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High Efficiency Water Heating Technology Development – Final Report: Part I, Lab/Field Performance Evaluation and Accelerated Life Testing of a Hybrid Electric Heat Pump Water Heater (HPWH)

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April 1, 2016

OAK RIDGE NATIONAL LABORATORY
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Energy and Transportation Science Division

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EXECUTIVE SUMMARY

In fiscal year 2008 the General Electric Company (GE) and Oak Ridge National Laboratory (ORNL) entered into a collaborative research and development agreement (CRADA NFE-07-0154) to develop and facilitate market introduction of a new generation of high efficiency water heating products. This report provides documentation of the activities conducted under the first phase of the CRADA, namely laboratory and field testing and analyses to evaluate the service lifetime and performance of GE's initial advanced electric water heater called a hybrid electric water heater (HEWH). ORNL conducted accelerated life testing on nineteen HEWH prototypes representing three different stages in the product's development. Ultimately ten of the prototypes successfully completed a program of >2500 water heat cycles with no fatal failures representing at least ten years of service in a residential application.

ORNL also conducted lab tests on four of the earliest stage prototypes to determine their energy efficiency (Energy Factor or EF) and 1st hour rating (FHR). The four units achieved an average EF of 2.03 and FHR of 56.2 gallons, exceeding the US EPA Energy Star performance criteria for electric water heaters (EF≥2.00 and FHR≥50 gallons). Analyses of the prototype's performance conducted by ORNL identified a number of design recommendations to improve efficiency. These recommendations were adopted by GE along with several other design changes of their own. This resulted in an improved EF of the initial production units of ~2.4 (20% improvement). The measured FHR also increased to ~62 gallons.

GE introduced the HEWH product to the US market in late 2009 under the brand name GeoSpring™. Initially the product was manufactured in China, but in 2012 production was moved to GE's Appliance Park facilities in Louisville, KY. This resulted in adding 1300 manufacturing jobs to the local economy. Assuming an EF of 2.4 and that the HEWH product would achieve 10% of the total US electric WH market from 2010-2020, it is projected that total cumulative national energy savings would reach ~0.9 Quads and consumer electricity costs would be reduced by > $8 billion (based on 2006 national electricity costs).

A production model of the GeoSpring was lab tested to determine its performance using a low global warming potential (GWP) refrigerant, R-1234yf, in lieu of R-134a. Results of drop-in testing indicated that the EF with R-1234yf was about 6% lower than the rated EF with R-134a. There was no change in the FHR with R-1234yf. Repeat tests with a revised expansion valve improved the EF with R-1234yf to 3% below the R-134a EF. Based on these results it appears that an optimized R-1234yf design may very closely match the Energy Factor of the current R-134a based product without compromising FHR.

A second production unit was field tested in a research house in Oak Ridge, TN. The GeoSpring unit was located inside the conditioned space of the house to determine the net impact of its operation on the house space conditioning system (air-source heat pump, ASHP). Results indicated that the ASHP energy use increased by ~0.4-0.5 kWh/d on average over the test year due to having the GeoSpring operating inside the conditioned space. However, this increased energy use was quite small relative to the reduction in water heating energy use compared to that of a standard electric storage WH (5.90 kWh/d for this field test).

INTRODUCTION

DOE has supported efforts for many years with the objective of getting a water heater that uses heat pump technology (aka a heat pump water heater or HPWH) successfully on the residential equipment market. The most recent previous effort (1999-2002) produced a product that performed very well in ORNL-led accelerated durability and field tests. The commercial partner for this effort, Enviromaster International (EMI), introduced the product to the market under the trade name “Watter$aver” in 2002 but ceased production in 2005 due to low sales. A combination of high sales price and lack of any significant infrastructure for “service after the sale” were the principal reasons for the failure of this effort. What was needed for market success was a commercial partner with the manufacturing and market distribution capability necessary to allow economies of scale to lead to a viable unit price together with a strong customer service infrastructure. General Electric certainly meets these requirements, and knowing of ORNL’s expertise in this area, approached ORNL with the proposal to partner in a CRADA to produce a high efficiency electric water heater. A CRADA with GE was initiated early in fiscal year 2008. GE initially named its product the Hybrid Electric Water Heater (HEWH). Figure 1 provides a photo of an early prototype of the GE HEWH product (a) and a cutaway view of a HPWH unit of similar design.

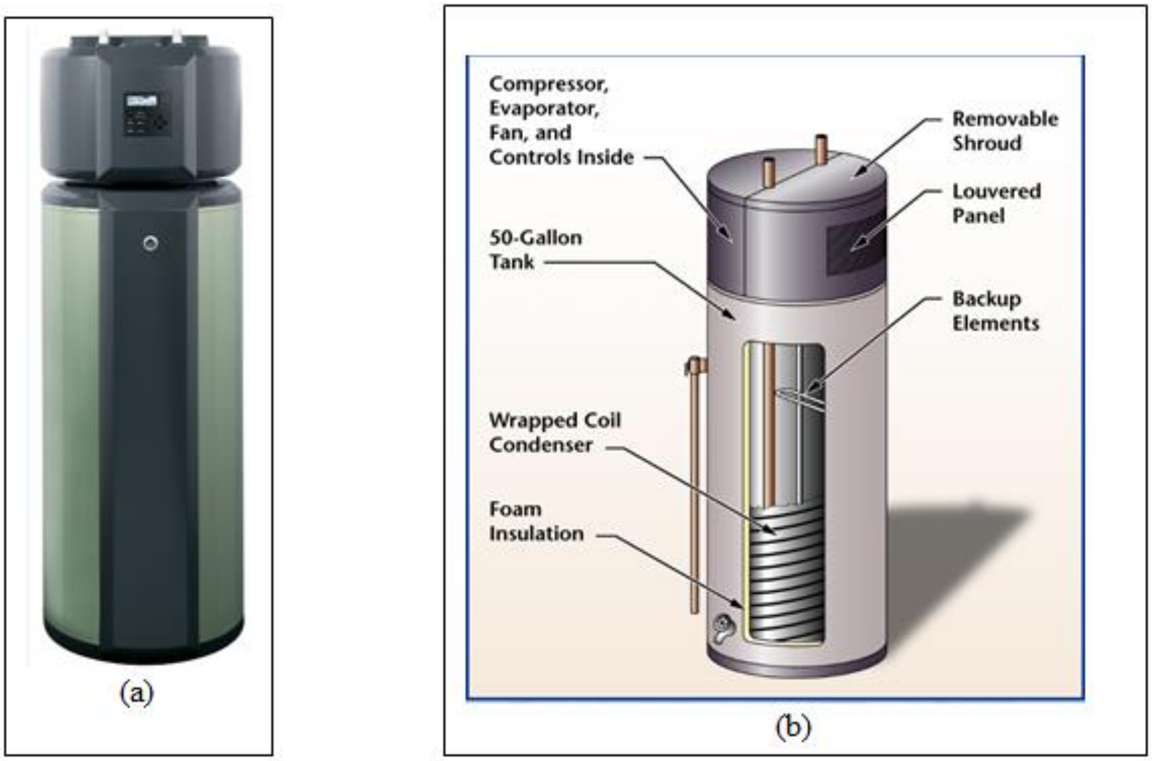

Figure 1 (a) early DG prototype of GE HEWH; (b) cutaway view of similar integral type HPWH design.

EQUIPMENT TYPES & SHIPMENTS

Note – much of the material in this section is summarized from the 2011 DOE Water Heating RD&D Roadmap (Goetzler, et al, 2011). The U.S. water heating market offers residential and commercial consumers several distinct technology options (Table 1).

Table 1. Water heating technology options

Fuel	Technology		Description
Fuel-fired; primarily natural gas fired	Storage	Standard	Heat loss occurs primarily through flue gases and stand-by.
		HE Standard	Better insulation, heat traps and burners, and, in some cases, a power vent or flue damper.
		Condensing	Captures the latent heat of combustion gases before they exit the tank.
	Tankless	Standard	Heats water in a continuous flow process, eliminating standby losses.
		Condensing	Eliminates stand-by losses, increases efficiency with secondary heat exchanger.
	Hybrid		Gas tankless water heater with small (≤ 20 gallons) storage tank. Stored hot water reduces wait time that may occur with a tankless heater as it fires up, yet preserves most of the efficiency benefit from low stand-by losses.
Electric	Standard		Heats water through two electric elements within the storage tank.
	Tankless		Heats water in a continuous flow process using electric heating elements.
	HE Standard		A more efficient version of the Standard model due to insulation and thermal improvements.
	Heat Pump Water Heater		Electric heat pump water heaters extract low-grade heat from the air and transfer this heat to water. Heat pump water heaters can be integrated models that fully replace standard electric water heaters or can be add-on units added to existing electric or gas storage water heaters.
Any	Solar with Back-up		Solar water heaters use captured solar thermal energy to heat water in a storage tank. SWH systems may be direct (uses water from the main) or indirect (uses a working fluid), active (electric pump) or passive. SWH systems can be backed up with a grid-tied gas or electric water heater.
	Drain Water Heat Recovery		Recovers heat from drain water and transfers to incoming cold water stream. Efficiency requirements do not exist in the U.S. but vary from 30-42% elsewhere.

The most prevalent options are gas-fired and electric resistance models with attached storage tanks. In 2010 tank-type models represented about 95% of shipments in the residential market (Table 2). Tankless models are relatively new entrants to the market but are gaining market acceptance and currently represent roughly 5% of total shipments, the vast majority of which are EnergyStar qualified gas tankless models. All other water heater types—including heat pump water heaters (HPWH), an alternative to electric resistance models; solar water heaters, which use thermal energy from the sun to heat water; and drain water heat recovery, which captures heat from water as it flows down the drain—capture a small fraction of the market.[1] HPWHs, a new entrant to the water heating market are still a very small portion of the overall sales of electric storage units but appear to be taking off (heat pumps represented approximately 1.6% of the market in 2010 versus 0.4% of the market in 2009).

Table 2. Breakdown of residential water heating system shipments

Residential	2009	2010
Gas Storage (Total)	**3,760,657**	**3,918,510**
Not E [2]*	*3,110,419*	*3,463,780*
E-qualifying [3]*	*650,238*	*454,730*
Gas Tankless (Total)[4]	**380,000**	**399,000[5]**
*Not E**	*46,987*	*14,974*
E-qualifying*	*333,013*	*384,026*
Electric Storage (Total)	**3,751,994**	**3,736,597**
Not E [6]*	*3,737,260*	*3,677,472*
E-qualifying (heat pump)[7]*	*14,734*	*59,125*
Solar [8]	**31,647**	**33,462**
*Not E**	**24,751**	**23,472**
E-qualifying*	**6,896**	**9,990**
TOTAL	**7,924,298**	**8,087,569**
*Total Not E**	*6,919,417*	*7,179,698*
Total E qualifying*	*1,004,881*	*907,871*

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[1] ENERGY STAR Water Heater Market Profile, 2010

[2] Air-Conditioning, Heating, and Refrigeration Institute (AHRI), AHRI December 2010 U.S. Heating and Cooling Equipment Shipment Data.

[3] ENERGY STAR Product Type Market Share, Preliminary data for 2009 and 2010, U.S. EPA

[4] ENERGY STAR Product Type Market Share, Preliminary data for 2009 and 2010, U.S. EPA

[5] Data not available through ENERGY STAR or AHRI. Based on estimates provided by Mike Parker, A.O. Smith, Keynote presentation at ACEEE Hot Water Forum, May 10, 2011.

[6] Air-Conditioning, Heating, and Refrigeration Institute (AHRI), AHRI December 2010 U.S. Heating and Cooling Equipment Shipment Data.

[7] ENERGY STAR Product Type Market Share, Preliminary data for 2009 and 2010, U.S. EPA

[8] Solar Energy Industries Association (SEIA), U.S. Solar Market Insight Year in Review 2010, March 2011.

Table 3. Breakdown of commercial water heating system shipments

Commercial[9]	2009	2010
Gas, storage	75,487	78,614
Electric, storage	55,625	58,349
Total	131,112	136,963

Residential models dominate annual shipments of water heating equipment (Tables 2 and 3). The residential installed base of water heaters is approximately 100 million units. With approximately 7 to 8% of these units requiring replacement in a given year, the vast majority of shipments go toward replacing old units. The new construction market represents a much smaller portion of annual sales, and the economic downturn that began in 2008 slowed new construction sales.[10]

An emerging trend in the residential market is an increase in sales of higher efficiency equipment. In 2009, one million units shipped (13% of the market) were ENERGY STAR-qualified models of all types compared with just 625,000 high efficiency units (comparable to ENERGY STAR's 2009 qualifying levels) shipped in 2006, even more remarkable considering that the overall market shrank during that period.[11] The ENERGY STAR program for residential water heaters launched in 2009 and well over a thousand qualifying models (gas storage, gas tankless, heat pump water heater, solar) have been registered through the program.

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[9] Air-Conditioning, Heating, and Refrigeration Institute (AHRI), AHRI December 2010 U.S. Heating and Cooling Equipment Shipment Data.

[10] ENERGY STAR Water Heater Market Profile, 2010

[11] Based on ENERGY STAR-qualifying equivalent models, ENERGY STAR Water Heater Market Profile, 2010

HYBRID ELECTRIC WATER HEATER

A. Reliability testing

Initial prototypes (Design Guidance or DG). GE provided ten early prototype units to ORNL for testing in June 2008 – units DG-18 through DG-27. These were samples assembled on a prototype of the production line to be used for final product manufacturing. The initial test plan called first for conducting standard 1st hour rating and 24-h simulated use tests of the type used to establish energy factor (EF) ratings (US Code of Federal Regulations 2010) on each unit in “as received” condition. Once these were completed, then a reliability test run would begin.

