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ANALYSIS OF HIGHLY EFFICIENT ELECTRIC RESIDENTIAL HEAT PUMP WATER HEATERS FOR COLD CLIMATES

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Energy and Transportation Sciences Division

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CONTENTS

ACRONYMS

BPM	Brushless permanent magnet
BTP	DOE Building Technology Program
COP	Coefficient of performance
DB	Dry bulb (temperature)
EER	Energy efficiency ratio
EF	Energy factor
EWT	Entering water temperature
EXV	Electronic expansion valve
GTWH	Gas tankless water heater
GWP	Global warming potential
HBP	High back pressure
HDD	Heating degree days
HPWH	Heat pump water heater
LSHX	Liquid-to-suction line heat exchanger
RH	Relative humidity
RSV	Rated storage volume
TXV	Thermostatic expansion valve
WH	Water heating

ABSTRACT

A scoping-level analysis was conducted to identify electric heat pump water heater (HPWH) concepts that have the potential to achieve or exceed 30% source energy savings compared to a gas tankless water heater (GTWH) representative of the type represented in version 0.9.5.2 beta of the BEopt™ software developed by the National Renewable Energy Laboratory. The analysis was limited to evaluation of options to improve the energy efficiency of electric HPWH product designs currently on the market in the United States. The report first defines the baseline GTWH system and determines its efficiency [source-energy-based "adjusted" or "derated" energy factor (EF) of ~0.71]. High-efficiency components (compressors, pumps, fans, heat exchangers, etc.) were identified and applied to current U.S. HPWH products and analyzed to determine the viability of reaching the target EF. The target site-based EF required for an electric HPWH necessary to provide 30% source energy savings compared to the GTWH baseline unit is then determined to be ~3.19.

EXECUTIVE SUMMARY

This report describes results of a scoping-level analysis of significantly higher efficiency electric heat pump water heaters (HPWH). The specific goal of the work was to identify electric HPWH design options that have the potential to achieve or exceed 30% source energy savings compared to a gas tankless water heater (GTWH) representative of the type represented in version 0.9.5.2 beta of the BEopt™ software developed by the National Renewable Energy Laboratory. This effort was undertaken in response to a clearly expressed target in the FY10 DOE Building Technology Program (BTP) Statement of Needs for the water heating (WH) program element. Specifically, the need expressed was for advanced water heating systems for (buildings in) cold climate locations that achieve 30% source energy savings vs. a GTWH with a cost premium of $2000 or less. For purposes of this analysis a cold climate location is loosely defined as having 5500 heating degree days (HDD) or more.

It should be noted at this point that our analysis was limited to evaluation of options to improve the energy efficiency of electric HPWH product designs currently on the market in the United States. We specifically did NOT consider concepts for HPWH's located in cold ambient conditions (e.g., make use of cold ambient air as a heat source). Rather, the implicit assumption was that the HPWH would be located in a semi-conditioned space in a residence (e.g., basement) with ambient conditions approximating those specified in the energy factor (EF) rating test (67.5°F and 50% RH) as described in Subpart B (Test Procedures), Appendix E (Uniform Test Method for Measuring the Energy Consumption of Water Heaters) of CFR Part 430.

The report first defines the baseline GTWH system and determines its efficiency (source-energy-based "adjusted" or "derated" EF of ~0.71). The target site-based EF required for an electric HPWH necessary to provide 30% source energy savings compared to the GTWH baseline unit is then determined to be ~3.19. High-efficiency components (compressors, pumps, fans, heat exchangers, etc.) were identified and applied to current U.S. HPWH products and analyzed to determine the viability of reaching the target EF. The report concludes with an evaluation of the analysis results against the criteria established for passage to the next development stage in the Stage-Gate Process.

1. NATURAL GAS TANKLESS WATER HEATER (GTWH) BASELINE

The baseline water-heating unit was taken to be the GTWH option described in BEopt™ version 0.8.7 (and unchanged in the recent, subsequent 0.9.5.2 beta version) to find optimal building designs along the path to reduced energy use. According to BEopt™, this option is based on a Takagi Model T-K1 GTWH with an EF of 0.84, derated by 8.8% (giving an "adjusted" or "derated" EF of approximately 0.76) to account for cycling inefficiencies, as proposed for California's 2008 Title 24. However, Takagi specifications for this unit give an EF of 0.81 operating with natural gas (see Appendix A).

The test procedures and computations to determine the EF for "instantaneous" water heaters covered under the Code of Federal Regulations Title 10, Chapter II, Volume 3, Part 430, Subpart B, Appendix E, include a provision that if electrical auxiliary energy (for pumps, fans, etc.) is included in the calculation, it must be converted using the following: 1 kWh = 3,412 Btu. Based on the specifications of the Takagi Model T-K1 operating with natural gas, the electrical power draw was estimated to be about 96.0 W during operation and about 6.2 W during standby. With the performance schedule imposed by the rule for the 24-Hour Simulated Use Test from which the EF is derived, it was estimated that the associated electrical energy consumption of this unit was about 0.177 kWh or 604 Btu, while the associated natural gas energy consumption was about 49,561 Btu. According to the rule, both these quantities are measured on a site basis for EF computation. To convert them to a source basis, the electricity source conversion factors used by BEopt™ (3.16 for electricity and 1.02 for natural gas) were employed to give 1,910 Btu and 50,552 Btu, respectively, for the electricity and natural gas contributions—or a total of 52,462 Btu of source energy. These values, in turn, imply a source energy-based derated EF of about 0.77.

