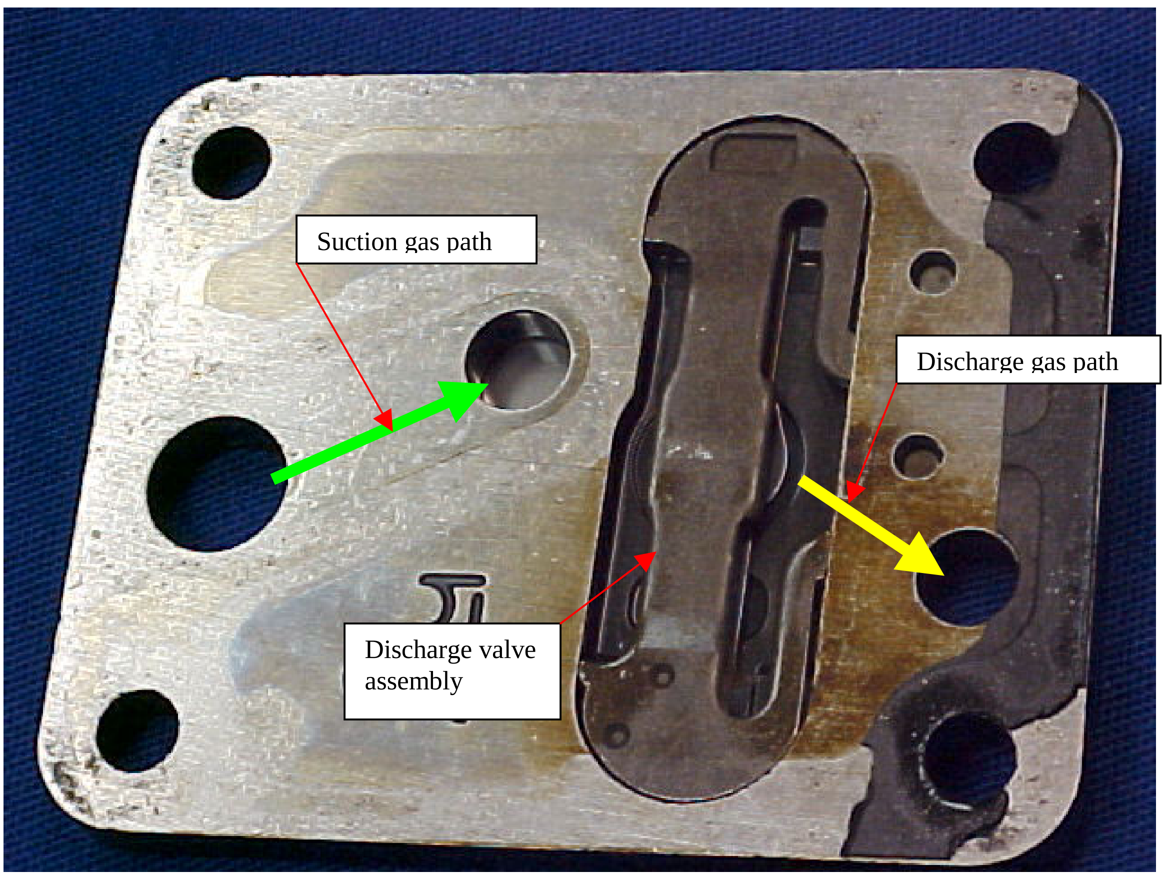

Fig. 4.13. Valve plate from compressor 5—top.

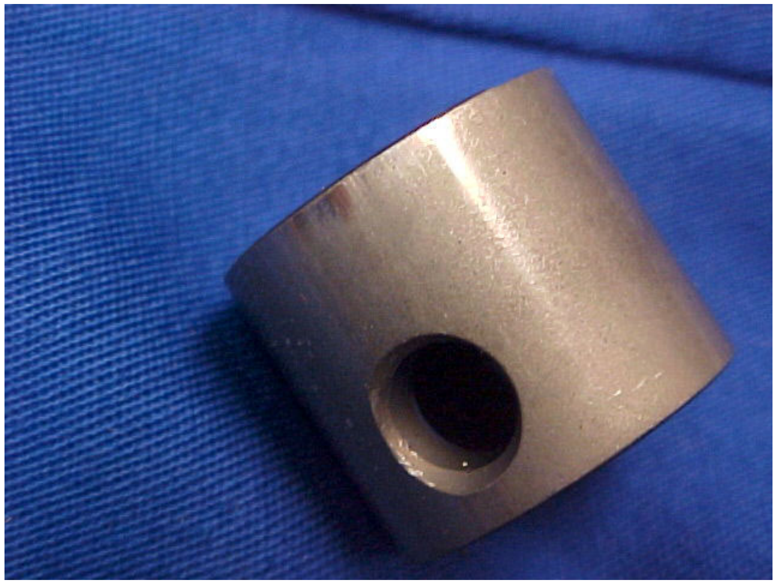

Fig. 4.14. Piston from compressor 5.

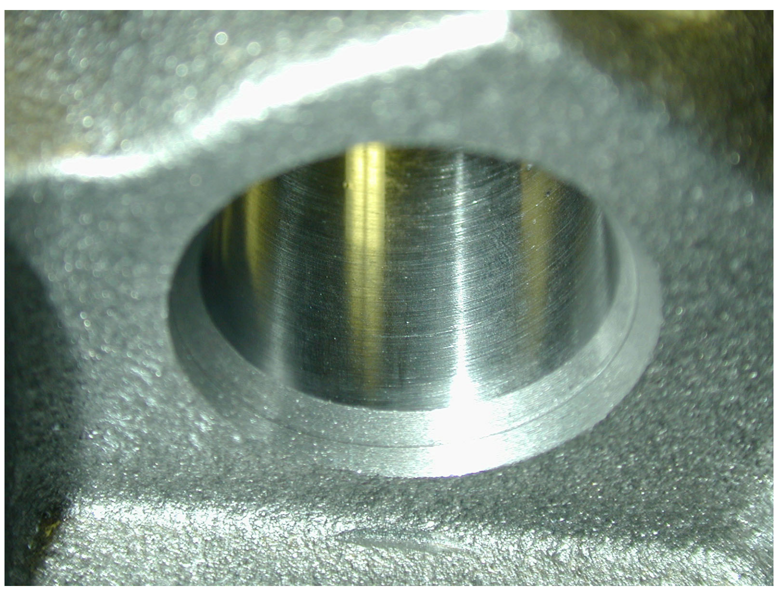

Fig. 4.15. Compressor 5 cylinder wall, view from bottom.

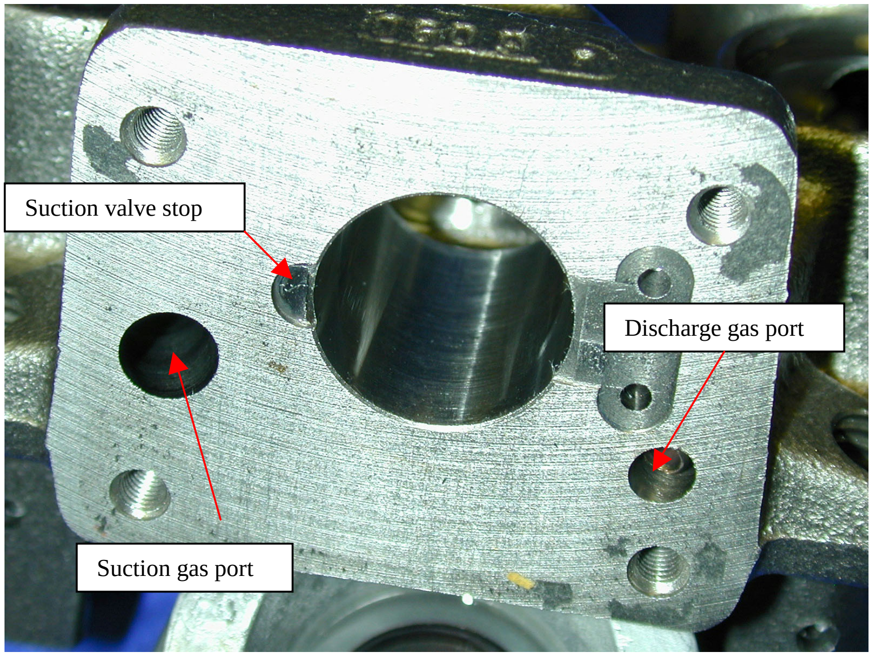

Fig. 4.16. Compressor 5 cylinder wall, view from top of cylinder block; note suction gas port to left of cylinder and discharge gas port to right of cylinder.

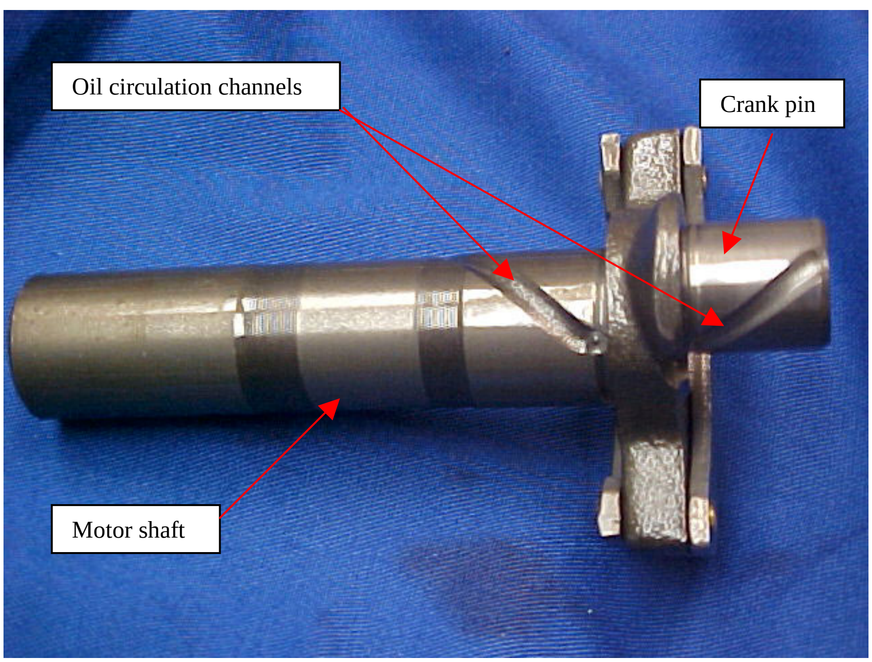

Fig. 4.17. Compressor 5 motor shaft. Piston rod crank pin is on right side of photo; grooves are for oil circulation.