After delivery GE determined that a number of modifications had to be made to the design of these prototypes before testing could begin. One unit, DG-24, failed the initial factory test protocol due to an irreparable refrigerant leak in the condenser coil shortly after delivery. It was returned to GE and a substitute, DG-31, was sent to ORNL to complete the reliability test line up. The major design change required was to modify the design of the dip tube so that the unit controller could reliably detect a hot water draw after 5-6 gallons were removed from the tank and replaced with line (cold) water. As received these prototypes were requiring over 20 gallon hot water draws before the controls would activate the heat pump section to heat the tank water. In addition, several control program modifications had to be implemented before the units would operate to the GE team’s satisfaction. The impact of all these redesign efforts was that the test plan had to be revised so that only a subset (four) of the units underwent EF and 1st hour rating tests and the start of the reliability test run was delayed until August 21. Figure 2 is an exterior view of the ORNL reliability/durability test stand and Figure 3 shows the ten original DG units installed in the ambient control chamber of the test stand.

Figure 2. ORNL water heater reliability/durability test stand.

Figure 3. Initial DG prototypes in ambient control chamber of water heater durability/reliability test stand.

The initial plan for the reliability testing is given in Table 4. A total of 2500 cycles was to be accumulated at various different entering cold water, ambient air, and humidity conditions. The initial protocol was weighted toward large water draws (35 gal). In May of 2009 the test protocol was changed (Table 5) to put greater emphasis on smaller water draws (15 gal) and was used for the balance of the reliability testing.

Table 4. Reliability test protocol August 2008 through May 2009

Run #	Air temp, °F	Air %RH	Inlet water temp, °F	HPWH mode[a]	Water draw, gal	# of cycles
Run 1	54	20	45/50	Hybrid	35	140
Run 2	54	20	45/50	Hybrid	15	60
Remove/replace heat pump covers; remove, clean, replace air filter on half of units						
Run 3	110	41	58	Hybrid	35	175
Run 4	110	41	58	Hybrid	15[b]	75
Remove/replace heat pump covers; remove, clean, replace air filter on half of units						
Run 5	75/80	80	45/50	Hybrid	35	35
Run 6	75/80	80	45/50	Hybrid	15	15
Run heat up test before proceeding to Run 7						
Remove/replace heat pump covers; remove, clean, replace air filter on half of units						
Run 7	75/80	80	58	Hybrid	35	315
Run 8	75/80	80	58	Hybrid	15[b]	135
Remove/replace heat pump covers; remove, clean, replace air filter on half of units						
Run 9	54	20	45/50	Std Elec	35	50
Run heat up test before proceeding to Run 1						
Total cycles per each time through test protocol						1000

[a] All tests run with tank water set point of 140 °F.

[b] Water draw taken through drain valve for one cycle of Runs 4 and 8.

Table 5. Reliability test protocol May 2009 through June 2010

Run #	Air temp, °F	Air %RH	Inlet water temp, °F	HPWH mode[a]	Water draw, gal	# of cycles
Run 1	50	20	45/50	Hybrid	35	20
Run 2	50	20	45/50	Hybrid	15	180
Remove/replace heat pump covers; remove, clean, replace air filter on half of units						
Run 3	110	41	58	Hybrid	35	25
Run 4	110	41	58	Hybrid	15[b]	225
Remove/replace heat pump covers; remove, clean, replace air filter on half of units						
Run 5	75/80	80	45/50	Hybrid	35	5
Run 6	75/80	80	45/50	Hybrid	15	45
Run heat up test before proceeding to Run 7						
Remove/replace heat pump covers; remove, clean, replace air filter on half of units						
Run 7	75/80	80	58	Hybrid	35	45
Run 8	75/80	80	58	Hybrid	15[b]	405
Remove/replace heat pump covers; remove, clean, replace air filter on half of units						
Run 9	50	20	45/50	Std Elec	35	50
Run heat up test before proceeding to Run 1						
Total cycles per each time through test protocol						1000

[a] All tests run with tank water set point of 140 °F.

[b] Water draw taken through drain valve for one cycle of Runs 4 and 8.

As noted in Tables 4 and 5, each trial through the test protocol would accumulate 1000 cycles (water draw followed by tank heat up and heat pump or resistance element shutdown). The ten test units were required to complete at least 2500 cycles (2.5 cycles through the test protocol) with no fatal failures to meet GE's reliability goals of ~97% and ~74% with a 50% confidence level after 1 year and 10 years, respectively, of normal operation in a residence. See Appendix A for details on the reliability test duration estimate.

ORNL began life or reliability testing on August 21, 2008 as noted earlier with the ten DG development prototype HEWH units. By early March 2009, about 1290 total cycles had been accumulated passing the 50% completion point. Only two major operational incidents occurred during this period. One unit developed a refrigerant leak early in the testing. A second unit experienced a failure of its lower electric (backup heat) element due to a poor wire connection. In both cases the problems were fixed and no further incidents occurred. At that time, the GE team indicated that they wanted to replace some of the DG units with 2nd generation design confirmation (DC) prototype units.

Table 6 summarizes the cycling history of the DG units through June 2010 (conclusion of reliability test run) and Table 7 summarizes failure incidents and other problems reported through the same time period. Seven of the DG test units were replaced in March 2009 with second generation design confirmation (DC) prototypes and completed only about half of the number of cycles originally planned. One, DG 27, completed 2469 cycles (almost the required 2500) before being replaced with an even later generation production prototype (PP) and the other two, DG-22 and DG-23, completed about 4700 cycles (with no failure incidents).

Table 6. Reliability test cycles for DG test units – August 2008 through June 2010

Run #	DG-18[a]	DG-19[a]	DG-20[a]	DG-21[a]	DG-22	DG-23	DG-25[a]	DG-26[a]	DG-27[b]	DG-31[a]
Run 1	140	140	140	140	320	320	139	140	300	140
Run 2	60	60	60	60	506	503	60	60	238	60
Run 3	175	175	175	175	274	277	175	175	209	175
Run 4	75	75	120	75	525	528	75	75	300	75
Run 5	35	35	35	35	51	52	35	35	40	35
Run 6	16	16	16	15	215	227	16	17	77	16
Run 7	448	452	478	473	645	637	461	480	530	473
Run 8	270	270	270	270	1635	1703	270	270	675	270
Run 9	50	47	50	48	224	251	47	42	96	50
Total	**1269**	**1270**	**1344**	**1291**	**4640**	**4733**	**1278**	**1294**	**2469**	**1294**

[a] Replaced with DC test unit in March 2009 & returned to GE for post mortem analysis.
[b] Replaced with PP test unit in October 2009 & returned to GE for post mortem analysis.

Table 7. DG unit failures/incidents and test stand issues; Aug 2008 through June 2010

DG units	Date(s)	Incident description
DG-18		
Failures	13-Jun-08	Unit developed a refrigerant leak at an evaporator return bend while being instrumented for 24-h use (energy factor, EF) test. There was a significant dimple at the return bend.
DG-19		
Failures		None.
Other incidents	13-Jun-08	Foam (and fiberglas insulation around element covers) on this unit was soaked on initial water fill at ORNL due to loose anode rod fitting.
DG-20		
Failures	17-Feb-09	Noted that unit was calling for lower element (LE) but LE was not operating. Found burned LE electrical fitting upon investigation. Replaced with new LE on 2/19 and replaced power wire with new one that was run up outside of tank as temporary fix. Allowed unit to run extra Run4 cycles to make up for days missed with LE failure.
Other incidents	Oct 08 & Feb 09	Noted very low flow (0.2-0.3 gpm) during drain valve draws (Run4 and Run8). Opened drain valve and bled a little water plus ran wire into valve. Then drain flowed freely. Problem most likely due to air lock in drain valve water line on ORNL test stand – see comments for DG-26.
DG-21		
Failures		None
Other incidents	Aug 08 – Mar 09	Unit took longer than the others to reach set point during Runs 3-6. Onboard control system reported higher superheat than the other units (>40F during Runs 3 and 4).
DG-22		
Failures		None

DG units	Date(s)	Incident description
Other incidents	Feb. 09	Trial 1, Run 4 drain valve draw; Flow gradually decreased throughout draw from 1.3 gpm to 0.8 gpm.
	Jun. 09	Trial2, Run3; first cycle this run was with LE – onboard control system reported ambient air temp >120F
	Oct. 09	Trial3, Run 3; two cycles in this run were with LE – controller again noted high ambient temp. Reset chamber set point from 115F to 110F and this behavior ceased.
DG-23		
Failures		None
Other incidents	May, 2008	Foam on this unit was soaked with water before delivery to ORNL. Used for corrosion study at GE after ORNL testing.
	Oct. 09	Trial3, Run3; five cycles this run ended with LE operation due to low superheat indication by controller. Reset chamber set point from 115F to 110F and this behavior ceased.
	Oct. 09	Trial3, Run4; three cycles this run ended with LE operation - low superheat indication most likely.
	4-Mar-10	After Trial 4, Run 2, filter became stuck and could not be removed for cleaning - possibly due to warping from Sept 2009 hot soak incident - will try again at next clean trial after Run 4
	25-Mar-10	After Trial 4, Run 4, we were able to remove and clean filter again. There was some warpage of the filter frame most likely associated with the Labor Day 2009 hot soak event.
DG-24		
Failures	13-Jun-08	Unit failed factory test at ORNL. Was determined to have a refrigerant leak in condenser wrap (not easily repairable since it was underneath the foam insulation). Unit returned to GE and replaced with DG-31
DG-25		
Failures		None
DG-26		
Failures		None
Other incidents	Oct 08 & Feb 09	No flow initially during Run4 and Run8 drain valve draws. After cracking open the line and bleeding a little water a couple of times drain flowed freely. The problem was most likely due to an air lock in the water line installed on ORNL test loop to connect the tank drain to the main loop (also most probably cause for DG-20 drain valve flow problems). The water coming from tank bottom was quite rusty looking.
DG-27		
Failures		None
Other incidents	Dec. 08	Early in Trial 1, Run1 all units experienced 1 or 2 cycles where the LE activated before heat pump finished heat up. Onboard controller giving low ambient temp indication. After chamber controls were set to raise ambient temp to 54F this behavior generally ceased for all units except DG-27 which picked up at least four more such events. So we made it go 4 cycles longer than the others.
DG-31		
Failures		None

DG units	Date(s)	Incident description
Other incidents	Feb. 09	Almost all Run3 cycles for DG-31 in Trial 1 ended with call to LE; most likely cause was low superheat indication by controller.
DG, general comments		
	Nov. 08	All Trial 1 cycles of Run 9 hampered by problem with ORNL test loop control system -- caused random 35 gallon water draws to occur in middle of some (but not all) cycles. Problem was corrected prior to heat up tests after Run9.
	19-May-09	Test protocol changed to revise apportionment of cycles among the nine Runs - increased number of 15-gallon draw cycles and reduced number of 35-gallon draw cycles.

2nd generation prototypes (Design Confirmation or DC). Ten of the revised 2nd generation prototype HEWHs were received on February 23, 2009 – DC units 75-84. Units 75-80 and 82 were installed on the ORNL reliability test stand. DC-81 was sent to the Electric Power Research Institute's (EPRI) facility in Knoxville, TN and DC's 83 and 84 were installed in test homes in Knoxville. GE personnel visited ORNL in early March to install production control software on the new units and to correct a wiring problem on the control boards that caused several of the units to fail the factory start up test. They also picked up the seven DG units removed from the durability test stand to return them to GE for tear down evaluation. Durability testing restarted with the new test unit lineup in early April 2009.

Table 8 summarizes the cycling history of the DC test units from March 2009 through June 2010 and Table 9 summarizes failure incidents and other problems reported during the reliability test run through the same time. Six of the DC prototypes completed about 3400 cycles on average. The 7th, DC 77, was pulled from the test stand after six months and replaced with a production prototype (PP) unit.

Table 8. Reliability test cycles for DC test units – April 2009 through June 2010

Run #	DC-75	DC-76	DC-77[a]	DC-78	DC-79	DC-80	DC-82
Run 1	191	192	140	191	185	190	186
Run 2	460	442	173	441	442	444	441
Run 3	94	93	41	92	121	113	118
Run 4	685	680	225	680	689	680	695
Run 5	23	23	7	20	24	24	17
Run 6	198	217	60	200	202	216	218
Run 7	176	177	45	142	161	156	152
Run 8	1395	1370	405	1496	1450	1366	1466
Run 9	150	150	50	169	192	190	189
Total	**3373**	**3344**	**1146**	**3428**	**3466**	**3388**	**3484**

[a] Replaced with PP test unit in October 2009 & returned to GE for post mortem analysis.

Table 9. DC unit failures/incidents and test stand issues; Apr 2009 through June 2010

DC units	Date(s)	Incident description
DC-75		
Failures		None
Other incidents	Apr. 2009	Trial2, Run1, one cycle this run was with lower element (LE) – control thermistor readings not within bounds
DC-76		
Failures	15-Oct-09	Trial3, Run3; unit failed early in cycle 14 during upper element (UE) operation to heat top of tank. Found burned wire and connector lug leading to common terminal on UE relay. Control board replaced October 19.
	10-Jun-10	Trial 4, Run 9; On June 11, we noted that the 4 blue led lights on the control panel were all lit. The panel display indicated "Water Heat System Failure" - pushing "Enter" on the panel display brought up screen displaying "WHF Code: F10" and asking for service code. Unit was still operating in electric heat mode (mode required for Run 9). Examination of the data from the control system showed that the controls had only called for LE operation for the past several cycles. Examination of the data logger plot showed that the problem first appeared in cycle 32. Cycles 32-37 were LE only. Cycle 38 was normal (UE heats top of tank then then LE finishes heat up). Remaining cycles in Run9 were LE only. Checked UE itself & found resistance at 13.1 ohms, nominal. Checked control board & found one connector on UE relay with evidence of scorching. New control board needed to make fully operational again. Unable to run final heat up test.
DC-77		
Failures	Apr. 2009	Unit failed early in Trial2, Run1, cycle 58 during UE operation to heat top of tank. Found burned wire connection and deformed double line break relay on control board April 17, 2009. Control board replaced on April 22 & unit restarted.
DC-78		
Failures	Mar. 2009	After installation on test loop, power supply board failed. Board replaced prior to starting reliability test run
	May, 2009	During 1st Trial2 filter remove/clean exercise noted that the filter was difficult to remove; had to completely remove front half of shroud to dislodge. Found filter frame cracked at one top corner. Able to replace OK - filter media section not compromised.
	5-Nov-09	Trial3, Run5; unit failed in cycle 2 during UE operation to heat top of tank. Found burned wire coming from onboard current transformer (CT) at the double line break relay connection. Board replaced Nov 9 and Run5 completed. Missed Run6 while down for repair. Added 33 cycles run during Jan 2010 heat up tests @ 67.5 F, 50% RH, and 58F entering water conditions. Finished Run6 in June, 2010 just before start of final Heat up tests.