If, consistent with the BEopt™ methodology, the 0.81 EF specified by Takagi for this GTWH is derated in the same manner by 8.8% for cycling inefficiencies (per California 2008 Title 24), the derated site-energy-based EF is approximately 0.74, and the corresponding site energy consumptions are about 0.194 kWh or 663 Btu for electricity and 54,343 Btu for natural gas. Using the same site–source conversion factors gives about 2,094 Btu and 55,430 Btu, respectively for the electricity and natural gas contributions—a total of 57,524 Btu. These values, in turn, imply a corresponding derated source-energy-based EF of about 0.71.

2. TARGET EF FOR ELECTRIC HPWH TO ACHIEVE 30% SOURCE ENERGY SAVINGS

To achieve the goal of 30% saving in source energy (relative to the derated GTWH described above), an electric HPWH would have to use ≤40,267 Btu of source energy for the 24 Hour Simulated Use Test. This yields a source-energy-based EF of about 1.01. The corresponding usage of site energy would be about 12,743 Btu or 3.735 kWh, giving a site-energy-based EF of about 3.19. This represents a 28-60% increase in EF compared to the rated EFs (2.0 to 2.5) of electric HPWH products currently marketed in the United States. It also represents an increase of ~230-240% over the proposed new minimum EF for electric storage water heaters of 55 gallons or less rated storage volume (RSV).

According to the final rule for residential water heaters issued March 22, 2010, the minimum EF for electric storage water heaters effective April 16, 2015, will increase to:

$$EF_{min} = 0.96 - (0.0003 * RSV) \text{ for } RSV \leq 55 \text{ gallons, and}$$

$$EF_{min} = 2.057 - (0.00113 * RSV) \text{ for } RSV > 55 \text{ gallons.}$$

From the first of these two equations, the minimum EF for an electric storage water heater with a rated storage volume of 50 gallons will increase to 0.9450. From the last equation, the minimum EF for an electric storage water heater with a rated storage volume of 80 gallons will increase to 1.9666. Figure 1 compares the new minimum EF standard to the current standard and to the rated EFs of currently marketed electric HPWHs and the 3.19 target EF. One implication of the new EF minimum standard will be that all electric storage WHs with greater than 55 gallons RSV <u>must</u> be of the heat pump type.

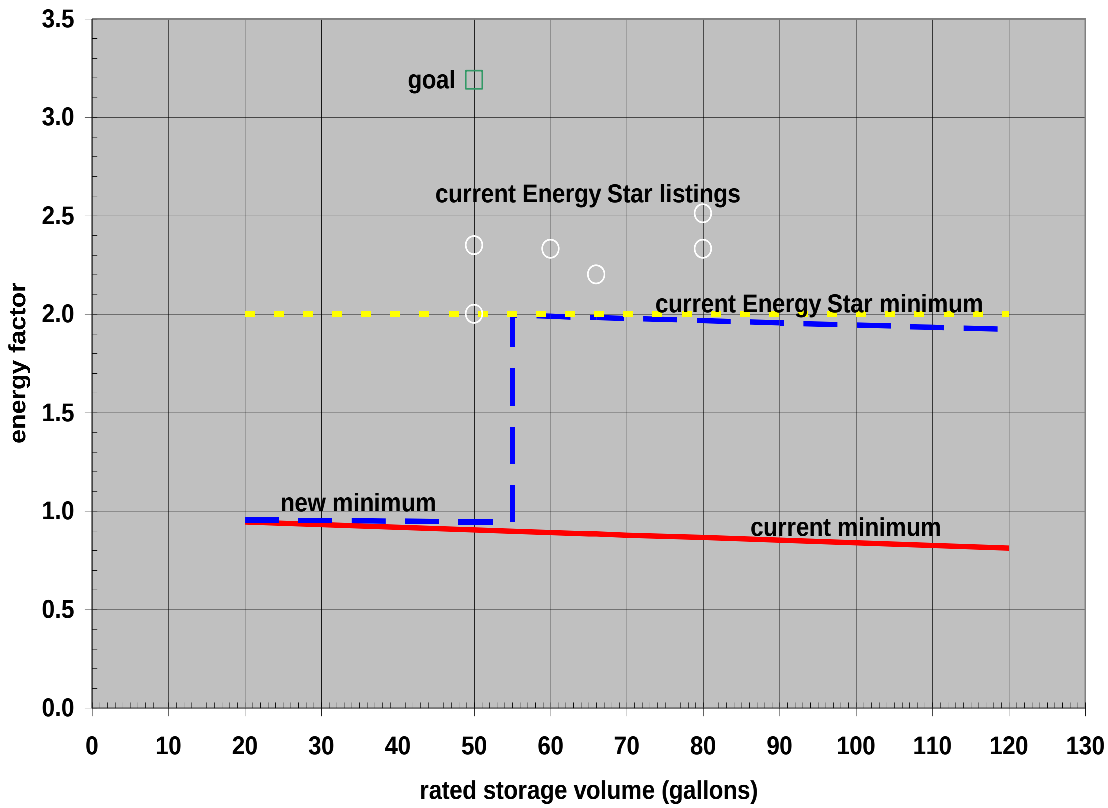

Figure 1. Current and new DOE minimum EF requirements for electric storage water heaters in comparison to the Energy Star minimum requirement, rated EFs of current products, and the target EF required for 30% source savings vs. GTWH baseline ("goal").

3. POTENTIAL FOR ATTAINING THE SOURCE ENERGY SAVINGS GOAL

The ECR International Watter$aver HPWH product (marketed 2002–2005) had an EF of 2.47. It employed a small R-134a reciprocating compressor that had an energy efficiency ratio (EER) rating of about 6.8 and a rated cooling capacity of about 3100 Btu/hr with a 45°F evaporating temperature (20°F exit superheat) and a 130°F condensing temperature (15°F exit subcooling) while operating with refrigerant 134a. These operating conditions are representative of typical surroundings of an HPWH in a cool basement or garage with average tank water at ~115-120°F).