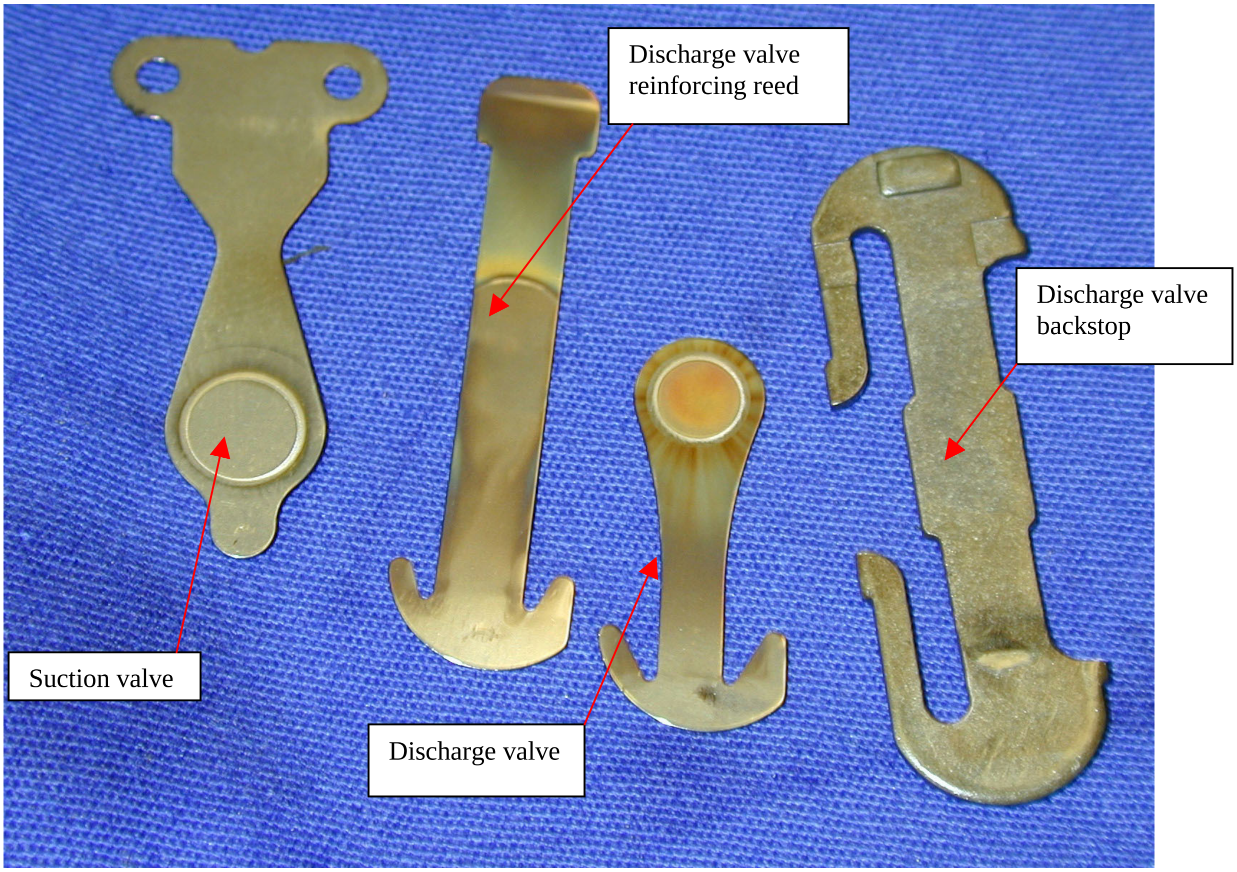

Fig. 4.18. Compressor 5 suction valve (far left) and discharge valve assembly (three pieces, left to right: reinforcing reed, valve, backstop).

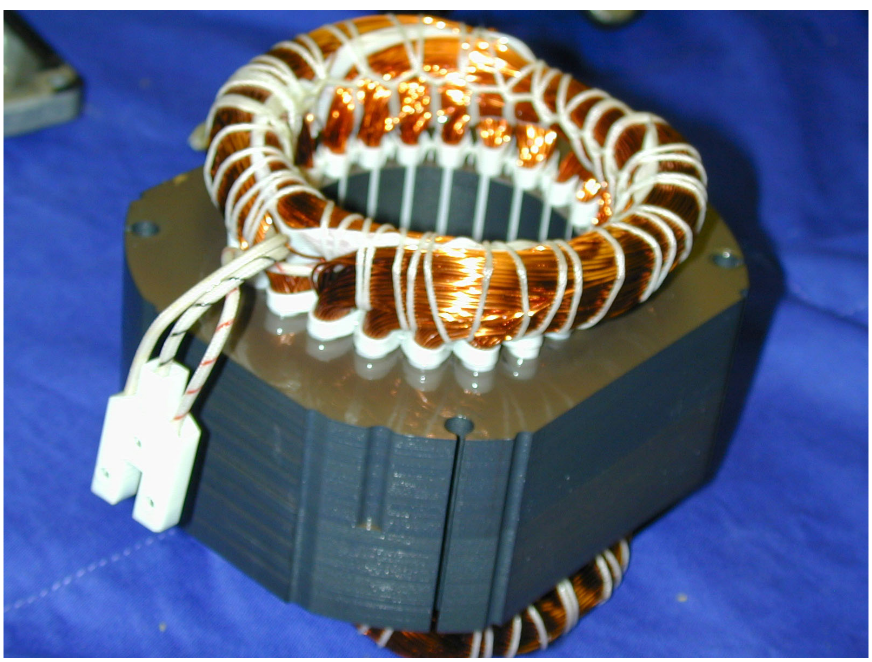

Fig. 4.19. Compressor 5 motor windings.

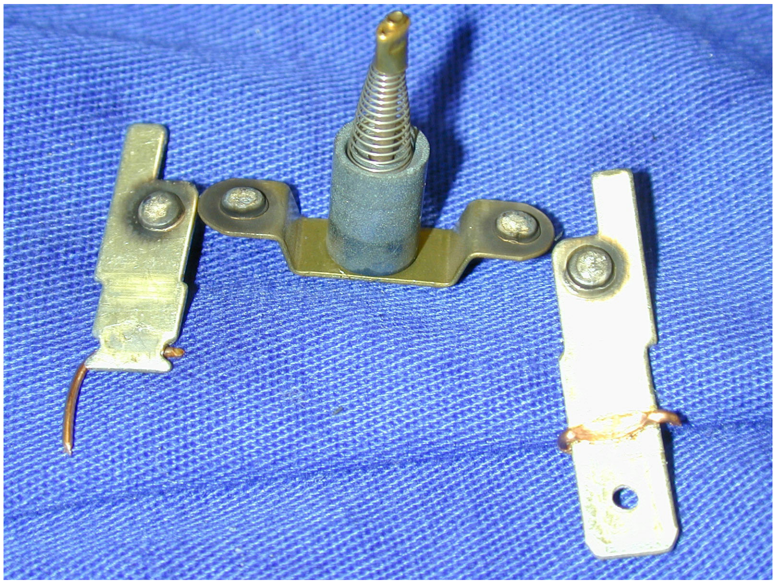

Fig. 4.20. Compressor 5 start relay points.

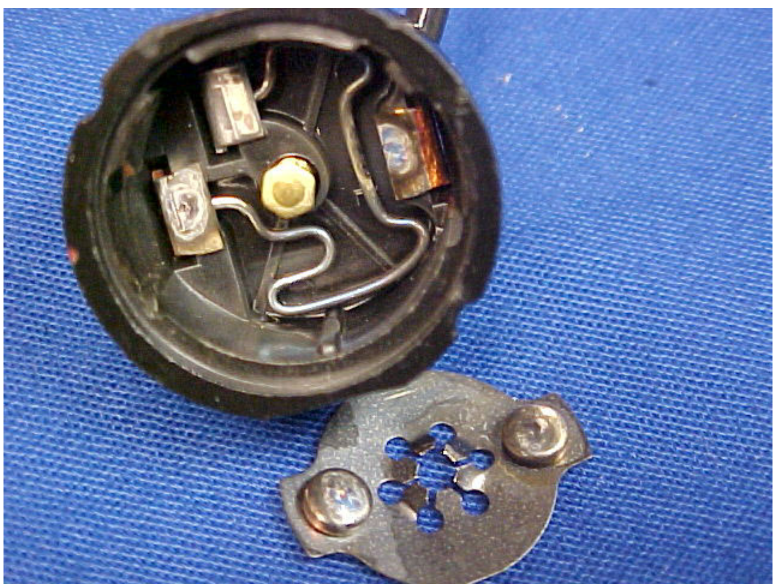

Fig. 4.21. Compressor 5 over-temperature cutout relay.