DC units	**Date(s)**	**Incident description**
	28-Mar-10	Trial4, Run6; unit failed at start of cycle 19 during UE operation to heat top of tank. Found burned wire connection and LE relay at the common terminal. Board replaced Mar 29 and Run6 completed.
DC-79		
Failures		None
DC-80		
Failures	Mar. 2009	Top plate was cracked at main condensate drain outlet port when received. Repaired crack with RTV-type sealant and connected drain hose to overflow drain for reliability tests.
Other incidents	Apr. 2009	Trial2, Run1, five cycles this run was with lower LE – control thermistor readings not within bounds.
DC-82		
Failures		None
Other incidents	Apr. 2009	Trial2, Run1, one cycle this run was with LE – control thermistor readings not within bounds.
	May, 2009	Trial2, Run2; initial cycle this run was with LE – control thermistor readings not within bounds; on cycle 2 unit did not start after 15 gallon water draw leading to a 2nd 15 gallon draw after which unit started but then switched to UE and finished with compressor.
	Nov. 2009	ORNL control program developed problems 11/5/2009 causing DC-82 to be temporarily off line. Problem resolved 11/17/09, unit back on line but missed Runs 5 & 6 of Trial3 - will try to back fill these later. Completed these Runs just before and after final heat up test in June, 2010.
DC, general comments		
Failures	Mar. 2009	Four DC units failed initial factory test upon delivery - 76 & 80 (on RLT stand), 81 (sent to EPRI), and 84 (sent to Campbell Creek test house). Compressor "fail" indicated due to low current sensor reading from control board's current transformer (CT), but clamp on ammeter indicated sufficient current draw. Latest version of unit control software (***9030911***) flashed to all DC control boards and the 240V power wire was "double wrapped" through the CT to boost signal on all DCs except 75 and 79 (not enough of the 240V wire lead available to do double wrap). All ten DCs then passed the factory test.
Other incidents	19-May-09	New unit control software version (***0904315***) applied.
	19-May-09	Test protocol changed - increased 15-gallon draw cycles and reduced 35-gallon draw cycles.
	8-Jun-09	New upper and lower elements installed in DC-76, DC-77, and DC-80 prior to start of Run5, Trial 2
	3-Aug-09	New software version (***09072903***) flashed to DC units near end of Trial 2, Run8.

DC units	Date(s)	Incident description
	6-Sep-09	During Run 1 of Trial 3, all units were exposed to 24+ hr "hot soak" at air temperatures up to 192 F caused by failure of ORNL test chamber climate control system. All DC units had severely warped shrouds as a result and DC-80 and DC-76 were disabled - the problems were ultimately traced to electrical connections that had worked loose during the hot soak. After these connections were restored both became operational. GE provided new shrouds to replace the heat warped shrouds. We ceased the remove/replace shrouds and clean filters steps for the DC units after this incident because it was no longer representative of these activities for typical units. These activities continued for the DG units and PP units, however.

Over the Labor Day 2009 holiday all ten test units were subject to an unplanned "hot soak" when the test chamber air temperature soared to over 190 °F for about 24 hours. This was caused when the chamber safety systems shut off the cooling system due to low refrigerant charge. The high temperature safety cut off was unfortunately set too high (~200 °F) to prevent the high temperature incident. Fortunately, however, all ten test units survived the "hot soak" without damage to their control or heat pump hardware systems and remain operational – albeit the cover shrouds suffered some deformation. The chamber air temperature safety switch was reset to a more reasonable level and the incident did not reoccur.

GE personnel indicated in early October that they wanted to replace two of the test units (one DG and one DC) with production pilot prototype (PP) units. These were very similar to the DC units but were built on the production line used for initial run of "for sale to the public" units later in 2009.

Production pilot prototype (PP) units. The new PP units were delivered to ORNL on October 2, 2009 and placed on the test stand the same day, replacing DG-27 and DC-77. The two older test units were returned to GE for tear down examination. The reliability (or accelerated life) test run restarted in late October and was completed in June 2010. A total of approximately 2600 cycles for the PP units (no failure incidents) were completed. Thus a total of 10 test units (2 DG type, 6 DC type, and 2 PP type) completed more than the minimum 2500 cycles called for in the reliability test. Final heat up tests were performed (details in the next section of the report), revealing no indication of any performance degradation for any of the test units during the course of the durability test period. All ten reliability test units were returned to GE for tear down analyses.

Table 10 summarizes the cycling history of the PP test units from October 2009 through June 2010. Both PP units completed 2600+ cycles on the reliability test stand with no failure incidents that could be attributed to the unit components or design. The primary condensate drain for PP-36 was discovered to be broken in April 2010 but that most likely occurred due to ORNL personnel inadvertently hitting it during a heat up test. The drain was sealed and the overflow drain used for the remainder of the reliability testing with no incident. Table 11 summarizes this issue and other problems reported during the reliability test run through the same time.

Table 10. Reliability test cycles for PP test units – October 2009 through June 2010

Run #	PP-23	PP-36
Run 1	51	52
Run 2	388	390
Run 3	53	53
Run 4	569	573
Run 5	16	16
Run 6	145	149
Run 7	110	94
Run 8	1189	1212
Run 9	100	100
Total	**2621**	**2639**

Table 11. PP unit failures/incidents and test stand issues; Oct 2009 through June 2010

DC units	Date(s)	Incident description
PP-23		
Failures		None
PP-36		
Failures		None
Other incidents	Apr. 2010	During the Heat Up test period after Run 6 the primary condensate drain fitting on PP-36 was somehow broken. We are not sure if it broke on its own or if we did it and just did not notice immediately (more than likely the latter was the case). We sealed off the primary drain and used the secondary drain fitting for the remainder of the reliability test period.
	8-Apr-10	During cycle 27 of Trial 4, Run 7 the electric circuit breaker tripped while the unit was heating the tank with LE. The test control system kept making water draws so the tank temperature dropped to ~65F. Breaker was reset early on 4/9/2010 and unit started normally with UE heating top of tank, but then the compressor came on to heat up rest of way (LE did not activate despite unit being in Hybrid mode and the tank being full of cold water). Similar incidents happened during cycle 29 later on 4/9 and cycle 30 on 4/12. We examined the unit's control board on 4/13 during the shroud removal exercise and found nothing to indicate any excessive current (overheating/scorching, etc.) at any of the 240V relay connections or any other problems. We similarly found no such indications at the LE connections itself. We also checked the resistance of the LE and found it to be ~13.5 ohms which was consistent with that of several 4500W elements we had on hand in the lab and a bit less than that of a 5500 W element. On 19-Apr we hooked the unit up to a different breaker and did a 35-gal draw. The unit ran normally, cycling off on meeting the tank thermostat setting of 140F and not on breaker trip, so it is likely that the original breaker used for PP-36 has weakened somewhat over the 1.5 year reliability test run.

Ultimately ten of the prototypes (2 DG units, 6 DC units, and 2 PP units) successfully completed more than 2500 water heat cycles with no fatal failures. The results exceeded the ~74% reliability criteria established by GE for ten years of HEWH service in a residential application (see Appendix A).

Production products. Initial production of a new HEWH product, based essentially on the PP prototypes, began in late 2009. The HEWH, under the brand name GeoSpring™, was initially manufactured in China. In 2011, GE initiated efforts to expand their domestic manufacturing operations at the Appliance Park facility in Louisville, KY. Approximately $1 billion was invested to establish US-based manufacturing operations for several products including French-Door Bottom Freezer Refrigerators, Clothes Washers & Dryers, Dishwashers, etc., as well as the HEWH. In February 2012, GE began manufacturing operations for the HEWH at Appliance Park, the first new product line to open at the facility since 1957. The new production line has created about 1300 US manufacturing jobs (http://www.geappliances.com/heat-pump-hot-water-heater/; also see press releases at http://pressroom.geappliances.com/news/ge-opens-first-new-manufacturing-228269 and http://www.courier-journal.com/article/20120209/BUSINESS/302100009/Production-heats-up-for-new-water-heater-at-Appliance-Park?odyssey=tab%7Cmostpopular%7Ctext%7CFRONTPAGE&nclick_check=1).

B. Laboratory performance testing

Energy Factor (EF) and 1st hour capacity testing – prior to reliability test start. Four of the original DG units – 18, 20, 23, and 27 – were subjected to 24-hr standard use and 1st hour tests during June and July of 2008 prior to starting the reliability test runs. These tests were conducted using the ambient air [67.5 °F, ±1°F, dry bulb temperature and 50% RH, ±1%RH], tank entering water [58 °F, ±2 °F], and tank set point [135 °F] conditions prescribed by the DOE test procedure for electric heat pump water heaters (HPWH) as given in the US Code of Federal Regulations (2010). Testing was conducted in the small, one-room appliance test cell in the ORNL Buildings Technology Research and Integration Center (BTRIC) laboratory, Figure 4. Table 12 provides a list of the instrumentation on the EF test units and on the reliability test units as monitored by the ORNL laboratory data acquisition system (DAS) – note that not all of the reliability test units had all of the instrumentation points given in Table 12. Figure 5 is a screen shot from the DAS control computer used to collect and reduce the lab performance tests covered in this section. In addition to the data monitored by the ORNL DAS several measurements taken by the test HPWH units' onboard control system were also monitored – these are listed in Table 13.

Figure 4. HPWH in small appliance test chamber.

Table 12. HPWH performance test instrumentation – ORNL DAS

DAS channel	Type	Description
1	TC[a] 1	Cin – refrigerant temperature leaving compressor
2 [b]	TC 2	C1 – TC mounted on condenser under tank insulation, bottom
3 [b]	TC 3	C2 – TC mounted on condenser under tank insulation
4 [b]	TC 4	C3 – TC mounted on condenser under tank insulation
5 [b]	TC 5	C4 – TC mounted on condenser under tank insulation
6 [b]	TC 6	C5 – TC mounted on condenser under tank insulation
7 [b]	TC 7	C6 – TC mounted on condenser under tank insulation
8 [b]	TC 8	C7 – TC mounted on condenser under tank insulation
9 [b]	TC 9	C8 – TC mounted on condenser under tank insulation
10 [b]	TC 10	C9 – TC mounted on condenser under tank insulation, top
11	TC 11	Cout – refrigerant temperature leaving condenser
12	TC 12	Compressor shell temperature (top of compressor)
13	TC 13	Water temperature entering HPWH
14	TC 14	Water temperature leaving HPWH
15	TC 15	Ein – refrigerant temperature entering evaporator (after expansion valve)
16 [b]	TC 16	E1 – TC mounted on evaporator return bend
17 [b]	TC 17	E2 – TC mounted on evaporator return bend
18 [b]	TC 18	E3 – TC mounted on evaporator return bend
19 [b]	TC 19	E4 – TC mounted on evaporator return bend
20	TC 20	Eout – refrigerant temperature leaving evaporator
21	TC 21	Tw1 – tank water temperature (top of TC tree)
22	TC 22	Tw2 – tank water temperature
23	TC 23	Tw3 – tank water temperature
24	TC 24	Tw4 – tank water temperature
25	TC 25	Tw5 – tank water temperature
26	TC 26	Tw6 – tank water temperature (bottom of TC tree)
27	Watt meter	HPWH total power
28	Voltage	HPWH power supply voltage
29 [b]	Pressure	Pdisch – refrigerant pressure at compressor discharge (psig)
30 [b]	Pressure	Preturn – refrigerant pressure at condenser exit (psig)
31 [b]	Pressure	Psuction – refrigerant pressure at compressor suction (psig)
32	Water flow	Cold water flow entering HPWH (gpm)
33 [b]	Watt meter	HPWH compressor power
34	TC 27	Refrigerant temperature entering compressor
35	TC 28	Refrigerant temperature at expansion valve inlet
36	TC 29	Test chamber ambient air temperature
37 [b]	TC 30	Air temperature entering HPWH shroud (heat pump cover)
38 [b]	TC 31	Ta1 – air temperature leaving HPWH evaporator
39 [b]	TC 32	Ta2 – air temperature leaving HPWH evaporator
40	RH	Test chamber relative humidity (%RH)
41	TC 33	Chamber cold water main supply temperature
42 [b]	TC 35	TC mounted on HPWH tank wall under insulation next to controller tank water temperature sensor

[a] Type T thermocouple (°F)

[b] Not included on DC or PP test units

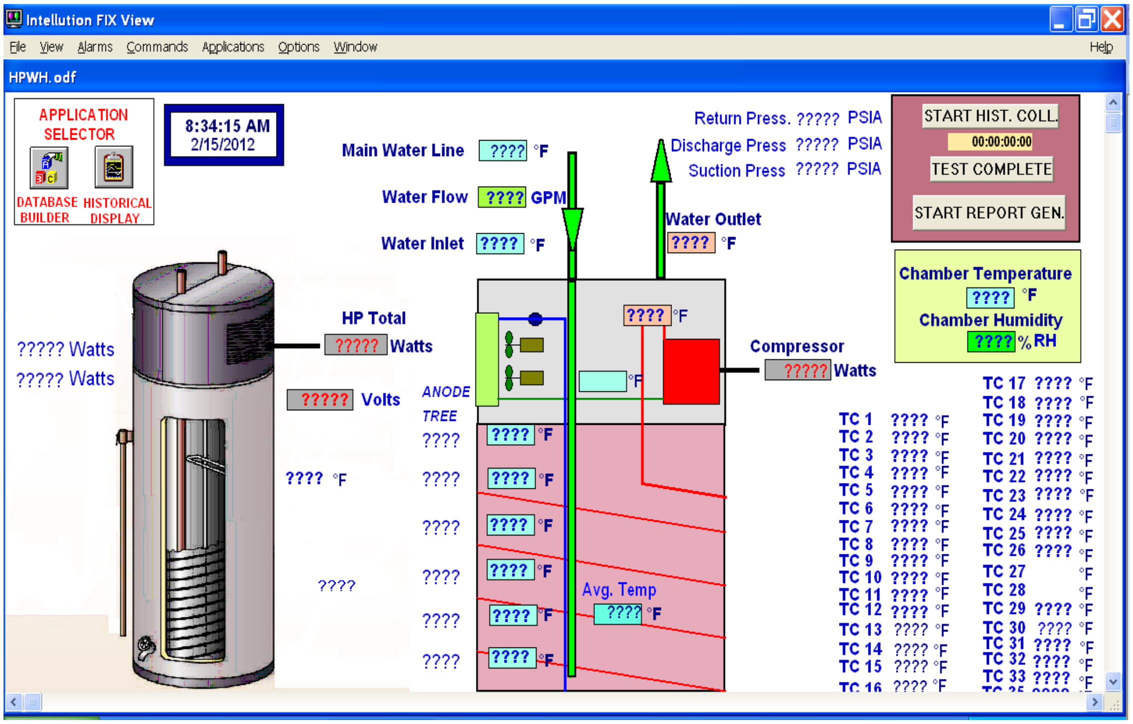

Figure 5. Main HPWH test control screen - DAS control computer

Table 13. Parameters monitored by HPWH onboard controller

Item	Type	Description
TH2	Thermistor	Tank temperature (°F)
TH3a	Thermistor	Refrigerant temperature entering evaporator (°F)
TH3b	Thermistor	Refrigerant temperature leaving evaporator (°F)
TH4	Thermistor	Refrigerant compressor discharge (°F)
TH5	Thermistor	Ambient air temperature (°F)
UE Ind	on/off	Upper electric element on/off indicator
LE Ind	on/off	Lower electric element on/off indicator
Comp Ind	on/off	Compressor on/off indicator
Fan1 Ind	Rpm	Fan 1 speed
Fan2 Ind	Rpm	Fan 2 speed

Results of the “as received” EF and 1st hour tests conducted on the four DG units are summarized in Table 14.