Figure 2 illustrates efficiency levels of the Watter$aver compressor along with a number of other reciprocating models and a few rotary models. The rotary models graphed in Figure 2 have an average EER of ~10.4 at the same operating conditions.

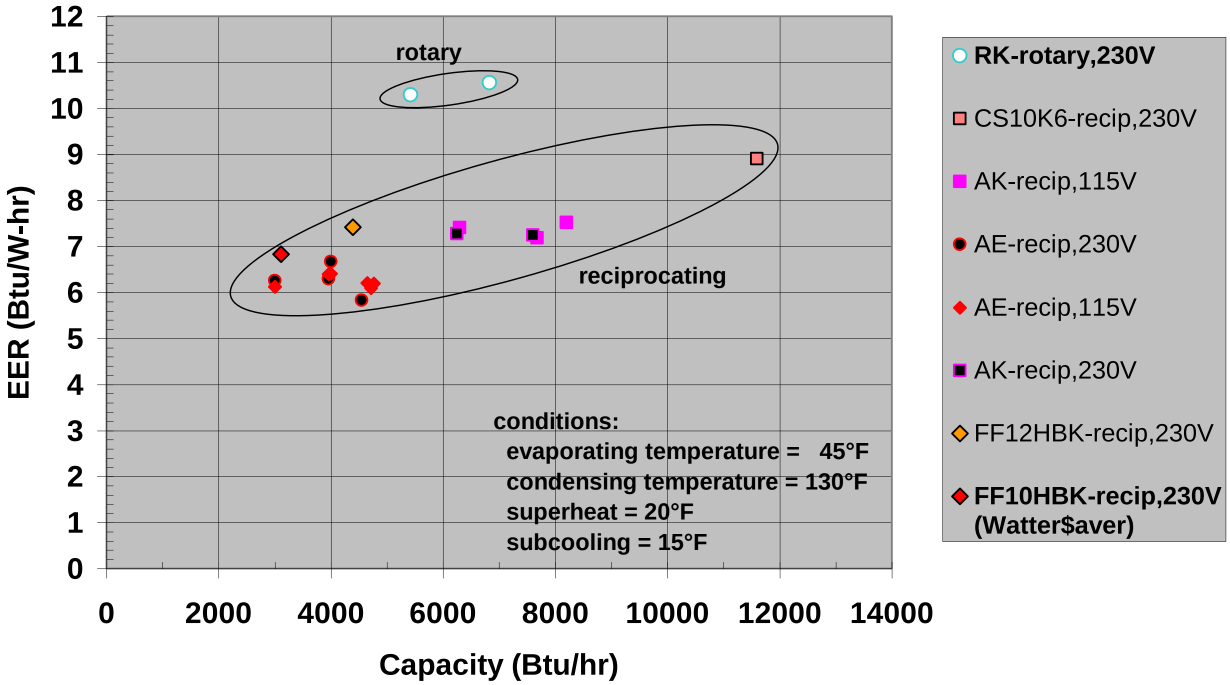

Figure 2. Rated high back pressure (HBP) performance of small R-134a compressors.

Of course, since EER is dimensional (Btu/hr/W) and is a cooling figure of merit, it does not directly translate to the heating situation of interest here. If we convert the EERs to cooling COP, the corresponding values are about 1.99 for the reciprocating compressor and about 3.05 for the rotary compressor. If all the energy input to the evaporator and compressor in each case were delivered as useful heat to the load (water in this case), the effective heating COPs would be about 2.99 and 4.05, respectively. If it is further assumed that the performance of the HPWH (as reflected in the EF) using the rotary compressor would be increased by the same ratio as that of the respective compressor heating COPs, then the resulting EF would be about 3.34. If this estimate proved accurate and if the associated costs were reasonable, then this change alone could produce a viable unit capable of exceeding the EF target of 3.19 outlined above and, through the previously described logic, the source energy savings of 30%. However, in the real system design, other effects such as those relating to compressor capacity and fan and heat exchanger sizing must be taken into account.

4. ENERGY FACTOR PERFORMANCE ANALYSIS FOR NINE HPWH DESIGNS

A more detailed analysis to estimate the potential EFs achievable was undertaken using models of equipment designed to use HFC and CO_2 refrigerants. These were used to develop system performance maps as a function of entering water temperature (EWT) at the DOE HPWH test rating condition of 67.5°F DB, 50% RH. Water heating capacity, total input power, and water flow rates (fixed or variable as required) were then input to TRNSYS models for the 24-hour EF and first-hour rating calculations. The system models were developed for designs using a pump to circulate water from the tank to a water/refrigerant condenser. HPWH designs with static condensers inside or wrapped around the tank were not considered in this analysis.

4.1 GENERAL HPWH EQUIPMENT DESIGN ASSUMPTIONS

The designs evaluated for HFC refrigerants were modeled assuming the use of R-134a and R-410A. Compressor performance maps for high-efficiency rotary compressors over the range of evaporating and condensing temperatures were obtained for appropriate rated cooling capacities ranging from 4,350 to 5,420 Btu/hr.

For the water-to-refrigerant condenser, we used a counterflow Packless double-walled, fluted-tube heat exchanger, model CDAX-6100, a size selected to give a relatively low refrigerant-to-water mean temperature difference of about 9°F for the R-410A case at 115°F EWT. For the pump, we assumed a brushless permanent-magnet motor (BPM) with the pump flow to be optimized for each design for an assumed system head curve.

For the evaporator, we used a cross-counterflow, finned-tube heat exchanger with about 10% more area than in models presently on the market. This sizing gave a mean refrigerant-to-air temperature difference of about 6.5°F for the R-410A case. We assumed 300 cfm airflow across the evaporator with 30 watts of fan power, which also implies a BPM motor.