The most worrisome observation was that there were small (iron and copper) particles in the oil at the bottom of the compressor can (Fig. 4.22). These were also found in the compressors examined at Embraco's Brazil facilities. As noted earlier, there was no evidence of scoring on the piston, cylinder wall, or bearing surfaces of Compressor 5 as might be expected if these particles were being circulated. They probably stayed at the bottom of the can during the test period or were lodged on the can wall during operation and migrated to the sump because of vibration during shipping to Embraco. Embraco personnel did not think it likely that these particles accumulated during the production process. The HPWH systems were opened up several times to add, repair, or replace the pressure transducers and other test fittings. It is probably more likely that these particles were introduced into the system during those operations.

Fig. 4.22. Debris particles found in bottom of oil sump on compressor 5.

Table 4.7 provides the compressor manufacturer's qualitative assessment of the condition of several of the test unit compressors following the durability run. Most of the compressor components in this relatively small sample were judged to have experienced only "mild" wear. The bushings (one case of severe and two cases of moderate wear) and crankpins (three cases of moderate polishing) are the most notable exceptions. Figure 4.23 is a photo of the compressor assembly showing the locations of the bushing and crankpin. Figure 4.24 shows close-up photos of the interior surface of bushing 2(a)

Table 4.7. Compressor post-test component wear evaluation (Wear codes: L-slight; 1-mild; 2-moderate; 3-severe)

Compressor	1	2	3	4	7	8	9	10	Wear severity index[a]
Component									
Piston top	2 PO[b]	1	1	1	1	1	1	1	1.60
Piston bottom	1	1	1	1	1	1	1	1	1.00
Cylinder top	1	1	1	1	1	1	1	1	1.00
Cylinder bottom	1	1	1	1	1	1	1	1	1.00
Thrust collar	1	1	1	2 PO	1	1	1	1	1.60
Thrust bearing	1	1	1	1	1	1	1	1	1.00
Crankshaft journal	1	1	1	1	1	1	1	1	1.00
Main bearing	1	1	1	1	1	1	1	1	1.00
Connecting rod end	1	1	1	1	1	1	1	1	1.00
Wrist pin	1	1	1	1	1	1	1	1	1.00
Bushing	1 PO	3 PO	2PO	1 PO	1 PO	2 PO	1 PO	1 PO	2.60
Crank pin	1	2 PO	2 PO	1	1	1	1	2 PO	2.29
Discharge valve	LCL[c]	LCL	LCL	LCL	LCL	LCL	LCL	LCL	1.00
Springs	1	1	1	1	1	1	1	1	1.00
Spring holder	1	1	1	1	1	1	1	1	1.00

[a]Weighted average value of wear for component with moderate and severe occurrences weighted more heavily than mild or slight occurrences

[b]PO = polishing

[c]CL = coloring

Source: Personal communications from R. Doering of Embraco NA to Van Baxter of Oak Ridge National Laboratory, December 14–18, 2001, and March 19, 2002.

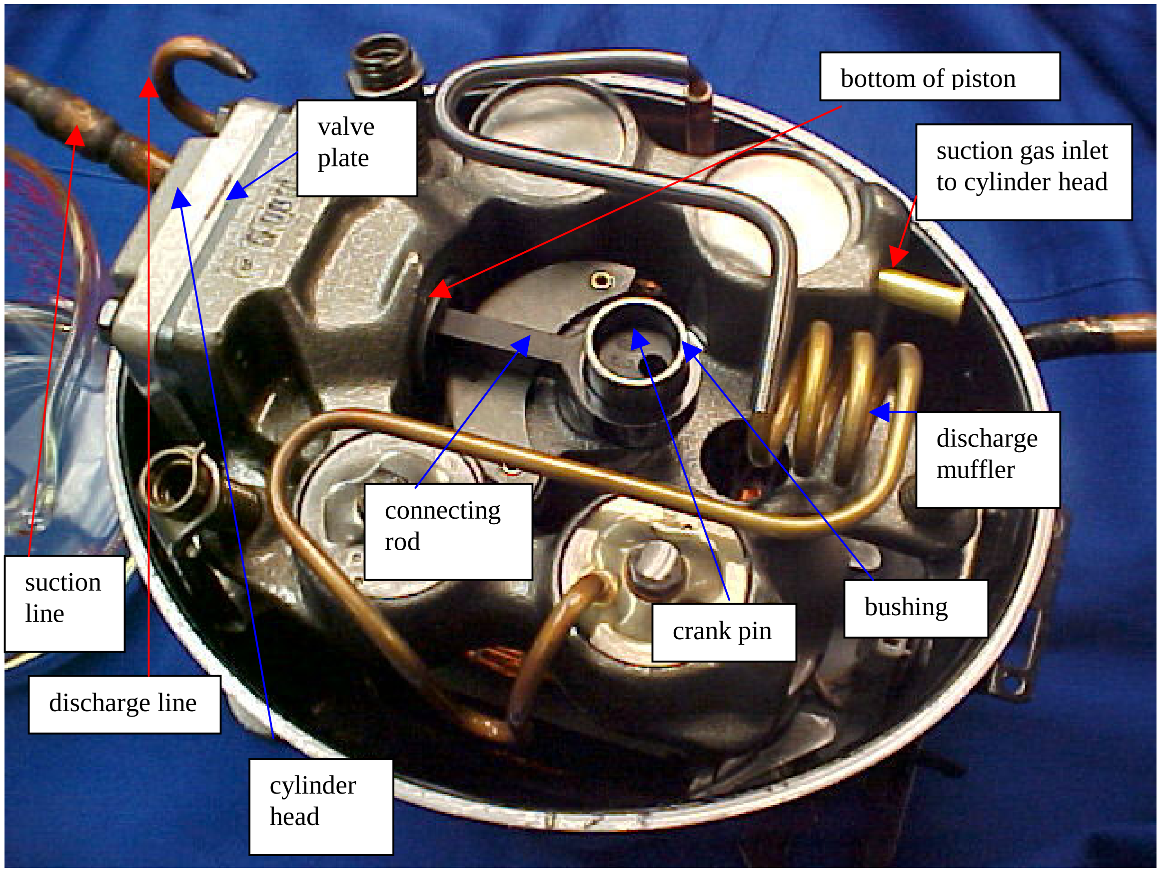

Fig. 4.23. Compressor 5 view showing top end of compressor assembly with crank pin, bushing, and piston connecting rod clearly visible.

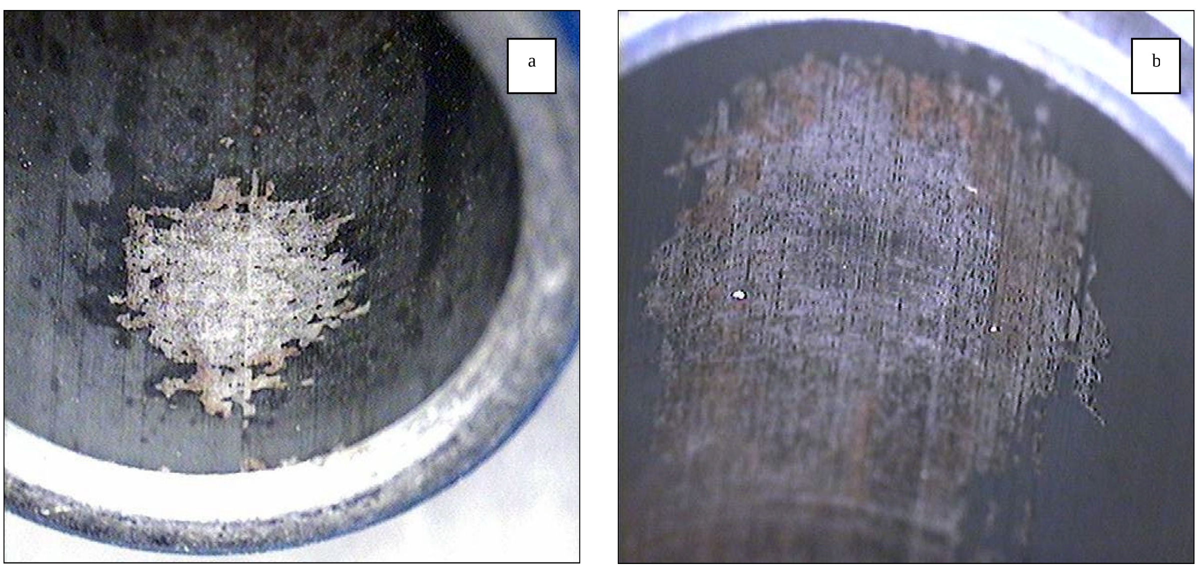

Fig. 4.24. Compressor bushings 2(a) and 8(b) showing severe (2) and moderate (8) polishing with detachment of the steam-treated layer (Fe_3O_4). This detachment was verified at least in a low intensity in all of the bushings, but it reached the moderate level in compressors 3 and 8 and severe level in compressor 2.

and bushing 8(b), respectively, illustrating the wear pattern. The bushing from compressor 5 (Fig. 4.25) exhibited a similar wear pattern but to a much less severe degree than the bushings from either unit 2 or 8. Both the bushing and the crankpin (and presumably many of the other load-bearing components) are given a steam treatment or hardening during manufacturing, which puts a layer of Fe_3O_4 on the material surface. In the case of bushing 2, this hardened layer has been completely worn away, hence the "severe" wear rating. The "moderate" wear rating for the #2, #3, and #10 crankpins and the #3 and #8 bushings indicated that, on these components, this layer had some significant polishing but was not worn through.