Table 14. Energy Factor (EF) and 1st hour rating test results for DG units 18, 20, 23, and 27

Unit	EF test result	1st hour test result (gallons)
DG-18	2.05	57.8
DG-20	2.09	56.2
DG-23	2.00	55.1
DG-27	1.97	55.8
Average values	2.03	56.2

The results above indicated that the early DG prototypes just met the energy efficiency criteria required for the Energy Star electric storage water program (EF≥2.0) as given on the Energy Star website - http://www.energystar.gov/index.cfm?c=water_heat.pr_crit_water_heaters. Energy Star also specifies a minimum 1st hour rating (FHR) of 50 gallons – the DG units achieved that criteria with more than a 10% cushion.

Heat up tests during reliability run. The procedure below was used for the periodic heat up tests performed as part of the Reliability Test protocol (see Tables 4 and 5). All tests performed at ORNL were done with the test units in Hybrid Mode (default mode).

> Heat Up Test Procedure:
>
> The test procedure below was used for the periodic heat up tests (HUT) done during the reliability test run (see Tables 4 and 5 for HUT schedule). A 135 °F tank set point and 240 volts AC power supply was used for all HUT tests performed at ORNL. Chamber ambient conditions for the HUT testing were 67.5 °F, ±1°F, dry bulb temperature and 50% RH. Water temperature entering the test units was controlled to 58 °F, ±2 °F.
>
> 1. After unit has been stable for 30±5 minutes draw 10.75 ± .25 gallons of water at a rate of 3±.25 gpm.
> 2. If lower heat source energizes (evaporator fan starts) before the 10.75 gallon draw ends, then complete 10.75 gallon draw, allow the water heater to stabilize to set point conditions, and go to step 4.
> 3. If lower heat source does not energize during the 10.75 gallon draw, then wait a maximum of 10 minutes to see if lower heat source will energize. If it does not do so, then draw more water until it does. After the lower heat source energizes cease the water draw and allow the water heater to stabilize to set point.
> 4. Allow unit to sit 5 to 10 minutes in stabilized conditions.
> 5. Draw water at 3±.25 gpm until the Upper Element energizes. Water draw must stop within 5 seconds after Upper Element energizes.
> 6. For DC and PP units: when the Upper Element stops and switches to Lower Element, switch the unit to eHeat Mode. This will switch the lower heating source to Compressor.
> 7. Allow water heater to stabilize this second time. (Recover water temperature back to set point).
> 8. Allow unit to sit a minimum of 30 minutes in stabilized conditions, then;
> 9. Stop Data Logger and HyperTerminal session.
> 10. Capture all data into spreadsheet.

Table 15 provides overall averages for the principal unit performance parameters measured in the HUT tests. Figures 6-8 illustrate typical results of HUT tests for a DG, DC, and PP unit, respectively. Figures 9 and 10 compare the time and energy required, respectively, for the different prototypes to heat tank water through a 20 °F rise (110 to

130 °F). These results indicate that the DC and PP units consumed ~22% less energy than the DG units on average while taking ~8% less time to heat the tank water from 110 °F to 130 °F. Further, the data in Figures 9 and 10 do not indicate any deterioration in performance with time and increased cycle count for any of the prototypes tested by ORNL over the 2008-2010 time frame.

As described earlier, ORNL conducted Energy Factor (EF) tests on four of the original DG units yielding an average EF of 2.0 (Table 14). Based on the HUT results in Table 15 it was estimated that the EF of ~2.4-2.5 for the DC and PP prototypes and the ultimate production units. This was confirmed in May-June 2010 when GE sent an instrumented production model of their HPWH product to ORNL. We conducted three 24-hr use tests on this unit "as received" and measured an average EF of 2.4. First hour tests on this unit yielded an approximate value of 62-63 gallons for FHR. Both values are well in excess of the minimums needed for Energy Star.

Table 15. Average test unit performance from heat up tests (HUT) performed throughout reliability test run.

	Time to heat tank water from average temperature of 110 °F to 130 °F			Energy required to heat tank water from average temperature of 110 °F to 130 °F		
	Average time, minutes	Standard deviation, minutes	% change from DG	Average energy use, kWh	Standard deviation, kWh	% change from DG
DG units (avg of 34 tests)	99.31	5.09	--	1.005	0.042	--
DC units (avg of 44 tests)	91.52	3.16	-7.8%	0.790	0.018	-21.4%
PP units (avg of 8 tests)	91.71	2.17	-7.7%	0.782	0.011	-22.2%

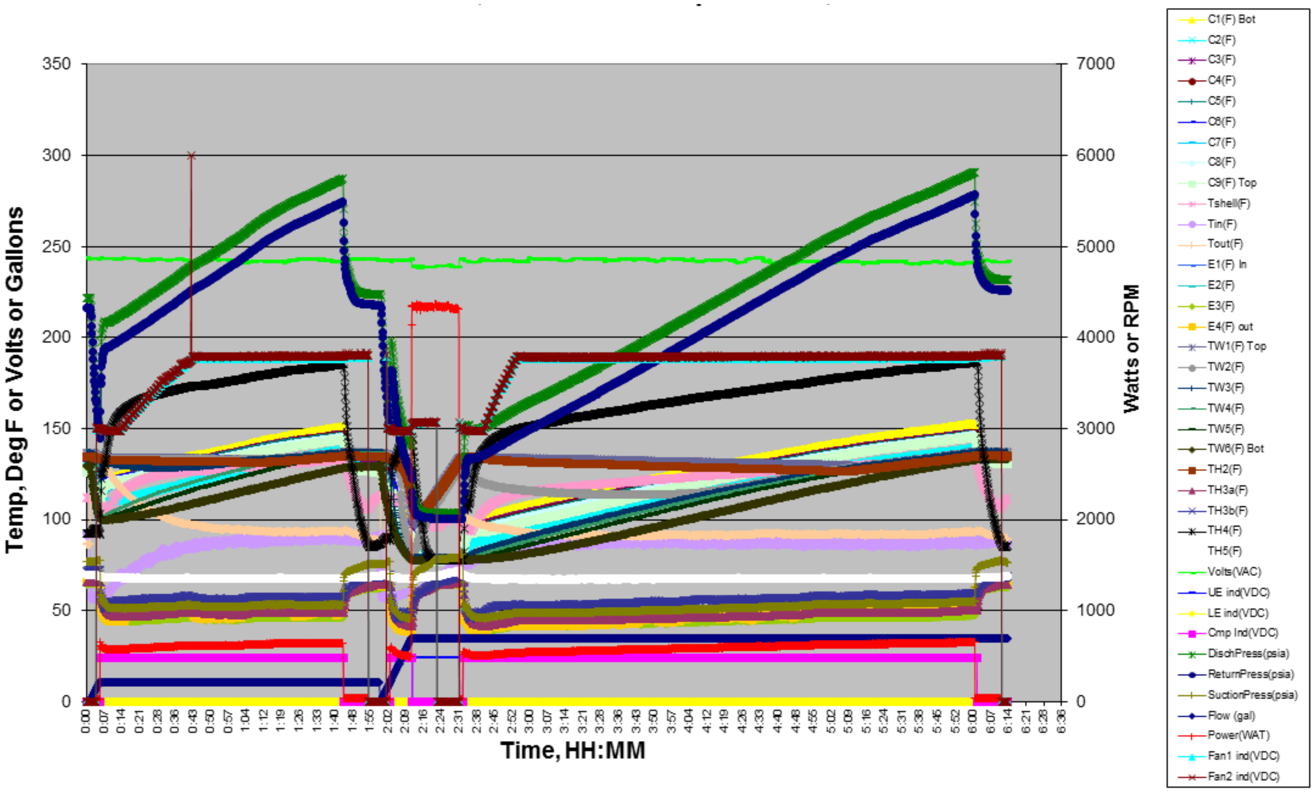

Figure 6. Typical heat up test (HUT) results for DG test units.

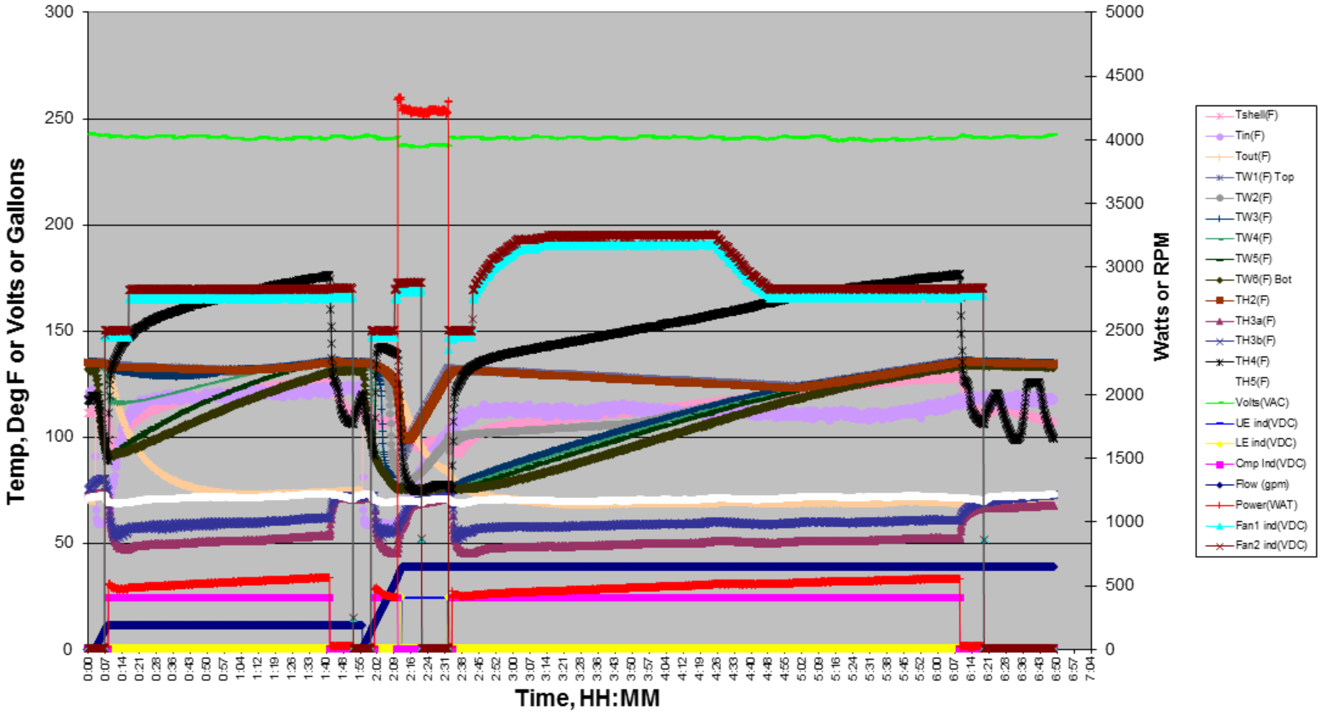

Figure 7. Typical HUT results for DC test units.