The assumptions for compressor shell heat loss have a direct bearing on the predicted COPs, as energy lost from the compressor shell in heat does not directly contribute to heating water in the condenser. We obtained calorimeter data from the manufacturer which provided measured discharge temperatures, from which we could calculate shell heat loss levels (as a fraction of compressor input power) over the range of condensing temperatures. These relationships were used in the simulations assuming there was full airflow over the compressor as recommended by the manufacturer. Further, the shell heat losses were assumed to preheat the inlet air to the evaporator. Minimal discharge line heat losses were also assumed.

As tank heat losses are included in EF calculations, we used two levels of assumed tank insulation for this analysis: (1) that required to give a 0.90 EF for a tank heated with electric resistance elements, and (2) that needed for a 0.95 EF resistance element tank.

4.2 HPWH OPERATION/CONTROL ASSUMPTIONS

The primary HPWH designs using HFCs were optimized for heating COP at an assumed average EWT of 115°F and 67°F DB, 50% RH inlet air conditions. The condenser subcooling and water flow were optimized for a BPM pump power versus flow curve developed for an expected system loop head curve. Optimum water temperature increases through the condenser of 5.6 to 6.6°F were found for the smaller to larger capacity units, respectively, at 115°F EWT. Accordingly, tank heat-up was accomplished in a stepwise manner for this design configuration.

Once the optimum water flow and subcooling were determined for an average EWT condition, the flow rate and required refrigerant charge levels were fixed. The compressor inlet superheat was assumed to be fixed in this analysis, as would be approximately the case for a system with TXV or similar EXV control. With the charge level and inlet superheat fixed (for an implicit TXV model), the design model was run for EWTs ranging from 55 to 135°F to generate the HPWH performance map needed for a TRNSYS 24-hour energy factor simulation.

For the transcritical CO_2 design, we adjusted the heat exchanger sizes to obtain the same refrigerant-to-air and refrigerant-to-water mean temperature differences as seen for the R-410A case (which were the lowest of the HFC cases). The pump and fan power and airflow assumptions were consistent with those for the HFC cases. A fixed-orifice refrigerant flow control was assumed which was sized for best performance at the mean EWT. For the compressor, we used a relatively high-efficiency reciprocating model for which we could obtain a performance map comparable to those for the HFC refrigerant compressors. For the CO_2 compressor, we assumed a fixed average shell heat loss fraction based on available system test data, a value which was larger than that found in the calorimeter tests for HFC compressors.

A once-through design for the water flow was assumed for the primary CO_2 HPWH systems to obtain best matching of the water and refrigerant temperature glides in the gas cooler. A fixed 140°F return water temperature was maintained by adjusting the pump flow and power assuming a BPM pump. For confirmation that the once-through design was more efficient than a stepwise heat-up approach for a CO_2 HPWH, one stepwise CO_2 design was also simulated with an optimal flow rate set to give 6.6°F delta-T at 115°F EWT. (For similar comparison reasons, two R-134a cases were also simulated for once-through designs, as discussed later in this section.)

Once system performance maps versus EWT were generated for the selected cases, these data sets along with the water flow rates were input to a TRNSYS simulation of the 24-hour EF test. This simulation included a nominal 50 gallon water tank model (actual assumed capacity of 45 gallons), divided into 6 equal volume regions from top (region 1) to bottom (region 6). For the stepwise heat-up cases, the water was removed from node 6 and returned to node 5 with small temperature rises on each pass, while for the once-through cases, the water was removed from node 6, heated to 140°F in one pass, and returned to node 1.

The primary HPWH control locations were at node 5 for all the stepwise heat-up cases and node 2 for the once-through designs. These nodes correspond approximately to the locations of the lower and upper resistance elements and their associated thermostats in electric resistance water heaters. For the once-through case a control location at node 2 rather than at node 5 was found to give higher EF performance by providing more tank stratification during tank reheat operations, while still maintaining higher top tank temperatures reasonably well. The HPWH thermostat had a ±5°F deadband and was adjusted so that at the end of tank pre-heat runs, the average tank temperature was 135°F. For the stepwise designs, this was typically a setting of near 125°F on, and 135°F off. For the once-through designs, a higher HPWH thermostat setting was needed to give an average tank starting temperature of 135°F with the larger tank stratification. An upper resistance element was included for all the designs (located at node 2) with an assumed ±11°F deadband and a setting of 108.5°F on and 130.5°F off.

4.3 SUMMARY OF REFRIGERANT, COMPRESSOR, AND SYSTEM DESIGN ASSUMPTIONS AND PERFORMANCE

A summary of the compressor types, efficiencies, sizes, and nominal cooling capacities used in the analysis for the nine design cases is shown in Table 1. This is followed for each case by the design configuration and control assumptions. Next the HPWH-on-time-averaged EWTs and corresponding heating capacities and COPs are given. The final entries are the calculated EFs for the two different tank heat loss levels. Note that the EF calculations used the HPWH system performance maps for each case with the EWTs varying from 55 to 135°F at the fixed 67.5°F DB, 50% RH inlet air condition of the DOE EF test. The first four cases are for a stepwise tank heat-up design, with a once-through heat-up design for the remaining five.

The first case listed is the larger of the two R-134a compressor sizes considered, with the highest rated compressor EER of 10.3. The second entry is the smallest high-efficiency rotary model for which a performance map was available to us. This unit as built has a lower-efficiency CSIR (capacitor-start, induction-run) motor; however, the manufacturer indicated that if this motor were replaced with a higher-efficiency PSC (permanent-split capacitor) motor, the rated efficiency could be increased to near that of the 10.3 EER model. Based on this information, we assumed a 10 EER could be achieved with this change.