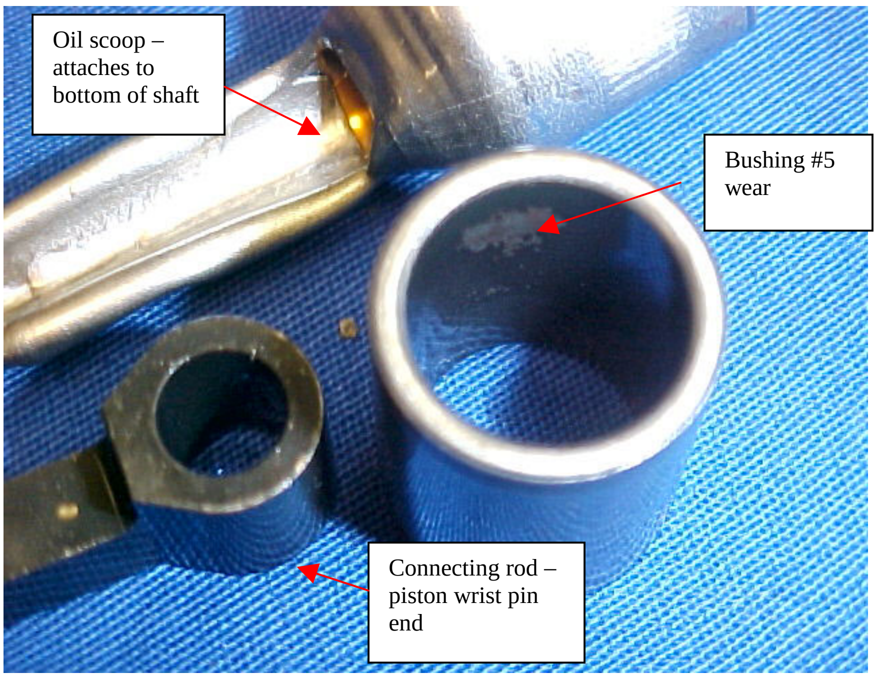

Fig. 4.25. Bushing from compressor 5 showing polishing wear on inside surface (area in contact with crank pin).

The discharge valves in all of the test units exhibited a light level of coloring similar to that observed for unit 5 (Fig. 4.18). Not noted in Table 4.7 is the fact that some of the electrical insulation in compressor 9 experienced a slight degree of embrittlement.

The wear severity index (WSI) in the last column of Table 4.7 is an approximate average of the wear experienced by each of the compressor components. It is calculated using the following equation developed by the manufacturer for wear evaluation of the manufacturer's own compressor life test results (Doering 2002a).

$$WSI(i) = [n_1(i) + 9n_2(i) + 16n_3(i)]/[n_1(i) + 3n_2(i) + 4n_3(i)], \qquad (2)$$

where

i = ith component,

n_1 = number of mild wear occurrences,

n_2 = number of moderate wear occurrences,

n_3 = number of severe wear occurrences,

$n = n_1 + n_2 + n_3$ = total number of compressors in test sample.

This approach is designed to weight severe and moderate occurrences much more heavily than mild occurrences in the WSI. If WSI across a given sample is between 1.00 and 2.00, the result is considered good. A range of 2.01 to 3.00 is considered tolerable, and anything over 3.01 is considered unacceptable. Based on computed values of this index, the wear results for all components fall into either the “good” or “tolerable” ranges. However, the compressor manufacturer’s (Embraco’s) report states “The Tribology Lab considers these results unsatisfactory, because of the moderate to severe wear level in the bushing.” (Doering 2002b) Embraco’s concern is that each bushing had the same wear pattern, some to a greater degree than others. Because the sample size is small, and every compressor was affected the same way, Embraco is reluctant to give a “passing grade” to the application. EMI (the HPWH manufacturer) has made arrangements to ship a complete unit to Embraco for further evaluation. It is hoped this will yield some insight into the causes of the observed bushing wear pattern. In the meantime, Embraco will be implementing an improved surface treatment on the bushing to safeguard against this problem in the future.

5. OBSERVATIONS AND RECOMMENDATIONS

Based on the operational experience and test results from the 9.5-month durability run, the following observations and recommendations are noted.

The basic heat pump system hardware seems to be very robust. During the entire durability run (and the 3-month pre-test checkout period prior to the durability testing), no compressor, fan, or power-switching relay failures were experienced. One compressor failed during calorimeter testing by the compressor manufacturer after the durability test run. One TXV had to be replaced because of a blocked bleed port. Two units (7 and 9) experienced refrigerant leaks. However, the leaks occurred at solder joints in pressure transducer fittings that were added to the compressor discharge lines to enable pressure measurement. These extra fittings will not be included in the normal production units.

1. The units' efficiency compared with an EWH baseline is very good. Approximate EFs measured at different times throughout the durability run were at least 2× those of conventional EWHs. EFs of the durability units were also comparable to the average efficiencies experienced by several similar units being field tested in about 18 actual residential applications scattered throughout the United States.
2. The efficiency of nine of the ten durability test HPWHs did not appear to have degraded significantly as a result of undergoing more than 7000 repetitive duty cycles. One unit (unit 9) experienced measurable degradation because of a refrigerant leak that was not discovered until after the durability run had ended.
3. The approximately 7300 compressor duty cycles accumulated by the durability test units is reasonably representative of about 8–10 years of compressor duty cycling for applications having low-to-moderate hot water demand (<500 gallons/week), based on the range of weekly cycles experienced by three units now undergoing field tests. Applications with high hot water usage can experience a greater number of compressor cycles, which are caused by more numerous heavy draws that require the upper element to energize to recover the top part of the tank. For units with such heavy usage, 7300 compressor cycles may be more indicative of perhaps 5–9 years of operation.
4. Post-test examination of the test HPWH condenser wrap noted no apparent degradation in the condenser/tank attachment (no sign that condensers had separated from the tank wall). The heat transfer mastic applied between the condenser and tank wall for good thermal contact had hardened somewhat from its original consistency, but it showed no signs of crumbling or other obvious degradation. Eight of the ten compressors were torn apart by the compressor manufacturer to check for wear. One component, the crank pin bushing (see Figs. 4.23 and 4.24), experienced a consistent wear pattern in each compressor. In some cases, the wear was severe enough to degrade the steam-hardened surface layer of the bushing. Because the sample size was small, and every compressor was affected the same way, the compressor maker was reluctant to give a "passing grade" to the application. The HPWH manufacturer and the compressor manufacturer are conducting further evaluations that it is hoped will yield some insight into the causes of the observed bushing wear pattern. In the meantime, the compressor maker will be implementing a new surface treatment on the bushing to safeguard against this problem in the future.
5. Because of a poor drainage system in our test apparatus, some of the units experienced overflows from the evaporator condensate drain pans. This overflow generally leaked into the electrical connection (junction box) at the top of the water heater tank and down through the insulation to the upper element access area. The leakage caused a failure of the high-temperature switch in unit 9. Although the overflows experienced in the durability testing were not due to any flaw inherent in the HPWH design, they do point out a potential problem that could occur in field installations should the HPWH drain pan become blocked.

6. The first-generation control system included on the durability units experienced many problems.
    - Sixteen temperature input sensors (thermistors) failed. Since there were four thermistors for each HPWH, this represents a 40% failure rate. All of these failures are related to either a short circuit or open circuit in the spliced connections in the thermistor lead wires of the original design.
    - One unit (unit 3) experienced failures of two control boards.
    - A number of the units were found to have thermostats (variable resistance potentiometers) that were not secured properly when delivered.
    - During pre-test checkout, severe problems were discovered with fluctuating, noisy, electric signal inputs from the temperature sensors and the thermostats. Much of this was determined to be due to 60-Hz noise, which was eliminated by grounding the boards to the HPWH frame. However, random noise spikes continued to occur throughout the checkout period and the durability run. The primary cause of these random spikes is thought to be that the control (low-voltage) wires were bundled together with the power (high-voltage) wires in the durability test units. This problem was exacerbated in the first-generation control software by the fact that the program made control decisions based on only single readings of the control inputs. A single errant input could therefore cause an anomalous control action to occur.

The field test units have experienced many of these same problems, also, and all have been communicated to the HPWH manufacturer. As a result of the durability and field-testing experience, a number of fixes have been developed and implemented for the ultimate production units. These include the following.

- The thermistor sensors have been upgraded to a more rugged (heavier) model with heavier-gauge lead wires, and splices have been eliminated from the sensor leads.
- Sensor cables have been rerouted away from the high-voltage power wiring to attempt to eliminate or at least reduce the occurrence of random noise incidents.
- A new control board design has been implemented, primarily to reduce manufacturing costs, but also to incorporate features to improve its reliability and to reduce susceptibility to random noise.
- The control program has been revised to sample the thermistor and thermostat inputs about 32 times and make control decisions based on the average value. This revision should help mask the impact of a single aberrant reading.
- The evaporator condensate collection and drain pans have been redesigned to incorporate a larger drain port and an overflow port to direct condensate away from the unit’s electrical connections in the event of an overflow incident.
- A more vigorous factory quality assurance program has been implemented to eliminate or minimize occurrences such as unsecured thermostats, control boards, and similar items.

In addition to the above measures, it is recommended that the low-voltage thermistor and thermostat leads be shielded as an additional measure to guard against random noise events.

ACKNOWLEDGMENT

The author wishes to acknowledge first of all the support of Esher R. Kweller of DOE for his enthusiastic support of this durability testing effort. Thanks are also due to John Tomlinson for his perseverance and leadership of the HPWH development efforts at ORNL. Rick Murphy, ORNL, provided much invaluable assistance and advice during the pre-test checkout of the HPWHs and is to be credited with diagnosing and developing fixes for many of the electronic noise problems experienced by the first-generation HPWH control system. The efforts of Bob Zogg and Richard Williams of ADL in developing the solid-state control system for the HPWHs and for providing much of the text describing the controls operation in Sect. 1 of this report are much appreciated. John Hoyt of EMI oversaw production of the durability test units and has worked tirelessly to improve the control system for the production models. Hoyt also personally conducted much of the post-test examination work of the HPWH condensers and tanks (see Appendix B). Ryan Doering, Embraco NA, provided crucial assistance in the post-test examination of the HPWH compressors.