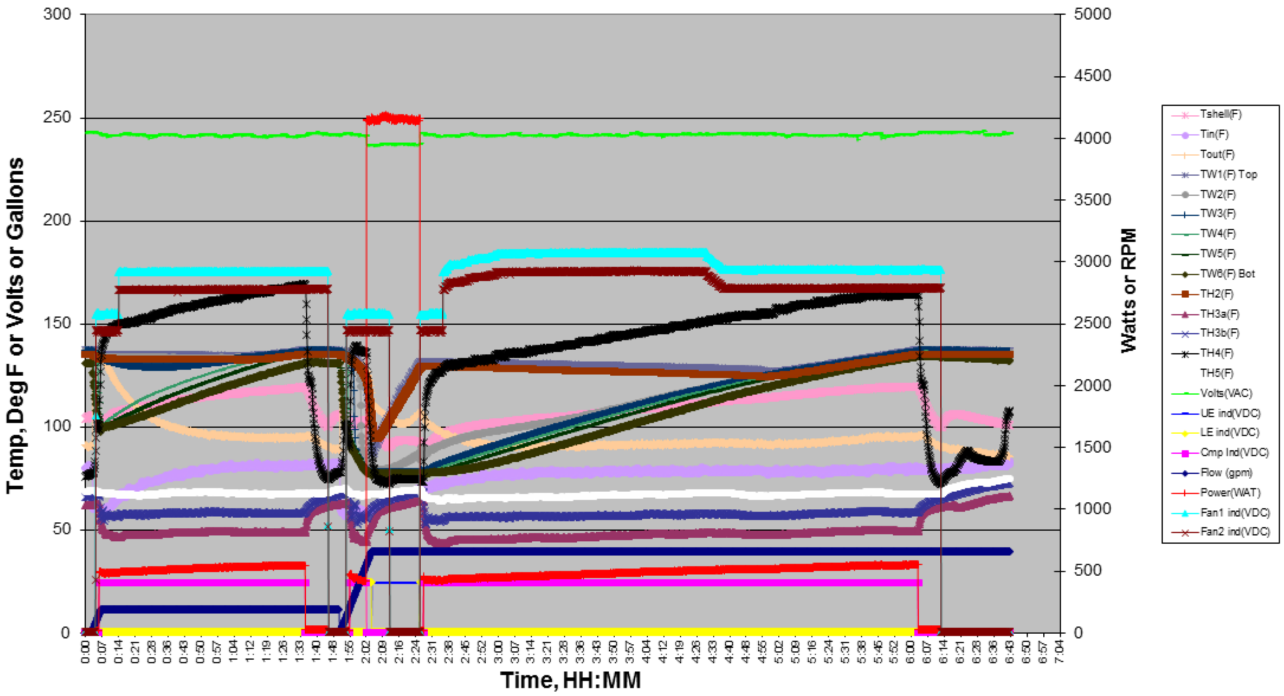

Figure 8. Typical HUT results for PP test units.

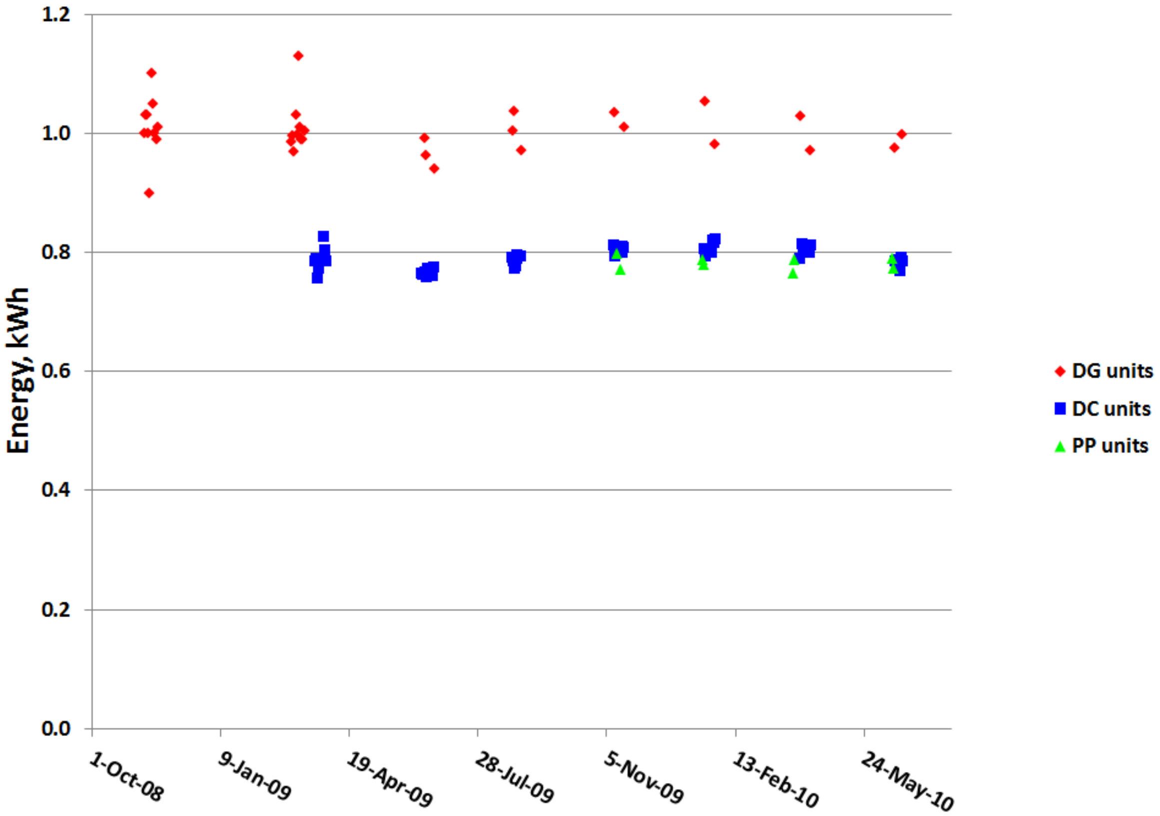

Figure 9. Comparison of energy required to heat tank water from 110 to 130 °F – DG vs. DC and PP prototypes.

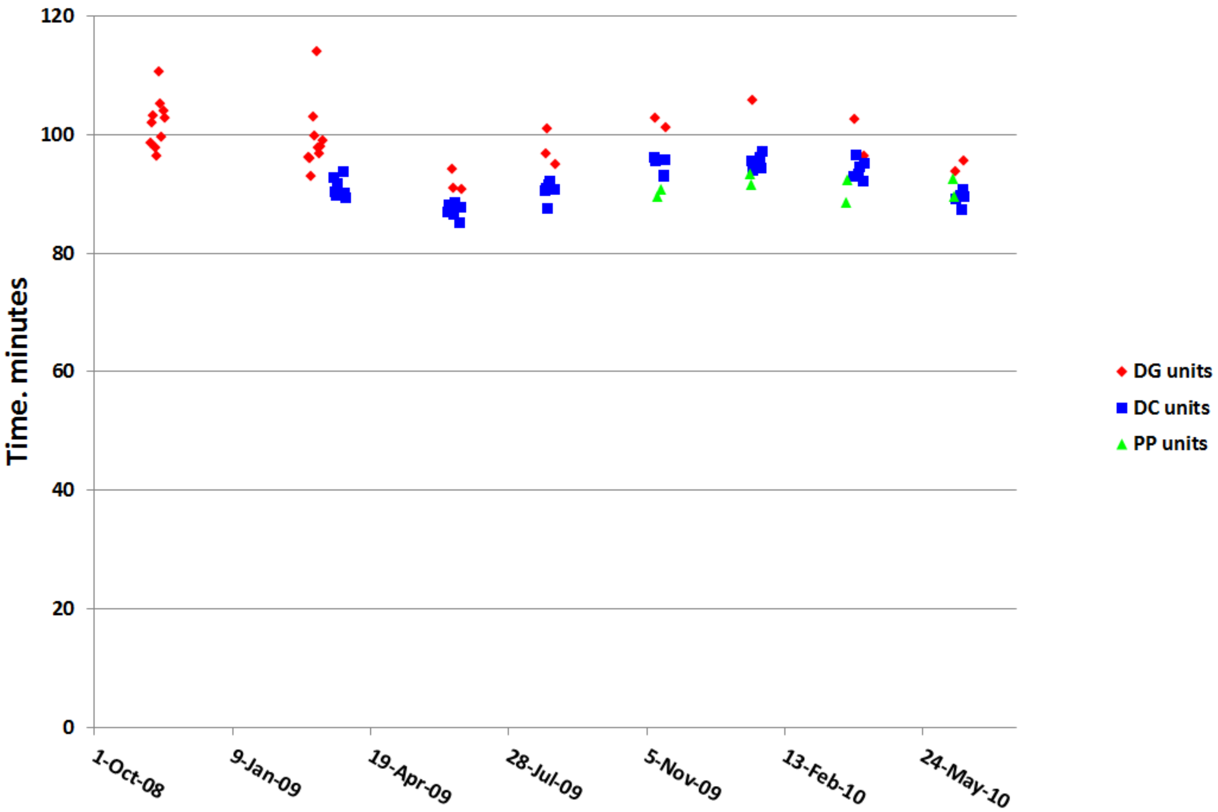

Figure 10. Comparison of time required to heat tank water from 110 to 130 °F – DG vs. DC and PP prototypes.

Laboratory performance tests with alternative refrigerant R-1234yf. In May 2010, GE provided ORNL with an instrumented production unit for purposes of evaluating its performance with R-1234yf, a low global warming potential (GWP) alternative to R-134a. Note – much of the material in this section dealing with the "drop in" testing of R-1234yf in the test unit was also documented in a paper presented at the 10th International Energy Agency (IEA) Heat Pump Conference (Murphy, et al, 2011).

Initial "as received" baseline first-hour rating and 24-hour simulated use tests (3 of each) were conducted with this unit to establish its first-hour rating and energy factor with the standard charge of R-134a (750 g). The results gave an average first-hour rating of ~62 gallons and an average energy factor of ~2.4 - closely matching the corresponding Energy Star-listed values of 63 gallons and 2.35, respectively.

The hydrofluoroolefin (HFO) compound R-1234yf has been suggested as a near drop-in replacement for the hydrofluorocarbon (HFC) compound R-134a because the two compounds have very similar thermodynamic and transport properties, based on the REFPROP database (Lemmon et al. 2007). Although R-1234yf is mildly flammable, it has an estimated global warming potential (GWP) of 4, substantially lower than that of R-134a (GWP=1370).

The purpose of this testing was to determine the efficiency and capacity performance of a GE production model operating with R-1234yf relative to that with the baseline R-134a.

Preparations. After leak-checking the system and removal of the standard charge of R-134a (leaving the associated oil in the unit because of its reported compatibility with R-1234yf), the refrigerant system was evacuated to high vacuum with a vacuum pump. No hardware or software modifications were made to accommodate the alternate refrigerant. Because the properties of R-1234yf were (similar, but) not identical to those of R-134a, matrix of preliminary heat-up tests at selected charge levels were planned over a range of ambient air temperatures to establish the correct R-1234yf charge within the limits of the standard system hardware and software. The target figure-of-merit suggested by GE to judge charge adequacy was a minimum evaporator exit superheat of approximately 7°F.

The liquid volume (758 ml) of the standard R-134a mass charge (750 g) was calculated at condenser operating conditions (approximately 160°F) corresponding to the maximum water temperature setting (140°F). This value served as the beginning volume estimate of the appropriate R-1234yf charge. The corresponding mass charge estimate (663 g) was determined from the liquid density of R-1234yf at the same condensing temperature. To avoid starting with an overcharged condition and potentially wasting our limited supply of the refrigerant, the initial charge of R-1234yf was chosen to be 87% of this value or 575 g.

Charge-Determination Tests. After the initial charge was weighed into the unit, HUT tests were conducted at four different ambient air temperatures - 50 °F, 70 °F, 90 °F, and 110 °F (10 °C, 19.7 °C, 32.2 °C, and 43.3 °C). As shown in Figure 11, the minimum evaporator exit superheat (in non-dimensionalized form relative to the R-134a baseline value at 50 °F ambient air temperature) was encountered at the lowest ambient air temperature, but frost formation on the evaporator prevented proper operation at this condition. The minimum condenser exit subcooling (in non-dimensionalized form relative to the R-134a baseline value at 50 °F ambient air temperature) is shown in Figure 12 to be about 0.89. Both the frost formation and the relatively low subcooling level were interpreted to be indications of a refrigerant undercharge situation. Because of the frozen evaporator at the lowest ambient temperature with the starting charge, emphasis was placed on monitoring evaporator performance as refrigerant charge was gradually increased.

The charge (represented on the abscissas of Figures 11 and 12 in non-dimensionalized form relative to the estimated appropriate mass charge of R-1234yf) was then increased to about 0.92 and the HUT tests were repeated. Although superheat decreased and subcooling increased after the refrigerant addition as expected, frost was again observed on the evaporator surface at the lowest ambient air temperature condition. The next refrigerant increment brought the total relative charge to about 0.98, within 2% of the original estimate for the appropriate charge. The associated heat-up tests showed no frost formation on the evaporator, identifying this as the minimum acceptable charge. At this point the evaporator exit superheat was about 16% higher than the target of ~7 °F.