The third case is for the smallest high-efficiency R-410A rotary available by the same manufacturer. This model has a somewhat lower cooling EER due to differing refrigerant thermodynamic properties. A better comparison of compressor-only performance is obtained by calculating the overall isentropic compressor efficiency from shell inlet to exit, as shown in the column following the rated EER values. By this measure, the compressor efficiency is 58% compared to 60 and 61% for the two R-134a compressors.

Table 1. Summary of compressor and system performance predictions for nine HPWH designs with high-efficiency components

		Compressor Specifications					Operation/Control Approach			Ave. Heating Performance			Energy Factors [4]	
Case	**Refrigerant**	**Type**	**EER [1,2]**	**Overall Isen.**	**Displ.**	**Q_c [1,2]**	**Heatup**	**Return**	**Control**	**EWT_ave**	**Q_h [3]**	**COP_h [3]**	**$EF_{0.90}$**	**$EF_{0.95}$**
			(Btu/W-hr)	**Efficiency**	(in^3)	(Btu/hr)	**Method**	**Node**	**Node**	(F)	(Btu/hr)			
1	**R-134a**	Rotary	10.3 [1]	0.61	0.697	5420 [1]	Stepwise	5	5	114.6	6810	3.43	**3.02**	**3.18**
2	**R-134a**	Rotary	10 [1]	0.60	0.580	4350 [1]	Stepwise	5	5	111.4	5865	3.63	**3.17**	**3.33**
3	**R-410A**	Rotary	8,7 [1]	0.58	0.330	4850 [1]	Stepwise	5	5	114.3	6560	3.22	**2.83**	**2.98**
4	**CO_2**	Recip	9.3 [2]	0.52	0.107	4890 [2]	Stepwise	5	5	110.5	5340	2.14	**1.84**	**1.94**
5	**CO_2**	Recip	9.3 [2]	0.52	0.107	4890 [2]	Once-Thru	1	5	91.9	6200	2.62	**2.23**	**2.35**
6	**CO_2**	Recip	9.3 [2]	0.52	0.107	4890 [2]	Once-Thru	1	2	82.9	6655	2.86	**2.51**	**2.63**
7	**CO_2 w LSHX[5]**	Recip	9.3 [2]	0.52	0.107	4890 [2]	Once-Thru	1	2	82.9	6470	3.01	**2.61**	**2.75**
8	**R-134a[6]**	Rotary	10 [1]	0.60	0.580	4350 [1]	Once-Thru	1	2	78	5380	2.55	**2.12**	**2.22**
9	**R-134a**	Rotary	10.3 [1]	0.61	0.697	5420 [1]	Once-Thru	1	2	80	6485	2.7	**2.42**	**2.54**

[1] At Te/Tc/SH/SC rating condition of 45/130/20/15 F

[2] At Te/Pc/RG/LL rating condition of 45F/85bar/90F/90F (ASHRAE HBP32)

[3] At approximate average HPWH operation conditions for EF Test at DB=67.5F, RH=50% inlet air

[4] Predicted based on performance maps from 55F to 135F EWTs, TRNSYS tank model with t-stat settings consistent with DOE 24-hour EF test procedure

[5] Idealized LSHX heating suction gas to 90F where possible, depending on EWT.

[6] Upper element was activated during EF draws for the 108.5F on, 130.5F off t-stat set point at node 2 (as used for all cases), lowering the effective EF for this case.

Note that all analyses use actual compressor maps except for the 10 EER R-134a case which uses an adjusted map with the EER scaled from 9.1 to 10 EER to reflect expected gains in switching from a lower efficiency CSIR to a higher-efficiency PSC motor.

The fourth case shown is for a reciprocating CO_2 compressor. Here the rated EER, as noted in the footnotes, is by necessity based on a different set of test conditions for the transcritical compression cycle. Again, the overall isentropic efficiency calculated at the rating point gives a better measure of the relative compressor performance. (No rotary CO_2 compressor performance maps could be obtained for this analysis.)

The column after the compressor efficiencies shows the compressor displacements, which decrease as the operating evaporating pressures increase from R-134a to R-410A to CO_2. The rated cooling capacities for the two R-134a cases bracket the rated capacities for the R-410A and CO_2 units. The predicted heating capacities and COPs for the first four cases at their respective average EWTs show similar capacities for the HFC cases and a slightly lower average capacity for the CO_2 case.

4.4 HPWH EF PREDICTIONS FOR STEPWISE HEAT-UP DESIGNS

The resultant EFs for the first two R-134a cases show values greater than 3 for both tank heat loss assumptions. The smaller capacity R-134a case is the only one predicted to have an EF greater than the 3.19 target, although two other R-134a configurations (the smaller compressor with less insulated tank and the larger compressor with better insulated tank) are very close. These cases would exceed the target if the shell heat loss fractions were moderately lower than the assumed values (which were based on 425 cfm of air over the compressor).

As the assumed airflow is 300 cfm for these cases and most likely all of this airflow will not flow over the compressor, it is likely that the applied airflow across the compressor could be 50% or less of the tested levels. However, since we did not have data to predict these lowered levels, we did not attempt such an analysis at this time. If the shell heat loss fractions (which ranged from about 20% at 55°F EWT to 40% at 135°F EWT for the HFC cases) were reduced by one half or more, this could also move the predicted EF values that are presently around 3.0 up close to the target level. Further calorimeter tests with reduced airflow over the compressor are recommended to allow these effects to be better quantified.

The smaller capacity R-134a case has about 5% higher EFs than the larger R-134a case. Three factors contribute to this result. First, the steady-state COP at the same EWT is slightly higher (1.5%) for the smaller unit due to lower loading on the equally sized heat exchangers, even though the compressor efficiency is slightly lower. The other factors are related to lower water temperatures during the draw and heat-up cycles. Figures 3 and 4 show the predicted top and bottom tank temperatures versus time for the 24-hour tests for the larger and smaller R-134a compressors, respectively. The smaller R-134a unit in Figure 4 has a lower average EWT, as seen from Table 1, which increases the operating COP. This is because the smaller capacity unit cannot maintain tank temperatures as high during the 6 water draws in the initial part of the 24-hour EF test. Finally, the lower tank temperatures result in less tank heat loss, which in turn also results in a higher EF.