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APPENDIX A: END-OF-CYCLE TREND PLOTS FOR EACH TEST HEAT PUMP WATER HEATER

Figures A.1 through A.10 show end-of-cycle trends of discharge pressure and temperature, average tank water temperature, and lower tank thermistor (T1) temperature for each test heat pump water heater over the course of the durability run. It should be noted that the tank and discharge temperature data for approximately days 35–55 are suspect because of a problem with the temperature data processing hardware during that period. Also, the fluctuation in the discharge temperature data for unit 6 (Fig. A.6) was due to a loose wiring connection in that thermocouple that was not discovered until after the durability run.

Figure A1. Unit 1 end-of-cycle discharge pressure & temperature, tank temperature, and lower tank thermistor temperature

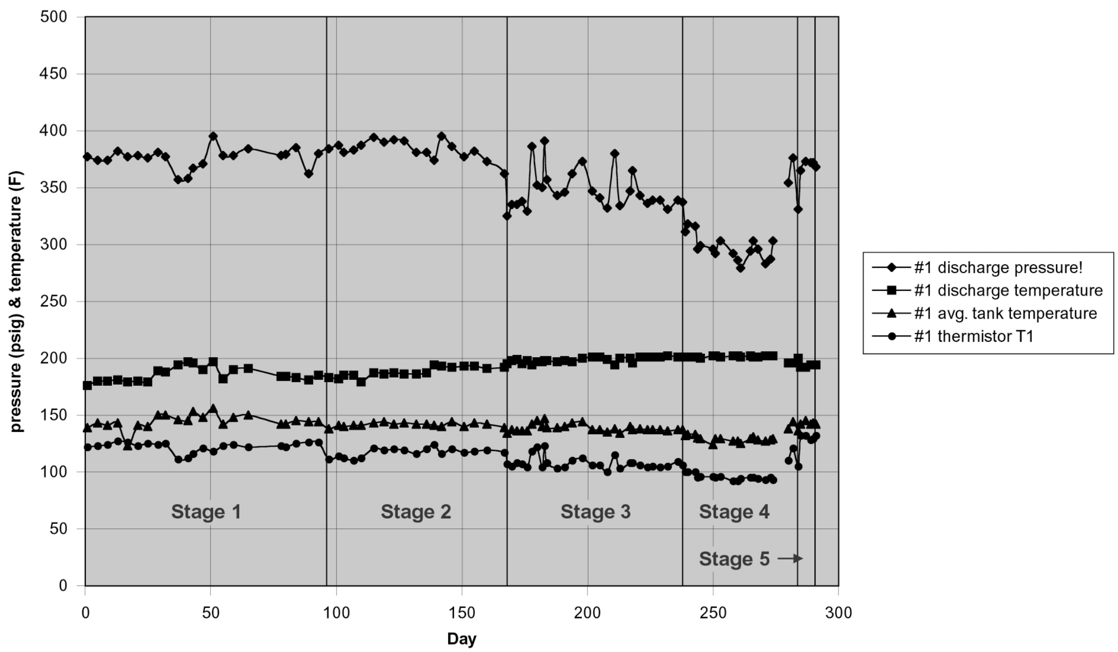


Figure A2. Unit 2 end-of-cycle discharge pressure & temperature, tank temperature, and lower tank thermistor temperature

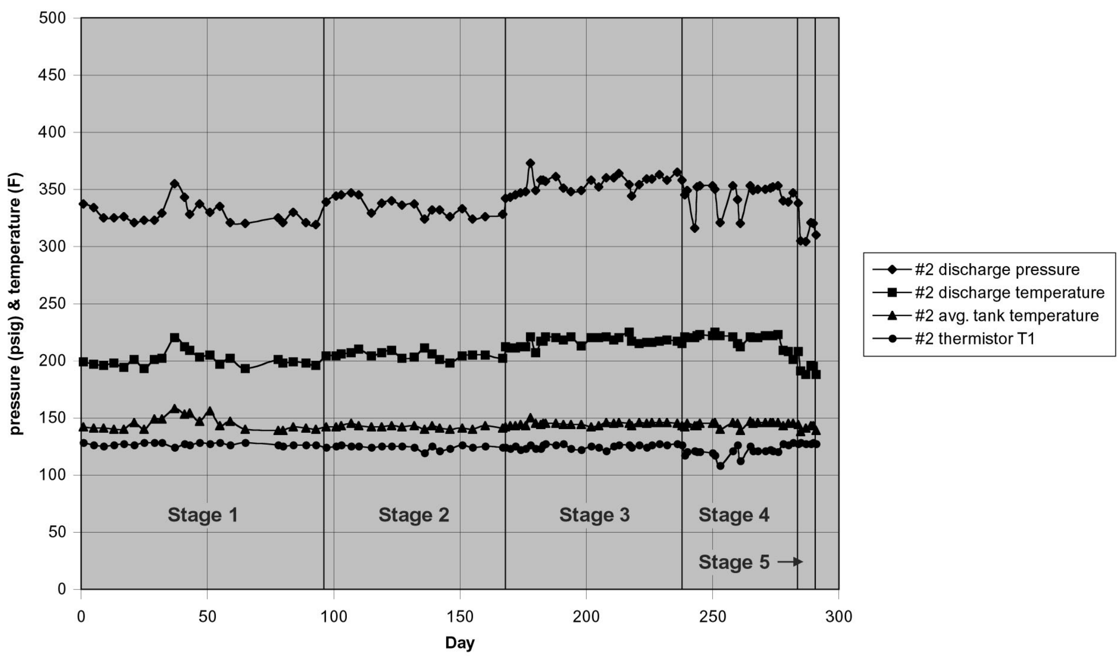


Figure A3. Unit 3 end-of-cycle discharge pressure & temperature, tank temperature, and lower tank thermistor temperature

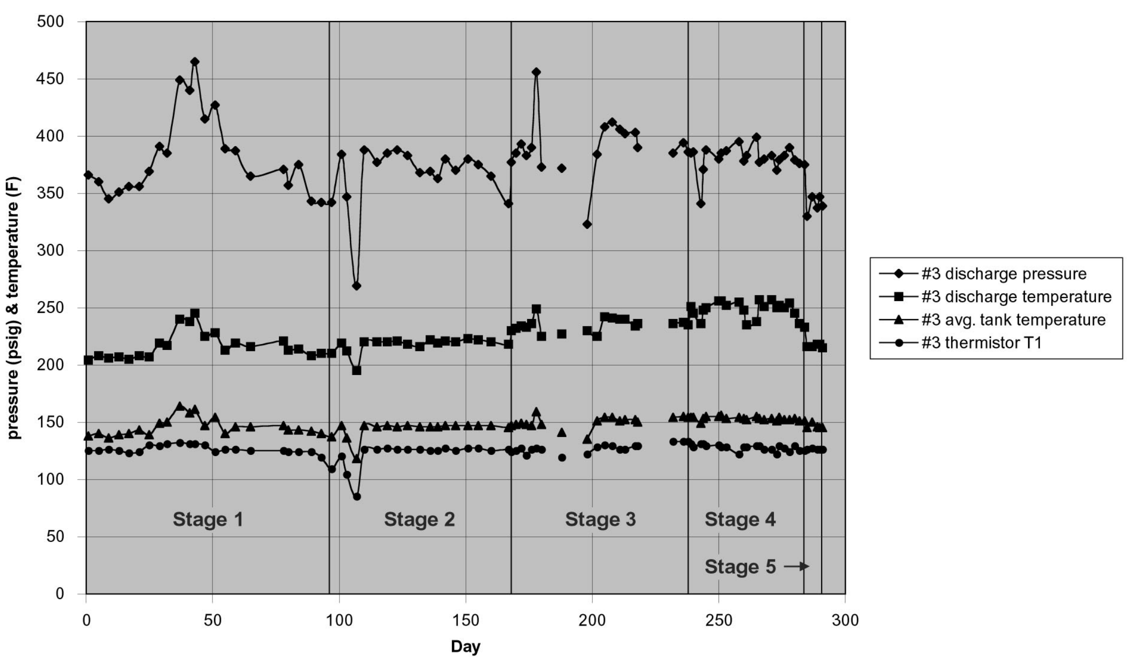


Figure A4. Unit 4 end-of-cycle discharge pressure & temperature, tank temperature, and lower tank thermistor temperature