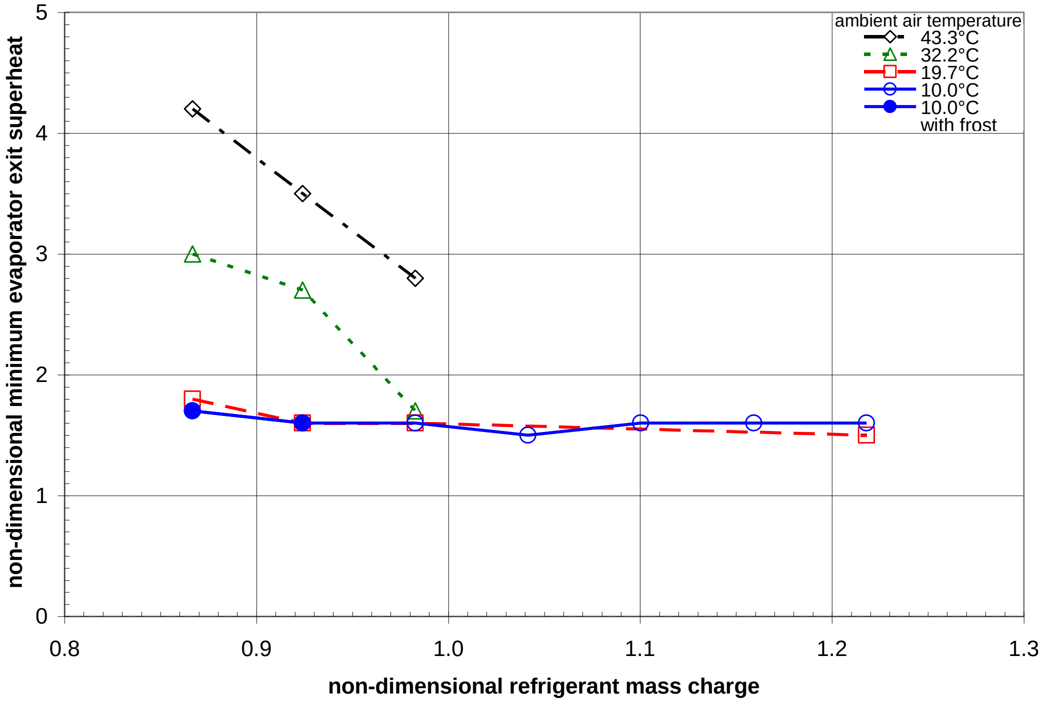

Figure 11. Evaporator superheat versus charge for various ambient air temperatures

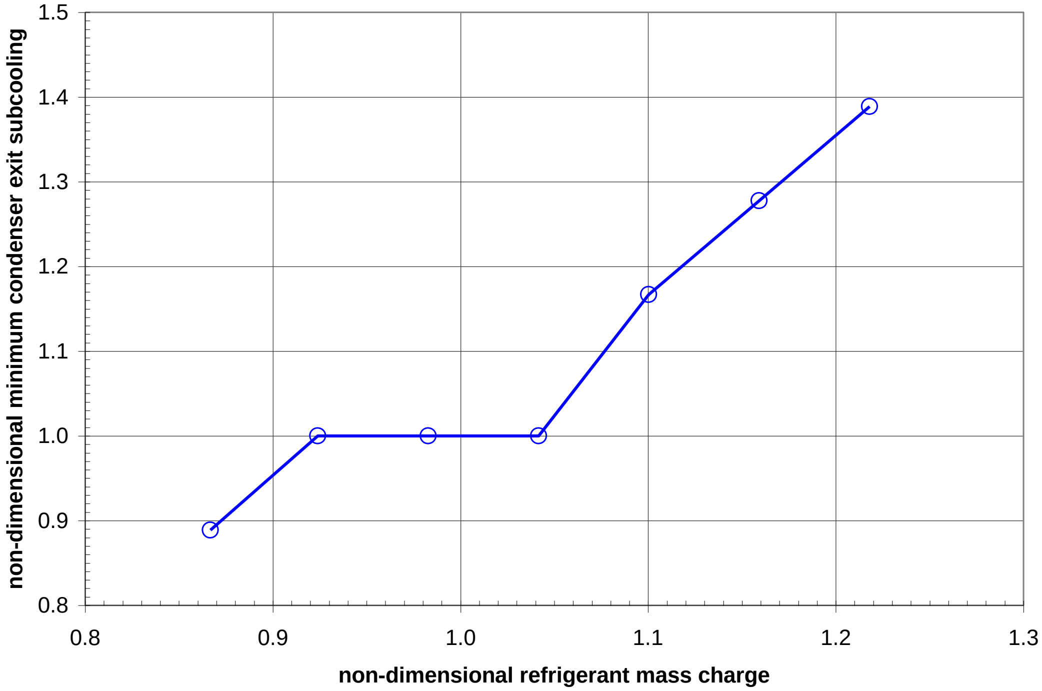

Figure 12. Condenser subcooling versus charge for 10°C ambient air temperature

As with the initial charge, the minimum evaporator exit superheat was achieved at the lowest ambient air temperature (50 °F or 10 °C). However, under these conditions, although the subcooling value reached 100% of the corresponding value for the unit operating with the standard R-134a charge, the observed minimum evaporator exit superheat was about 60% above the corresponding R-134a system value. In addition, with the normalized R-1234yf charge at the 98% level the normalized (relative to the R-134a baseline value at 10°C ambient air temperature) recovery energy consumption and recovery time, given in Figure 13, were approximately 14% and 15%, respectively, greater than the corresponding R-134a system values.

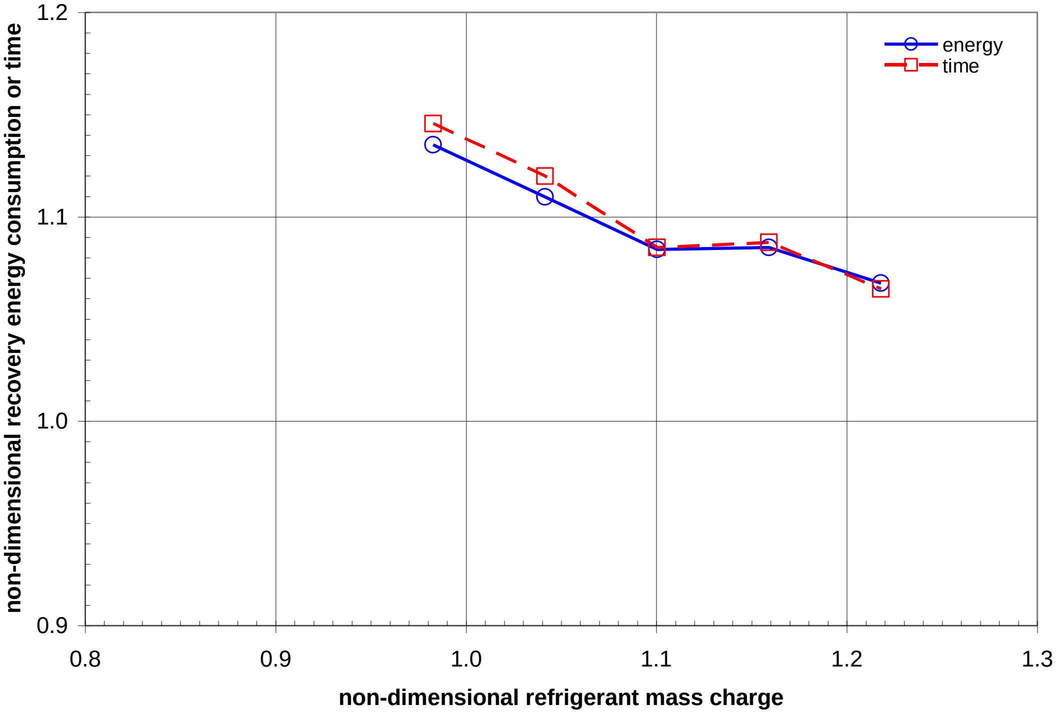

Figure 13. Recovery energy consumption/time versus charge for 10°C ambient air temperature

In order to possibly (a) reduce the evaporator superheat (and, consequently, improve its heat transfer effectiveness), (b) reduce the recovery energy consumption, and (c) reduce the recovery time, more refrigerant was added. Stepwise addition of R-1234yf (up to a total charge about 22% in excess of the original estimate for the "appropriate" charge) produced no significant change in the minimum evaporator exit superheat (see Figure 11), but did produce a significant increase (for non-dimensional charges above 1.04) in the minimum condenser exit subcooling (see Figure 12) to about 39% above the corresponding R-134a system value. As shown in Figure 13, the heat pump recovery energy consumption and time decreased somewhat as the non-dimensional charge increased from 0.98 to 1.10, but their rate of decrease was much lower in the charge range from 1.10 to 1.22 -- ending with about 107% and 106% of the R-134a system values, respectively, at the highest charge. The close proximity of the two lines in Figure 13 reflects the fact that the average power draw for the unit during the recovery period was almost identical for the R-1234yf-charged system as it was for the R-134a-charged system. From these results, it was judged that further refrigerant charge increases were not likely to produce significant reductions in either minimum evaporator exit superheat (toward the goal of 7°F) or reductions in recovery energy consumption. We suspected that the thermostatic expansion valve used in the system (for R-134a) was limiting the minimum evaporator exit superheat.

Performance Testing. To complete the “drop-in” performance characterization of the GE HPWH unit with R-1234yf both a first-hour rating test and a 24-hour simulated use test were conducted with the R-1234yf charge at ~808 g (22% in excess of the original estimated R-1234yf charge requirement). The resultant first-hour rating was determined to be approximately 62 gallons—matching the value previously determined for the same unit operating with the standard charge of R-134a. This was expected since the primary determinants of first-hour rating in this situation are the characteristics associated with the water tank (volume, mixing characteristics, etc.) and the upper element (location, power, activation period, etc.)—neither of which were altered by the refrigerant replacement. The resultant energy factor or EF was determined to be approximately 6% lower than the ~2.4 value determined for the unit in “as received” condition.

Some operational parameters that differed during the 24-hour simulated test using R-1234yf from those observed during the baseline R-134a tests were: evaporator exit superheat (higher), condenser exit subcooling (higher), condenser pressure drop (higher), evaporator temperatures (lower), fan speeds (higher), fan energy consumption (higher), compressor energy consumption (higher), and refrigeration system heating capacity (lower).

Performance Testing Results Interpretation. Our speculation was that the mismatch between the R-134a-charged thermostatic expansion valve of the production unit and the R-1234yf-charged refrigeration system limited refrigerant flow with R-1234yf causing underutilization of the evaporator (increased dry regions, increased exit superheats, depressed evaporator temperatures). It also probably led us to create a slightly overcharged condition in the condenser (added liquid leg backup, increased condenser exit subcooling, increased condenser pressure drop) in our attempt to reduce the evaporator exit superheat. Side effects of the reduced evaporator temperatures and increased superheat were likely the observed increased fan speeds and increased fan energy consumption. The combination of various effects likely led to slightly increased compressor energy consumption and the small reduction in EF values observed.

To test this hypothesis we obtained a similar TXV with adjustable superheat control (by means of spindle setting) and a larger orifice to replace the factory TXV. We conducted another charge determination test series with the revised TXV and found that the same 1234yf charge (808g) resulted in minimizing the tank recovery energy use and time but with lower evaporator superheat. Figures 14 and 15 illustrate relative COP and capacity over the 105-130 °F tank temperature range for a base R-134a HUT test and the R-1234yf HUT tests (drop-in, and with revised TXV). Both COP and capacity for R-1234yf with the revised TXV were slightly better than for the drop-in case. Energy factor and 1st hour tests were conducted with the revised system and confirmed that unit EF performance was slightly improved. The EF result in this case was about 3% below the "as received" value of 2.4 and the FHR was again the same, 62 gallons. Resources available did not permit further system optimization for R-1234yf but based on the limited work conducted, it is surmised that a fully optimized R-1234yf design may very closely match the Energy Factor of the current R-134a based product without compromising FHR.

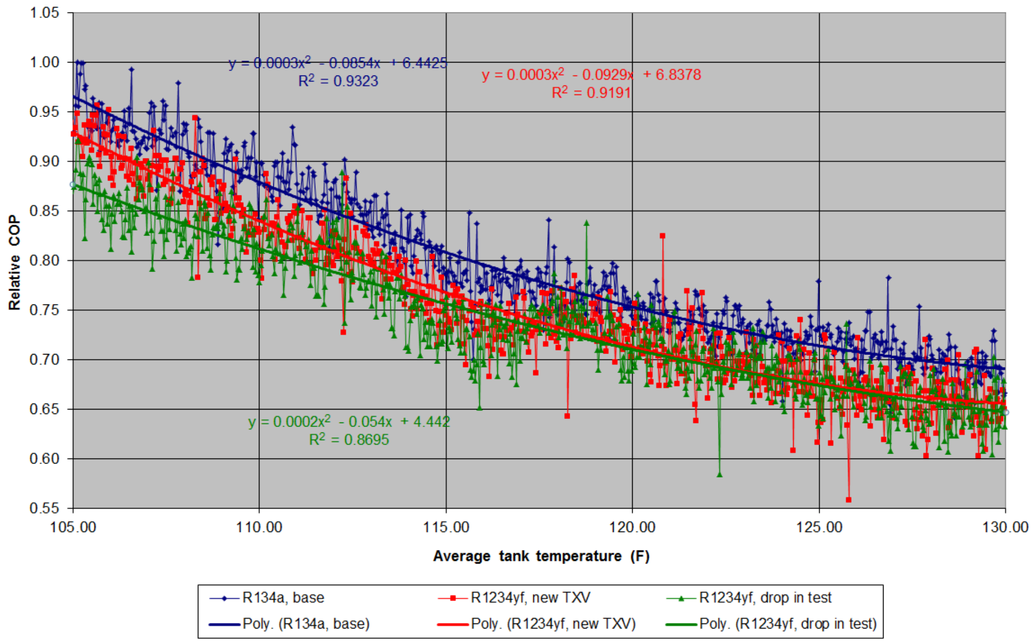

Figure 14. Average relative HPWH COP while raising the average tank water temperature from 105 °F to 130 °F (COP relative to baseline R-134a COP @ 105 °F).

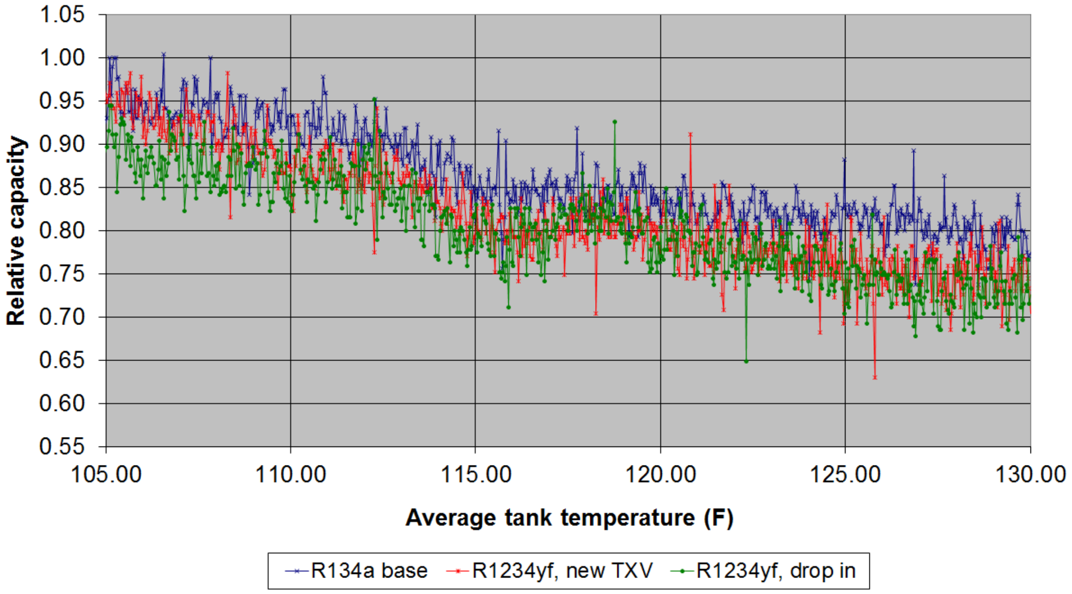

Figure 15. Average relative HPWH heating capacity while raising the average tank water temperature from 105 °F to 130 °F (relative to baseline R-134a capacity @ 105 °F).