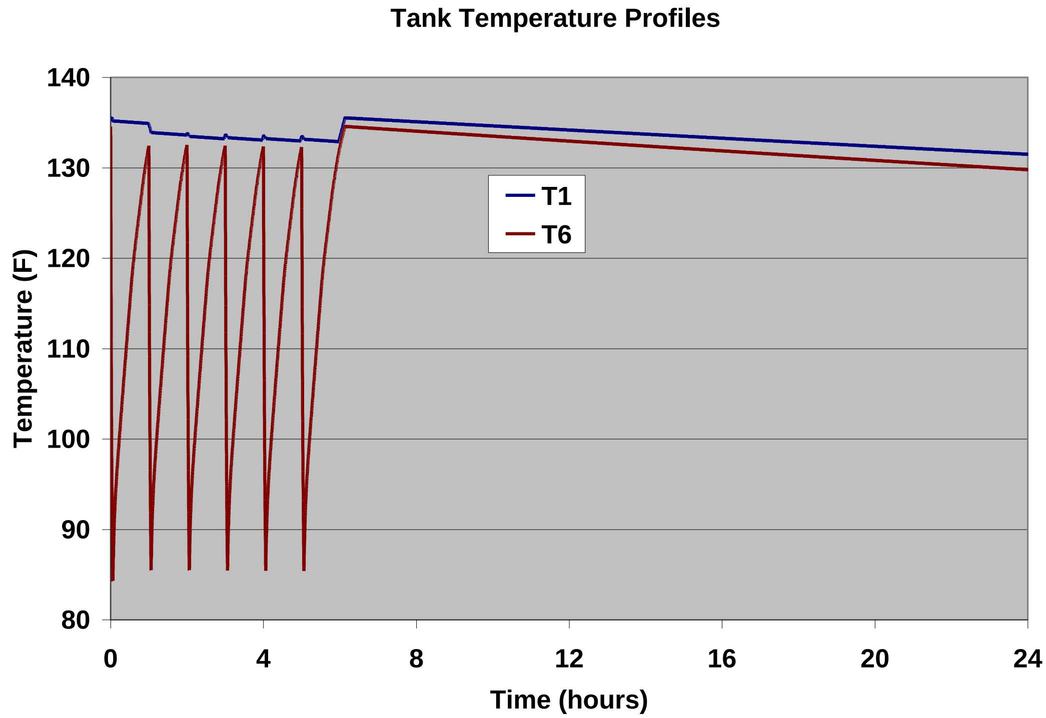

Figure 3. Predicted top and bottom tank temperature profiles over the EF test simulation for the larger capacity R-134a system with 0.95 EF tank.

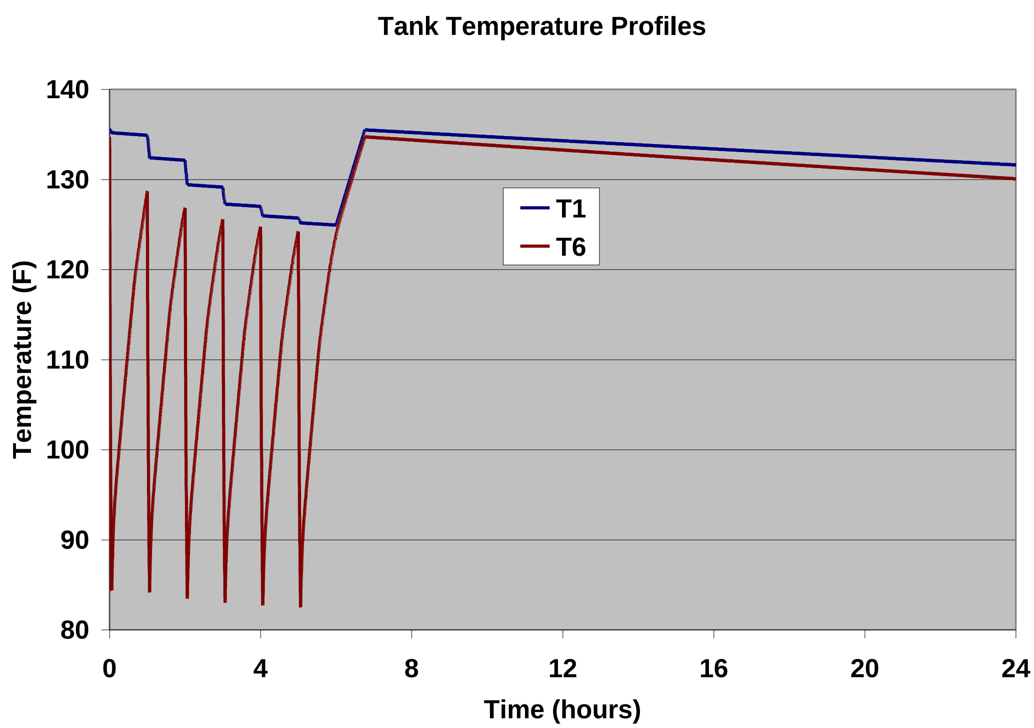

Figure 4. Predicted top and bottom tank temperature profiles over the EF test simulation for the smaller capacity R-134a system with 0.95 EF tank.

The on and off settings assumed for the upper electric element thermostat (located at node 2) for all the EF and the first-hour rating tests were such that no resistance element energy was used in the EF test for any of the stepwise cases examined. Predicted first-hour ratings were > 64 gallons in all cases.