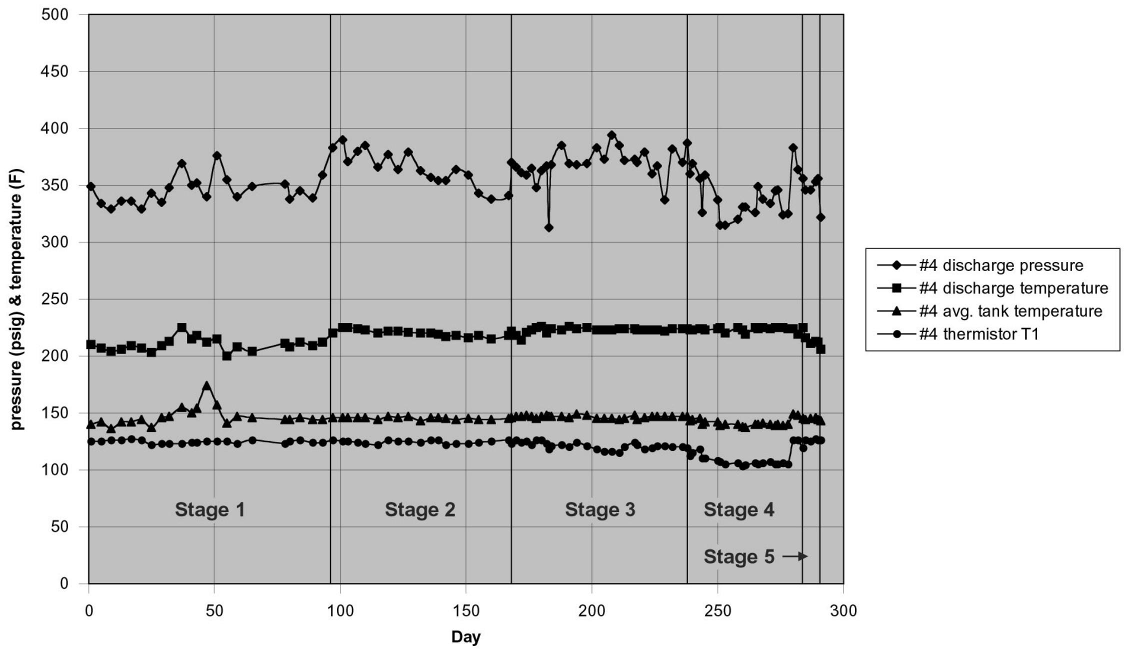


Figure A5. Unit 5 end-of-cycle discharge pressure & temperature, tank temperature, and lower tank thermistor temperature

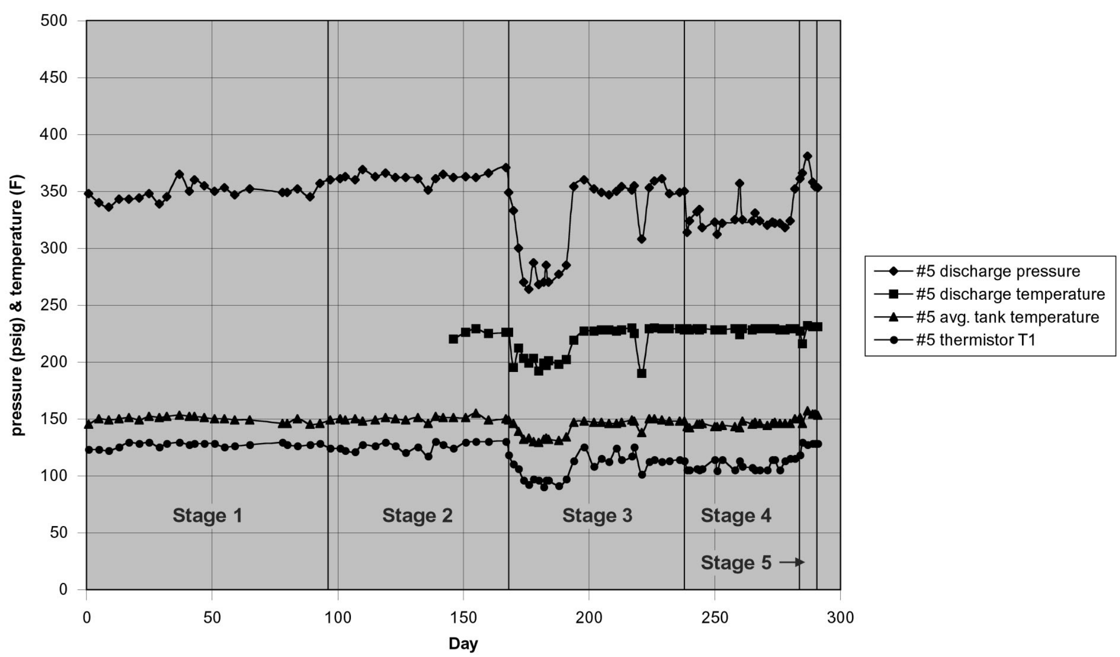


Figure A6. Unit 6 end-of-cycle discharge pressure & temperature, tank temperature, and lower tank thermistor temperature

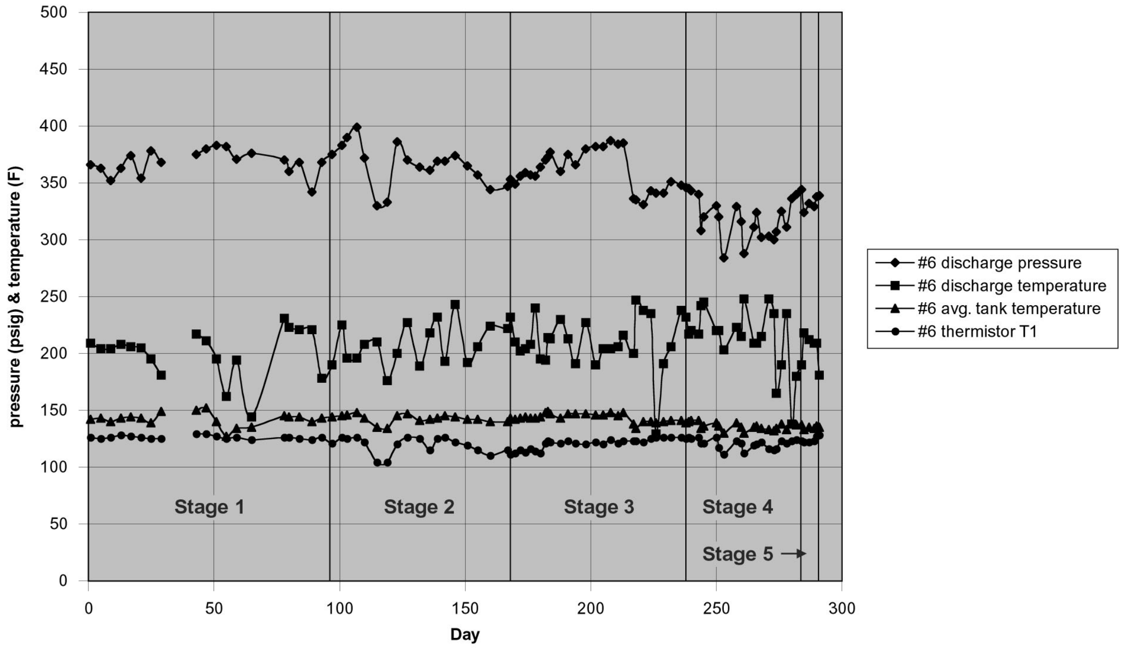


Figure A7. Unit 7 end-of-cycle discharge pressure & temperature, tank temperature, and lower tank thermistor temperature

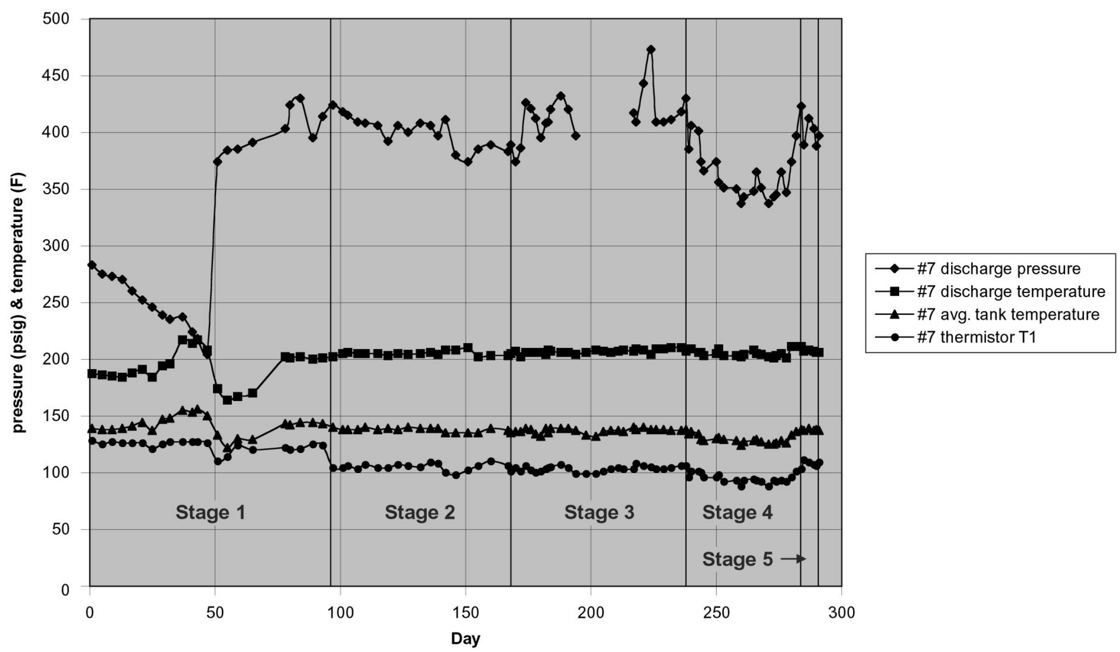


Figure A8. Unit 8 end-of-cycle discharge pressure & temperature, tank temperature, and lower tank thermistor temperature