C. Field evaluation at ZEBRAlliance site – impact on HVAC energy use due to indoor location

In August 2010 GE provided one of their production HEWH products for installation at one of the field test houses in the ORNL ZEBRAlliance site at the Wolf Creek development in Oak Ridge, TN. This unit was installed inside the conditioned space of the house. The following section describes the performance of the HEWH and its impact on the house's HVAC system. Note – the material in this section is summarized from a paper presented at the 2012 ASHRAE Annual Meeting in San Antonio, TX (Munk, et al, 2012).

HPWHs located within the conditioned space of a home will cool the air and increase the space heating load in the winter and decrease the space cooling load in the summer. The net impact of this effect was evaluated over the one year period from December 2010 to November 2011 under simulated home occupancy conditions. The house has a clothes washer, dishwasher, and shower that were operated on a schedule to draw between 50 and 60 gallons/d (189-227 L/d). The shower was adjusted throughout the year in order to maintain an average temperature of 105°F (40.5°C) at the shower head. The HPWH was installed in a 3 ft. by 10 ft. (0.9 m by 3.0 m) utility closet with two louvered doors connecting to the laundry room. It was set to provide 120°F (48.9°C) water and was switched between standard electric mode and heat pump only mode every other week. Space conditioning was provided by an air-source heat pump (ASHP) that maintained a temperature of 71°F (21.7°C) in the heating season and 76°F (24.4°C) in the cooling season. The indoor section of the ASHP was installed in the same utility closet as the HPWH.

Two different approaches are used to estimate the HPWH impact on space conditioning energy use for the test period, both for heating and cooling seasons and the net annual impact. First an energy balance was performed on the water heater (WH) to determine the net impact (increase or decrease) to the space conditioning load on the ASHP. These load impacts together with the average measured ASHP seasonal performance factors were then used to estimate the net impact on HVAC energy use. Second, the measured energy use of the ASHP was analyzed directly to determine the net change in its energy use due to HPWH operation.

Energy balance on water heater. Using a simple energy balance on the WH itself, shown in Figure 16 and Equation 1, the net heat transferred between the water heater and its surroundings can be calculated. A positive value for Q indicates heat transfer into the WH, and a negative value indicates heat transfer out of the WH.

$$Q = \dot{m}_w c_{p,w}(T_{out} - T_{in}) - W_e \tag{1}$$

where $\dot{m}_w = water\ flow$

$c_{p,w} = \ water\ specific\ heat$

T_{out} = hot water leaving temperature
T_{in} = cold water inlet temperature
W_e = energy input to WH
Q = heat transfer

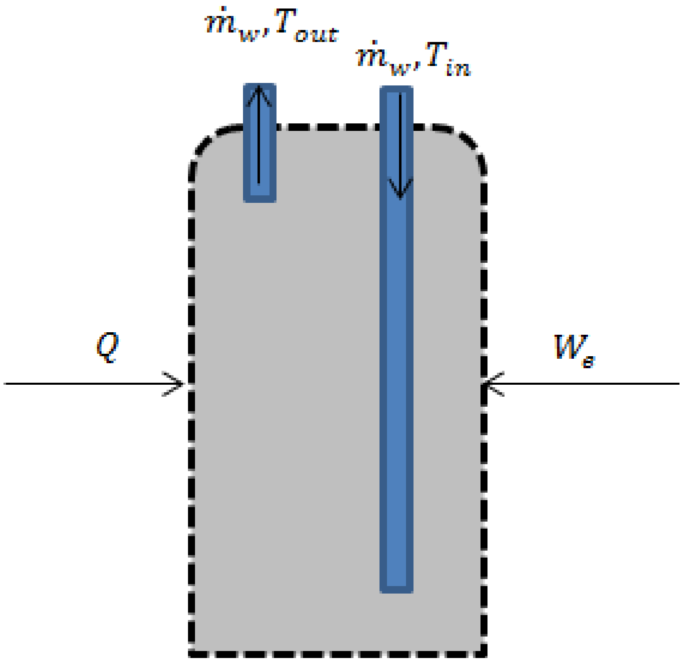

Figure 16. Water heater energy balance

Table 16 provides average daily heat transfer to/from the WH, WH energy use, and hot water use for each month of the test period. Annual as well as heating and cooling season averages are also given. In the heating season, the additional space heating load due to HPWH operation (difference between the average WH heat transfer in the HP mode and the standard mode) is 6685 Wh/d. For the cooling season the average daily net space cooling provided by the HPWH is 5302 Wh/d.

These values were used in conjunction with the measured heating and cooling season average coefficient of performance (COP) values for the ASHP to determine the average impact on space conditioning energy use as seen in Equations 2 and 3. In the heating season, the average COP for the ASHP was 3.11, which means that in order to make up for the space cooling effect of the HPWH, the ASHP must use an additional 2.15 kWh/d on average. In the cooling season, the average COP of the ASHP was 4.54, which would result in 1.17 kWh/d less space conditioning energy use due to the HPWH's space cooling effect.

$$\Delta W_{e,Heat,SC} = \frac{(Q_{Heat,HPWH} - Q_{Heat,Standard})}{COP_{Heat}} \qquad (2)$$

$$\Delta W_{e,Cool,SC} = \frac{(Q_{Cool,Standard} - Q_{Cool,HPWH})}{COP_{Cool}} \qquad (3)$$

Over the course of the year, this averages out to an additional 0.48 kWh/d of space conditioning energy use in this house at the test location as shown in Equation 4. While this number will vary based location in the country, on the performance of each home's space conditioning equipment, and total hot water use and usage patterns (and other factors), it will still typically be nearly an order of one magnitude less than the annual water heating energy use savings between a HPWH and a standard electric water heater, which in this study was 5.90 kWh/d on average.

$$\Delta W_{e,annual,SC} = \frac{(\Delta W_{e,Heat,SC} d_{Heat} + \Delta W_{e,Cool,SC} d_{Cool})}{(365)} \qquad (4)$$

Table 16. Average monthly field performance and hot water use; HPWH vs. Standard WH

	HPWH Mode			Standard Mode		
Month	Heat Transfer kWh/d	Energy Use kWh/d	Water Use gal/d (L/d)	Heat Transfer kWh/d	Energy Use kWh/d	Water Use gal/d (L/d)
Dec 2010	5.40	3.14	53.7 (203.3)	-1.06	9.65	54.8 (207.4)
Jan 2011	6.10	3.47	56.2 (212.7)	-1.17	10.59	55.9 (211.6)
Feb 2011	6.07	3.51	57.0 (215.8)	-1.13	10.74	58.1 (220.0)
March 2011	6.08	3.53	60.8 (230.2)	-1.09	10.20	58.2 (220.3)
April 2011	5.53	3.37	59.6 (225.6)	-1.11	9.49	59.0 (223.3)
May 2011	4.62	2.93	57.1 (216.1)	-1.18	8.84	59.2 (224.1)
June 2011	3.96	2.64	56.3 (213.1)	-1.34	7.90	56.8 (215.0)
July 2011	3.57	2.47	52.5 (198.7)	-1.28	7.20	52.6 (199.1)
Aug 2011	3.55	2.53	53.8 (203.7)	-1.32	7.42	54.0 (204.4)
Sept 2011	3.86	2.48	53.8 (203.7)	-1.30	8.15	54.1 (204.8)
Oct 2011	4.14	2.74	54.2 (205.2)	-1.21	7.80	51.9 (196.5)
Nov 2011	4.65	2.99	53.5 (202.5)	-1.26	8.76	53.1 (201.0)
Heating Season Average	5.54	3.29	56.4 (213.5)	-1.14	9.79	56.0 (212.0)
Cooling Season Average	4.04	2.67	55.0 (208.2)	-1.26	7.99	55.2 (209.0)
Annual Average	4.79	2.98	55.7 (210.8)	-1.20	8.88	55.6 (210.5)

Measured space conditioning energy use. As noted earlier, the actual ASHP energy use was recorded throughout the test period as well. This data was summed over each day and tabulated along with the average outdoor air temperature and the operating mode of the HPWH. Prior to the analysis, the data was filtered by removing days when there was no energy use by the ASHP for space conditioning, as well as any days when there were known issues with the data. Days in which the operating mode of the HPWH was switched were also removed.

Heading season. The heating season data covered the time periods of 12/1/2010 to 4/20/2011 and 10/20/2011 to 11/30/2011 and is plotted in Figure 2. Also shown in Figure 2 are curve fits for the space heating energy use when the WH operated either in heat pump mode or standard mode, as a function of daily outdoor air temperature.

As seen in Figure 17, there is significant variance in the ASHP energy use that is not accounted for solely by the average outdoor air temperature. This variation appears to be much larger than the difference in energy use indicated by operating the HPWH in the HP mode. Applying the curve fits over the entire heating season suggests that on average the ASHP in this test house required 1.00 kWh/d more energy with the HPWH operating in the HP mode when compared to the standard mode. For outdoor temperatures below about 35°F (~1.7°C) the additional space heating energy use exceeded 2.00 kWh/d.

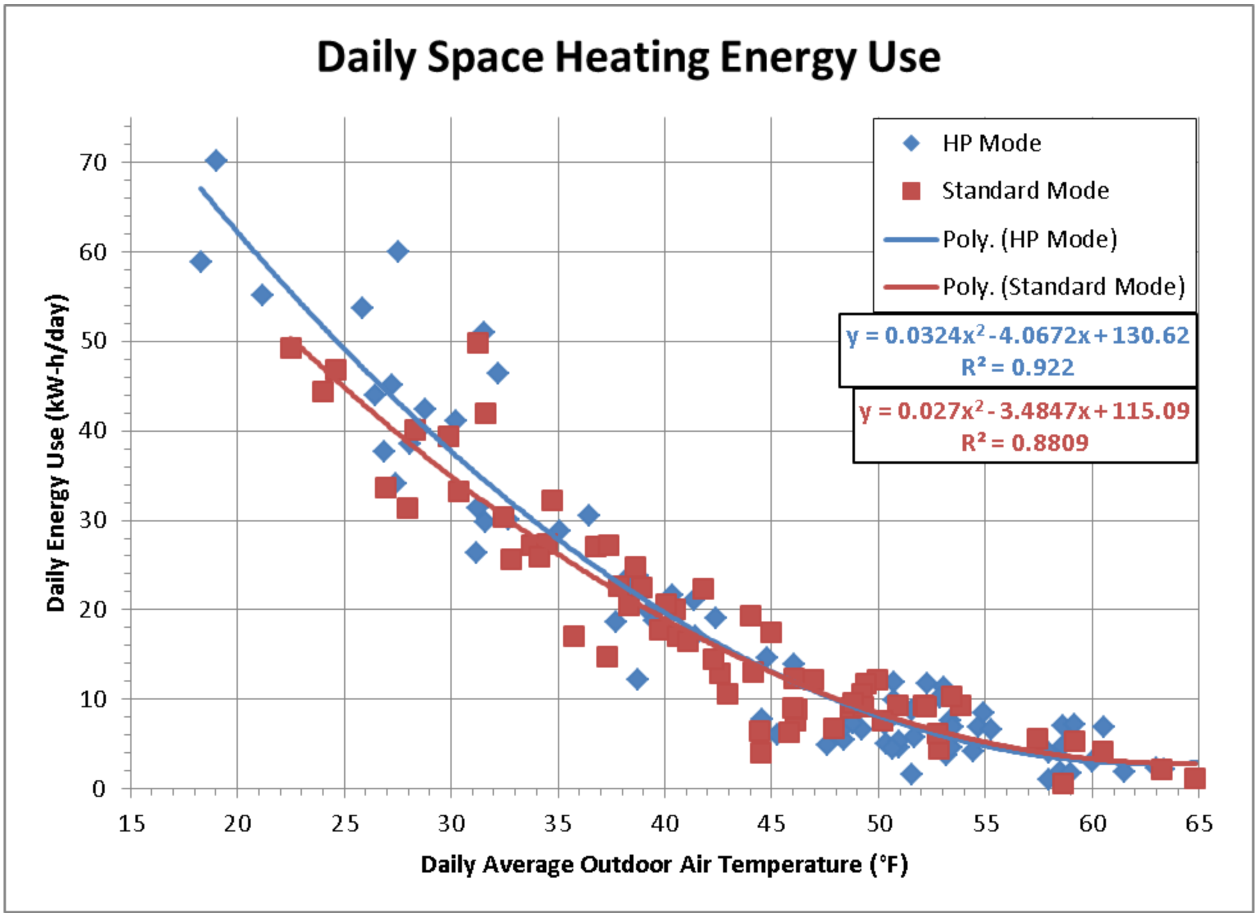

Figure 17. Daily space heating energy use vs. daily average outdoor temperature

Cooling season. The cooling season data, 4/21/2011 to 10/19/2011, was evaluated in the same fashion as the heating season data. Unfortunately, there was a zone damper in the central air distribution system that was malfunctioning during a large portion of test period. This caused higher than expected space cooling energy use and this data was removed before analysis. Figure 18 shows the resulting space cooling energy data which was available for the cooling season analysis. The curve fits shown in Figure 18 indicate that the ASHP used less energy when the WH was operating in the HPWH mode when compared to the standard mode at outdoor air temperatures above about 77°F (25°C). Applying these curve fits over the entire cooling season suggests that the ASHP in this house required about 0.21 kWh/d less energy on average with the HPWH operating in the conditioned space.