4.5 HPWH EF PREDICTIONS FOR ONCE-THROUGH HEAT-UP DESIGNS

With regard to the CO_2 HPWH with a stepwise heat-up design in case 4, the predicted EF performance was substantially lower than for the HFC cases. In case 5, the same equipment is controlled in a once-through design which yields a much lower average operating EWT than in case 4 and thus a higher EF. For this case, the control node was left at node 5, which reduced the temperature stratification during the heat-up cycles.

In case 6, the control point was moved to node 2, which maintained more tank stratification. In combination with the glide matching between the CO_2 and the water in the gas cooler with the larger water delta-T's, the EFs for case 6 are 36% higher than in case 4. Even so, the EFs for the CO_2 HPWH design of case 6 are 21% lower than for the highest performing R-134a case. About 12 percentage points of this can be attributed to the lower compressor efficiency of the available reciprocating model. Some further loss may be attributable to the fixed-orifice refrigerant flow control assumption used for this analysis as compared with a variable refrigerant flow control (TXV or EXV) for the HFC cases.

Next we added a liquid-to-suction line heat exchanger (LSHX) in the CO_2 cycle analysis, which can be beneficial to transcritical cycle performance. We assumed an idealized LSHX where the suction temperature was heated where possible to 90°F (the limiting suction inlet temperature for the compressor), depending on the EWT. This increased the EFs 4 to 4.5%, leaving them still about 18% lower than the best R-134a case. <u>It appears that all of these improvements would be needed along with a lower compressor shell heat loss for CO_2 HPWHs to match the EF performance potential of R-134a or its low GWP alternative R-1234yf (which is expected to have an EF similar to that of R-134a).</u> One possible advantage of the once-through CO_2 design could be hotter water temperatures provided more quickly to the top of the tank. As shown in Figure 5, the top tank temperatures in the EF test for the CO_2 system are somewhat higher than for the highest performing R-134a system in Figure 4 during the draw periods. However, the average of these higher top tank temperatures in Figure 5 is about the same as that for the higher capacity R-134a case in Figure 3, which has EFs more than 15.6% higher.

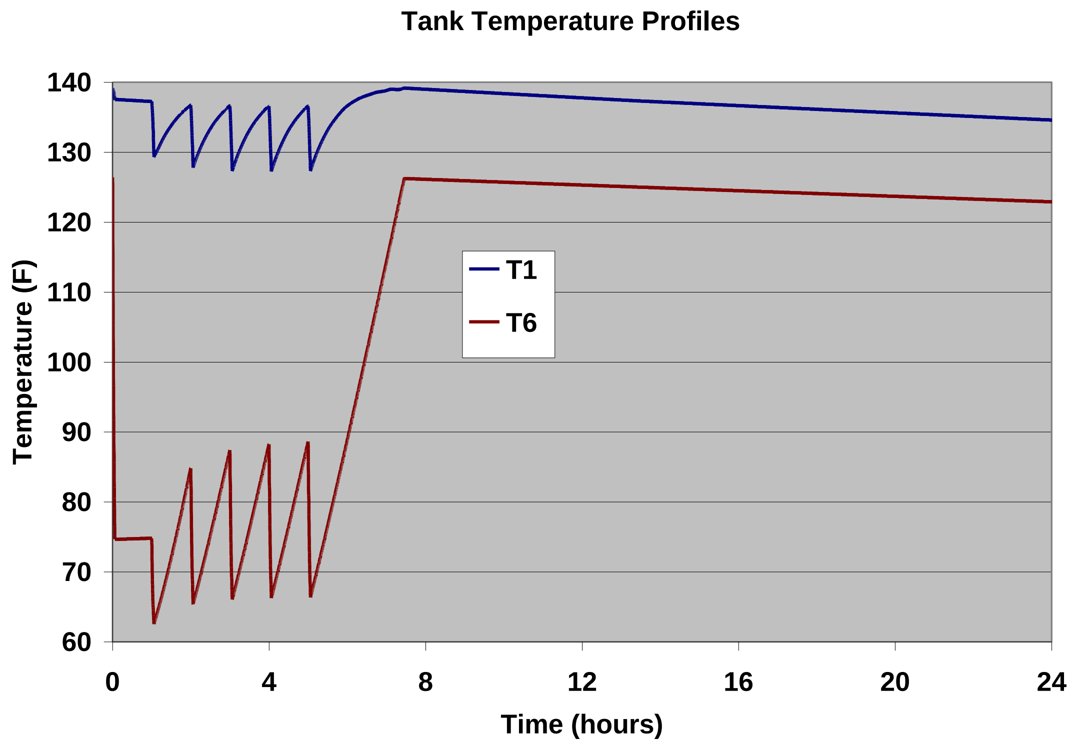

Figure 5. Predicted top and bottom tank temperature profiles over the EF test simulation for the CO_2 system with LSHX with 0.95 EF tank.

In the last two cases shown in Table 1, the two original R-134a design setups are re-configured for once-through operation by lowering the water flow rates and changing the return node and control locations. These two CO_2-design-equivalent setups show that conventional HPWH refrigerants operating at constant condensing temperature conditions lose considerable performance in a once-through setup. Note that the higher performing, lower capacity R-134a unit in a stepwise configuration is now the lowest efficiency case because the upper element was activated due to insufficient heating capacity during the EF test draws. The larger R-134a unit in once-through mode performs almost as well as the non-LSHX CO_2 case (case 5), but with the compressor operating at elevated condensing temperatures throughout the EF test. Predicted first-hour ratings were again > 64 gallons in all cases since this value is primarily determined by the upper element size, location, and control settings.