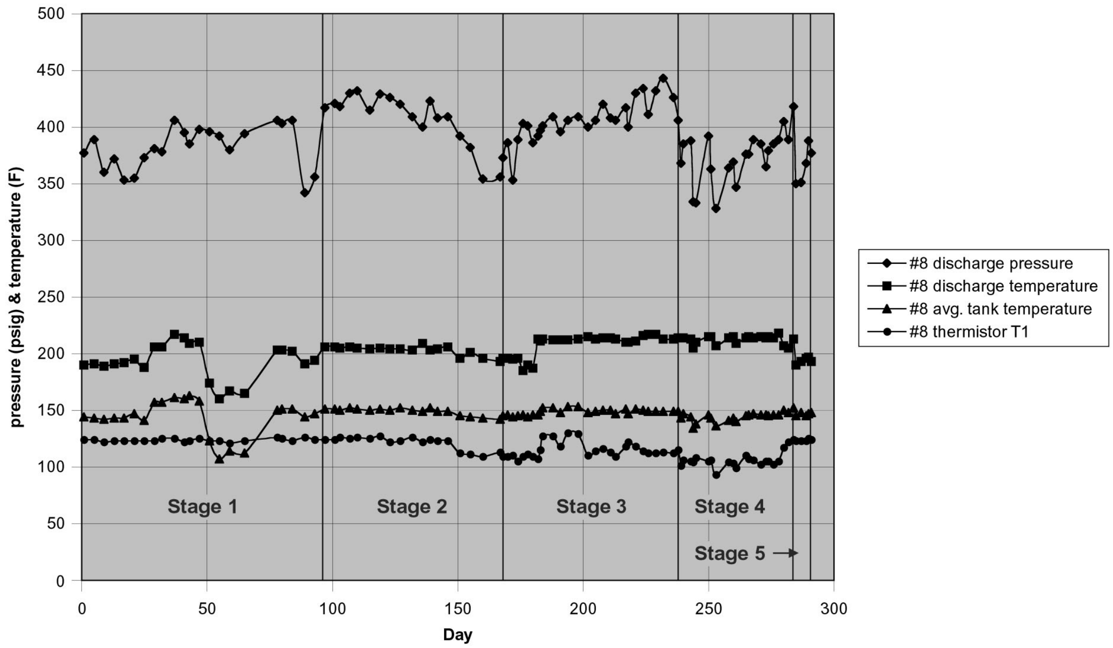


Figure A9. Unit 9 end-of-cycle discharge pressure & temperature, tank temperature, and lower tank thermistor temperature

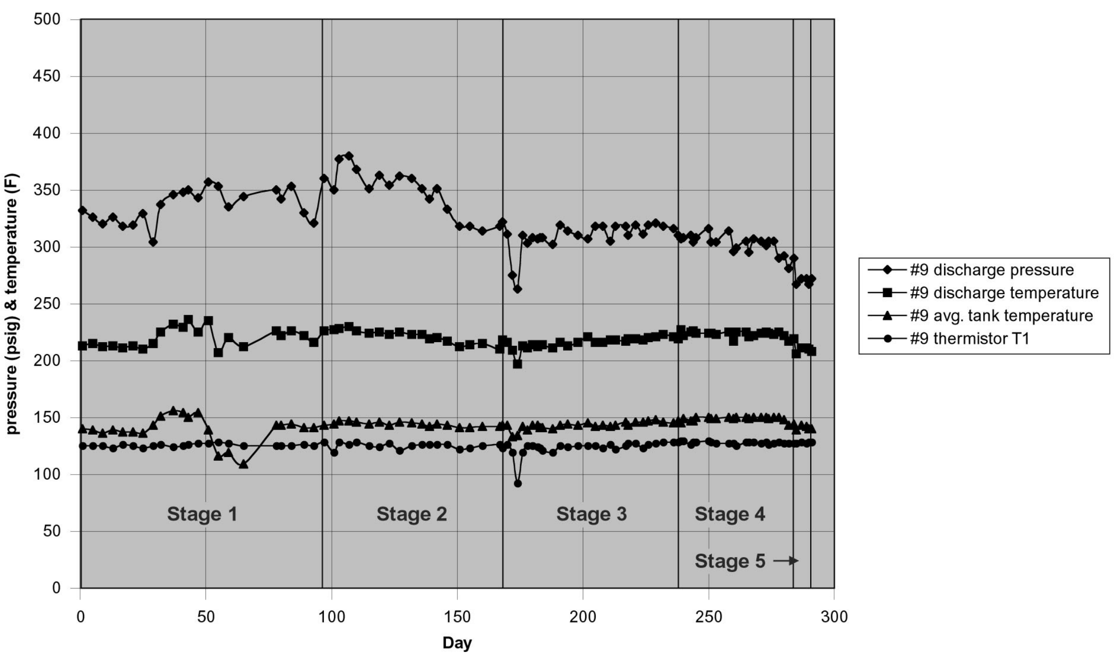
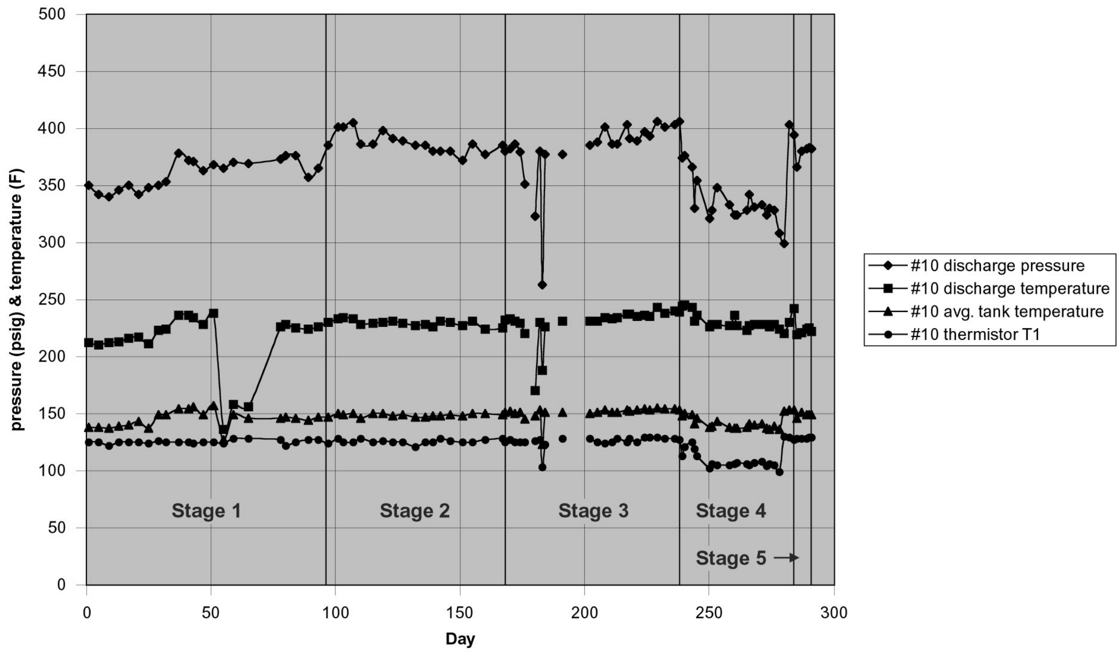


Figure A10. Unit 10 end-of-cycle discharge pressure & temperature, tank temperature, and lower tank thermistor temperature



APPENDIX B: HEAT PUMP WATER HEATER POST–DURABILITY-TEST EXAMINATION REPORT

HEAT PUMP WATER HEATER POST–DURABILITY-TEST EXAMINATION REPORT

Prepared for
Oak Ridge National Laboratory
Date: December 13, 2001

By
John F. Hoyt, Project Engineer
Enviromaster International, LLC

TABLE OF CONTENTS

Abstract

Oak Ridge National Laboratory (ORNL) performed durability tests on ten Heat Pump Water Heaters (HPWH) from December 2000 to October 2001. The purpose of this accelerated test was to simulate ten years of field operation, and provide information on the reliability of components, and robustness of the overall design.

When testing was complete, ORNL removed the compressors from each unit, and sent them to Embraco for evaluation. Nine units were returned to EMI for examination, and one unit remained at ORNL for reference.

EMI received tanks numbered 1, 2, 3, 4, 5, 7, 8, 9, 10 (Tank #6 remained at ORNL). Each tank was dissected as shown in Figs. B.1–B.3 and thoroughly examined for degradation of components and assembly processes.

Fig. B.1

Fig. B.2

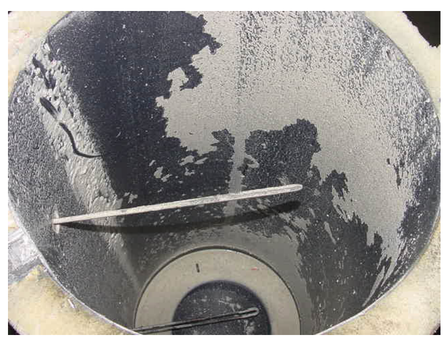


Fig. B.3

A summary of observations follows.

Observations Common To All Units

External Observations

Figures B.4 and B.5 illustrate evidence of moisture (rust) inside J—boxes in front of evaporator, and on the upper thermal cut out switch bracket. This is possibly due to leaking plumbing connections, inadequate condensate drainage, or a combination of the two.

Note: Condensate drain issues have been addressed and resolved. A larger drain stub, and an overflow drain, which channels condensate outside of the unit, is incorporated in the current design.

Fig. B.4

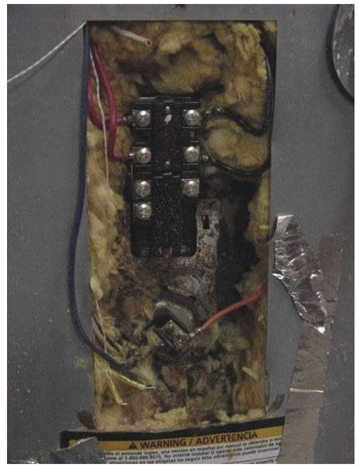


Fig. B.5

Condenser/Top and Bottom Condenser Brazes: Condenser coil tight to outer tank wall. No evidence of movement, shifting or kinking. Brazes in tact.