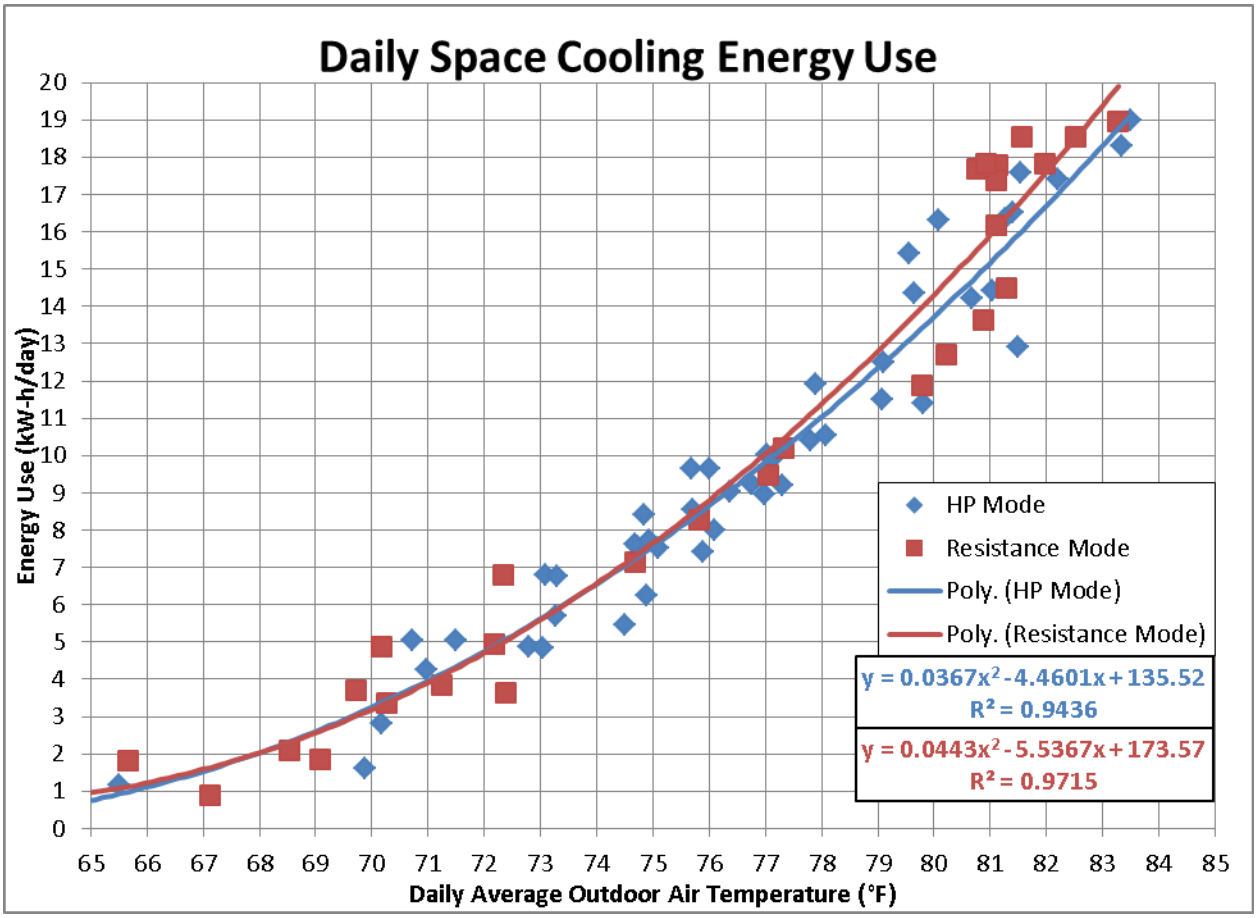

Figure 18. Daily space cooling energy use vs. daily average outdoor temperature

<u>Net annual HVAC energy impact</u>. Combining the heating and cooling seasonal curve fits in Figures 17 and 18 to the entire 2010-2011 test period, the annual net impact on HVAC energy use of the HPWH operating indoors was estimated to be about 0.39 kWh/d.

<u>Discussion & observations</u>. Both methods of analysis indicated a small net increase in space conditioning energy use over an entire year due to operating the HPWH within the conditioned space for the test house in its mixed-humid climate location. The energy balance analysis estimated an additional 0.48 kWh/d over the entire year, while the measured ASHP data estimated an additional 0.39 kWh/d. While the energy balance indicated a somewhat larger magnitude effect in both the heating and cooling seasons, when averaged over an entire year, the results were very close to those of the actual measured ASHP energy use data. Both methods indicate that the impact on space conditioning energy use of operating a HPWH inside the conditioned space of a residence in a mixed-humid climate is quite small in comparison with the water heating efficiency

gains achieved by HPWHs over standard electric storage WHs (~0.4-0.5 kWh/d vs. 5.90 kWh/d in this case).

This study has presented the results of a single case study in one location – Oak Ridge, TN. HVAC energy use impacts due to indoor location of a HPWH will vary for other homes based on climactic location, hot water use pattern, entering cold water temperature, HPWH efficiency, HVAC system efficiency, etc. However, in many cases, additional HVAC energy use will be substantially less than the associated water heating energy use savings between a HPWH and a standard electric water heater.

D. Analyses tasks

HEWH Design Evaluation

We used data from the initial DG unit EF tests (Table 14) to calibrate a steady-state modeling approximation to the tested configuration using the ORNL Heat Pump Design Model (HPDM) (Rice and Jackson, 2005). The approximation is primarily in our simplified treatment of the condenser in lieu of a more detailed and time-dependent tank/condenser wrap model. This quasi-steady-state modeling enabled us to evaluate the overall performance of the evaporator and condenser heat exchangers (HX). The calibrated model was used in 2008 to evaluate a number of refinements to the DG prototype design under consideration by GE (primarily new evaporator, fan, and compressor components) to improve the product's energy factor. This included looking at different compressor sizes and the effect of a postulated higher performance condenser on heating COP and capacity. Results of the analyses showed that it was possible to improve the rated energy factor (EF) from 2.0 (ORNL test results on early prototypes, see Table 14 above) by at least 10% to 2.20.

Initial projections of national energy savings impact from estimated HEWH market penetration – 1/1/2010-1/1/2020

Annual and cumulative energy savings estimates were prepared early in the project (September 2008) based on a number of assumptions.

1. The energy factor (EF) for the initial product will be at least 2.20 (based on results of the design evaluation presented above and discussion with GE project staff).
2. Market introduction occurs at the end of 2009.
3. Annual electric WH shipments total ~4.8 million per year for 2010-2019 (based on GAMA shipment data for 2006).
4. The HEWH displaces 10% of conventional electric storage water heater shipments each year leading to an installed "stock" of ~4.8 million HEWHs by 1/1/2020.

Using the method prescribed in the Energy Star® water heater analysis[12] annual savings per unit for the GE HEWH over a conventional electric storage WH is estimated to be ~2883 kWh/y. With the 10% new and replacement market penetration assumption, by the end of 2010 total savings are over 1.3 billion kWh or ~0.015 Quads (based on a 3.18 site-to-source electric conversion factor[13]). As illustrated in Figure 19, after ten years (by 1/1/2020) total cumulative source energy savings exceed 0.8 Quads and annual savings reach ~0.15 Quads/y. Cumulative consumer cost savings through 1/1/2020 are projected at ~$8 billion with annual savings by 2020 reaching ~$1.4 billion/y (based on 2006 average electricity price of $0.104[14]). Using the actual rated EF of 2.4 for the GE HEWH, these savings estimates would be ~0.9 Quads cumulative and 0.16 Quads/y by 2020.

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[12] Energy Star Residential Water Heaters: Final Criteria Analysis, April 2008.

[13] 2007 EERE Buildings Energy Data Book, September 2007.

[14] From Energy Information Administration data; average rates from January 2006 through December 2006.

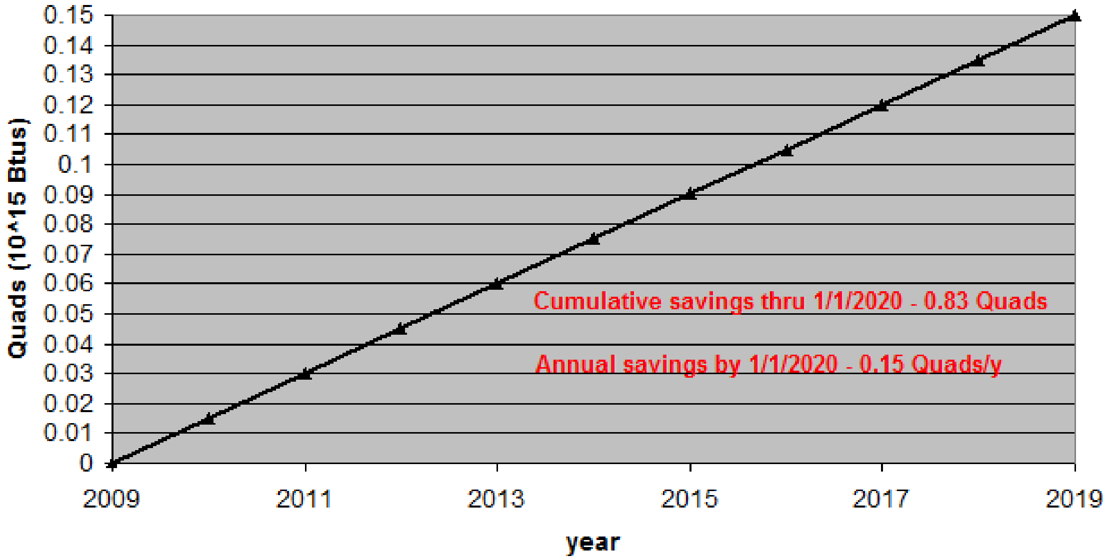

Figure 19. HEWH vs. baseline electric storage WH – projected energy savings by 1/1/2020

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APPENDICES

APPENDIX A – ESTIMATION OF NUMBER OF TEST CYCLES ON ORNL DURABILITY TEST STAND NECESSARY TO SIMULATE 10 YEARS OF NORMAL RESIDENTIAL OPERATION OF THE HEWH UNITS

Acknowledgements – The analysis & calculations summarized in this appendix were compiled by Dr. R. W. Murphy, ORNL (retired) in verification of the reliability test protocol established in May 2008 by General Electric's Jennifer Floyd for the HEWH.

The GE reliability test protocols (see Tables 4 and 5 in main text) were based on an estimate of 1000 water heating cycles per year (~19 per week) for a typical residence using 65 gallons of hot water per day. Reliability goals established by GE for the HEWH are as follows:

- 1-year = 97.4% with a confidence level (CL) of 50%
- 10-year = 73.8% with a CL of 50%

To calculate how many units should be tested and for how long, GE used the Weibull++ Reliability Calculator or Design of Reliability Testing (DRT) feature (www.reliasoft.com/Weibull/index.htm). Since the ORNL reliability test facility has a maximum capacity of 10 units, it was determined that the ten units would need to be run through the test protocol (Tables 4 and 5) a total of 2.5 times (representing 2.5 years normal operation) to test to the 10-year reliability target of 73.8% with zero (0) failures.

ORNL used a Weibull distribution based reliability analyses (http://reliabilityanalyticstoolkit.appspot.com/weibull_distribution) to verify the GE estimate.

Using the cumulative binomial equation (c.f., http://www.weibull.com/hotwire/issue118/relbasics118.htm), the number of test units required (n_{req}) to achieve the 1-year and 10-year reliability (R_{req}) goals was determined to be 27 (26.3 rounded up) and 3 (2.3 rounded up), respectively.

$$n_{req} = \ln(CL)/\ln(R_{req})$$

or

$$n_{req1} = \ln(0.5)/\ln(.974) = 26.3, \text{ and}$$
$$n_{req10} = \ln(0.5)/\ln(.738) = 2.3.$$

The Weibull reliability function R(t) can be used to determine reliability after some time period, t,

$$R(t) = e^{-(t/\eta)^\wedge\beta}$$

where β – Weibull shape factor,
η – Weibull distribution parameter or characteristic time, and
t – time.

Solving the reliability function simultaneously for GE's 1-year and 10-year R_{req}s, the shape factor, β, is determined to be 1.0619, implying a gradually increasing failure rate with time. (An exponential distribution (constant failure rate) would have a β of 1.0, while a Weibull normal distribution would have a β of 3.5.) Given the shape factor, the distribution parameter, η, was determined to be 30.71.

Given the limit of 10 test units (n_{test}) maximum on the ORNL facility, the cumulative binomial equation was used to estimate the associated reliability (R_{10}) at 93.3% for the 50% CL.

$$R_{10} = CL^{(1/\ ntest)} = 0.5^{0.1} = 0.933.$$

Using this reliability level and the β and η values from above, the total test time (number of cycles through the test protocol required, or t_{test}) to be 2.49 years (2.49 times through protocol).

$$\begin{aligned} t_{test} &= \eta * (\ln(1/R_{10})^{1/\beta} \\ &= 30.71 * (\ln(1/.933)^{1/1.0619} \\ &= 2.4868 \end{aligned}$$

This is almost an exact match with GE's prior determination of 2.5 times through the protocol.

As a double check on the above calculation, the test time (t_{test}) was assumed to be exactly 2.5 years (times through the protocol) and the associated 50% CL reliability calculated using the Weibull reliability equation.

$$\begin{aligned} R(t) = R_{10}^{*} &= e^{-(t/\eta)^\wedge\beta} \\ &= e^{-(ttest/\eta)^\wedge\beta} \\ &= e^{-(2.5/30.71)^\wedge 1.0619} \\ &= 0.9327\ (93.27\%) \end{aligned}$$

With this reliability estimate and the specified 50% CL, the number of required test units was calculated to be 9.94, or 10, rounded up. This confirms the first ORNL estimate above and verifies the original estimate developed by GE.

$$\begin{aligned} n_{test}{*} &= \ln(1\text{-}CL)/\ln(R_{10}^{*}) \\ &= \ln(0.5)/\ln(.9327) \\ &= 9.94 \end{aligned}$$

APPENDIX B – INVENTION DISCLOSURES FILED UNDER CRADA WORK PROGRAM

This appendix lists invention disclosures resulting from work done under this CRADA project.

1. Joint disclosures by General Electric and ORNL – none

2. Disclosures by ORNL – none

3. Disclosures by General Electric – none filed or granted during CRADA program. All GE patents related to the GeoSpring HEWH product were either granted or applied for prior to the start of the CRADA.

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