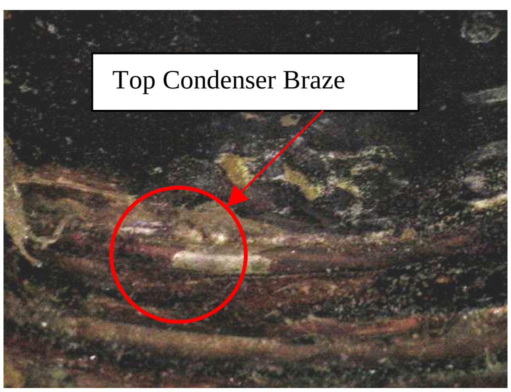


Fig. B.6

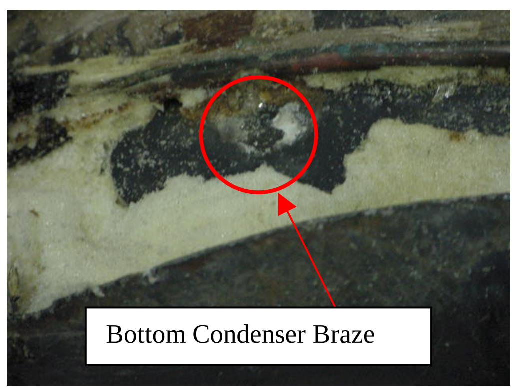


Fig. B.7

External Observations Continued

Foam Insulation: No evidence of degradation. However, some discoloration was observed on foam insulation that was in direct contact with the top of the condenser coil. See Figs. B.8 and B.9. Noted small pockets where foam insulation was not present under the compressor and evaporator.

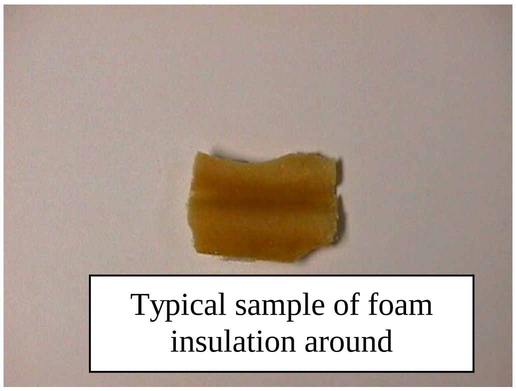


Fig. B.8

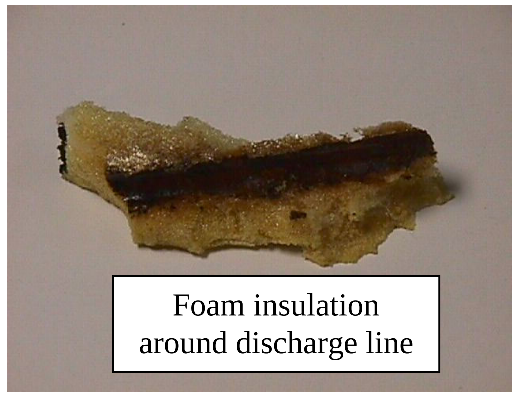


Fig. B.9

Plastic Shrink Wrap: Intact—slightly brittle. Prevented the foam insulation from entering area between condenser coil and tank wall.

Fig. B.10

External Observations Continued

Heat Transfer Compound: Slightly discolored, remained in place and solidified to a wax-like texture.

Fig. B.11

Internal Observations

Sediment / Scale: The thin (approximately 1/8 in.) coating of sediment/scale that is shown in Figs. B.12 and B.13 are typical of what was present on all tanks.

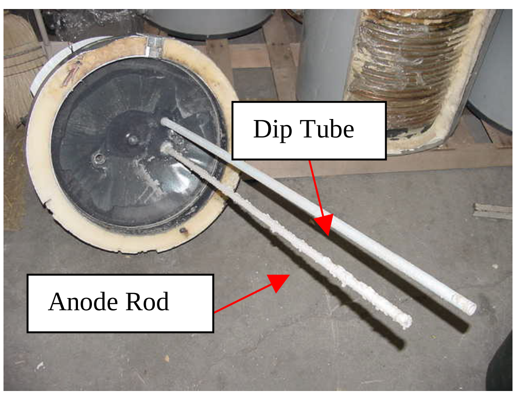


Fig. B.12

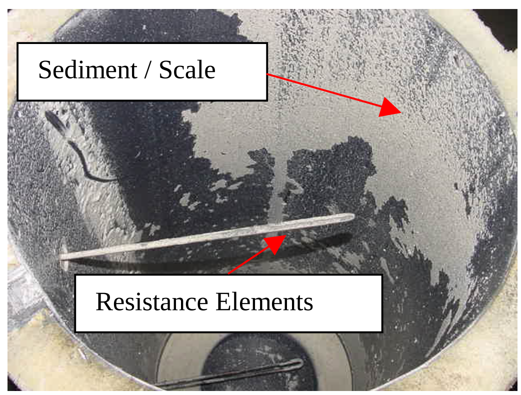


Fig. B.13

Internal Observations Continued

Glass Lining: Slight change in texture of the glass lining inside the tank opposite the braze points. Glass lining did not become separated from the tank wall, and did not show evidence of corrosion at the braze points. See Fig. B.14.

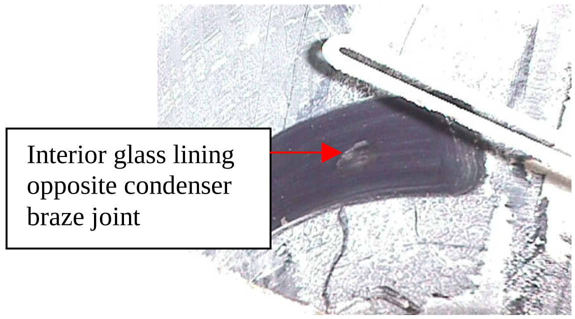


Fig. B.14

Observations *NOT* Common To All Units

External Observations

Electrical Short: Figure B.15 is a photo of the control mounting bracket, and associated wiring on unit #5. A scorched area on the bracket, and partially burned wiring are evidence of an electrical short. (As noted in Sect. 4, this was caused when an inadvertent leak in the test apparatus piping allowed water to drip onto the main power connection of this unit.)

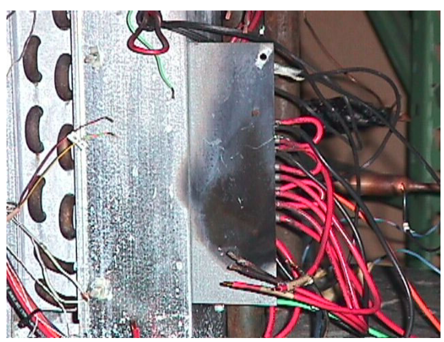


Fig. B.15

Internal Observations

Anode Rods: Figure B.16 depicts the anode on units 3 and 4 that were partially dissolved, but still intact. Figure B.17 illustrates the condition of the anodes on units 2,7,9, and 10 - mostly dissolved, and split along the vertical axis. The anode on units 5 and 8 was completely dissolved, with only the steel core remaining (see Fig. B.18).

Fig. B.16

Fig. B.17

Fig. B.18

Other Notes

- No appreciable scale build–up on electric resistance elements.
- pH levels of 7–8 were found in water samples from units 3 and 7.

Summary

In general, the overall condition of the units appeared to be very good considering they were in operation for a time frame equivalent to 10 years.

Concerns regarding the condenser / tank wall interface, integrity of the braze joints on the condenser, and the condition of the glass lining were investigated, and found to be satisfactory.

However, it must be noted that no feedback from Embraco regarding the condition of the compressors was received at the time of this report.

APPENDIX C: CALIFORNIA ENERGY COMMISSION/ARTHUR D. LITTLE (CEC/ADL) HPWH TESTING PLANS

Table C.1 outlines some of the key similarities and differences between Oak Ridge National Laboratory's (ORNL's) durability test program for the U.S. Department of Energy (DOE) and the laboratory test work that Arthur D. Little (ADL) planned for the California Energy Commission (CEC). Task 2.3 in ADL's plan dealt with performance verification and durability testing. As part of our collaboration in development of this heat pump water heater (HPWH) technology, CEC agreed to accept the DOE/ORNL durability testing for the requirements of their Task 2.3. This allowed ADL and CEC to expend more time and resources to complete development of the first-generation microprocessor control system used on the drop-in HPWH system. It also allowed them to focus some additional effort on a field test program in several California locations.

Table C.1. Comparison of ORNL laboratory testing with CEC Task 2.3 requirements

Description	Task 2.3 requirement	ORNL test	Comments
Test duration	One year	Ten months	ORNL testing simulates about 10 years of normal use, that is, the normal life of a water heater
Number of test units	3	10	Exceeds Task 2.3 requirement
High-mineral-content water testing	High-mineral-content water test on one unit.	None	The field test will include at least one HPWH in a location with hard water.
Pre-test performance testing	Heat-up	Heat-up, energy factor	Exceeds Task 2.3 requirement
Post-test performance testing and evaluation	Heat-up test and destructive testing	Heat-up test and destructive testing	Meets Task 2.3 requirements
Reporting	Laboratory Test Plan; Laboratory Test Report	Test protocol; final report to DOE	ADL Test Plan incorporates ORNL's Test Protocol; ADL final report will cover ORNL testing.

One exception should be noted. The third row of Table C.1 implies that "heat-up" and energy factor (EF) tests would be done before the durability testing. In fact, all the pre-test efforts were devoted to making sure the units would run as designed during the durability test. A number of initial problems with the unit controls (thermistor failures, electronic noise problems) and components had to be corrected. We ran each of the ten test units through a number of heat-up and cycling tests in the course of this shakedown period, but unfortunately all this was done before they were installed in the test facility with full instrumentation hookup. The shakedown work is described in more detail in Sect. 4 of the main body of this report. We had intended to run initial EF tests, but we were unable to do so because we did not get the EF test control software working correctly until after Stage 2 (see Table 2.5) of the durability protocol was under way.