4.6 POSSIBLE FURTHER ANALYSIS WORK

For both the HFC and CO_2 systems, the compressors were assumed to be single speed. Higher efficiencies could likely be obtained with variable-speed compressors but no data were available on the performance or operating envelopes of such compressors. Variable-speed compressors provide the opportunity to slow down the compressor speed when the tank is approaching the set point temperatures, thus unloading the heat exchangers and reducing the pressure ratios during the lowest efficiency part of the heat-up cycle. This could be an area of further investigation and additional performance gains — gains which would need to be justified in energy savings and/or improved hot water delivery, relative to the added cost.

Lastly it should be noted that tank models used to predict temperature distributions in the water tanks could be improved by more validation for EF test conditions with forced external flow of both fixed and varying flow rates with water return nodes at the top or near the bottom of the tank. The absolute accuracy of the EF predictions depends on good prediction of the average EWTs seen by the HPWHs during heat-up operation. Further lab testing and validation work are recommended to refine the mixing assumptions used in the TRNSYS analysis for specific configurations.

5. PRELIMINARY ESTIMATE OF NATIONAL ENERGY IMPACTS FROM TARGET HPWH TECHNOLOGY

5.1 COMPARED TO THE GTWH BASELINE

According to Table HC9.8 of the 2005 Residential Energy Consumption Survey (RECS 2005, http://www.eia.doe.gov/emeu/recs/recs2005/hc2005_tables/hc8waterheating/pdf/tablehc9.8.pdf) about 33% of all U.S. housing units were located in cold climate areas (defined as having 5500 HDD or more for purposes of this study), and 60% of those used natural gas as the primary water heating fuel. Data from RECS 2005, Table WH4 (http://www.eia.doe.gov/emeu/recs/recs2005/c&e/waterheating/pdf/tablewh4.pdf) indicate that some 36% of total U.S. natural gas consumption for water heating can be assigned to those residences, or a total of ~0.39 quads based on data in the 2009 update of the DOE Building Energy DataBook (BED 2009, http://buildingsdatabook.eere.energy.gov/). The 2005 Guide for Evaluation of Energy Savings Potential indicates that the average EF of gas water heaters to be used as the base for energy savings computation in the U.S. is 0.59. Based on these data, if all gas water heaters in cold climate residences switched to GTWH with an EF of 0.81 (e.g., 100% market penetration of GTWH technology), total annual WH energy consumption for those regions of the U.S. would drop to ~0.284 quads (0.39 x 0.59/0.81). If these were all replaced by electric HPWHs meeting the target EF criteria (e.g., 30% source energy savings vs. GTWH baseline), total annual WH energy consumption would further drop to~0.199 quads (0.284x70%) for a maximum national energy savings of 0.085 quads (0.284 – 0.199). As noted, this is maximum savings potential assuming eventual 100% market penetration of the advanced electric HPWH technology.

5.2 COMPARED TO THE CURRENT ELECTRIC STORAGE WH STOCK

Table HC9.8 of RECS 2005 indicates that ~27% of residences in cold locations use electricity as the primary WH energy source. Table WH4 indicates that these residences consume ~26% of total annual U.S. water heating electricity or ~ 0.35 quads. Using the baseline electric WH EF of 0.9 from the 2005 Guide for Evaluation of Energy Savings Potential, maximum annual energy savings from 100% replacement of electric WH stock with HPWHs of 3.19 EF would be ~0.251 quads [0.35 x(1-0.9/3.19)]. As noted, this assumes 100% replacement of current technology with the advanced target HPWH. Depending upon market forces alone, this is obviously optimistic — reality would suggest maximum penetration closer to perhaps 30% (~0.075 quad savings). But if future DOE rulemaking results in requiring HPWH technology for storage WH of 40 gallons RSV or higher, then ultimately almost all electric WH in U.S. residences would be replaced by HPWH technology. If all are replaced by HPWHs with average EF of 2.2 (assumed typical performance of current HPWH products), total annual energy savings would amount to ~0.207 quads. Incremental annual savings from use of the advanced 3.19 EF technology would amount to 0.044 quads.

6. QUALITATIVE DISCUSSION OF OTHER FACTORS AFFECTING "AS-INSTALLED" ANNUAL PERFORMANCE IN A RESIDENCE

The EF analysis above is useful for comparison of "rated" performance of the target electric HPWH to existing HPWHs and the GTWH baseline. However, detailed analytical comparison of "as-installed" annual performance would require an hourly or sub-hourly annual performance simulation taking into account several other application factors (beyond the available resources of this initial scoping project). One of these is the actual ambient conditions surrounding the unit, which depend on its location in the residence. Another factor is the unit's impact on house space heating and cooling loads if installed in conditioned space. A basement location is likely to have somewhat cooler ambient temperature than that specified in the EF rating test (67.5°F), which would tend to degrade the HPWH performance relative to the GTWH baseline. However, offsetting that effect, the tank thermostat setting is likely to be 10-15°F lower than that used in the EF test (135°F), leading to higher performance of the HPWH relative to the GTWH baseline.

On a first order estimate we assume that these effects would approximately negate one another, i.e., no significant change in the energy savings estimate. Location of the HPWH within the conditioned space would provide the HPWH with 3.5 to 8.5°F warmer ambient source air (assuming 71°F and 76°F winter and summer thermostat settings) than would a basement location with concomitant improvement of its water heating performance, particularly in summer and shoulder periods, relative to the GTWH baseline. This improvement (added to that due to lower tank thermostat setting as described above) would be offset to some extent by the increase in house space heating loads due to the HPWH operation in winter.

We have done some analysis to obtain a rough estimate of this impact for a Chicago, IL, location. For a 2600 ft^2, two-story house in that location the space heating load was estimated to increase by ~9% and the space cooling load to decrease by ~26% with a small (2-3%) net increase in annual space conditioning load, assuming that the air in the room with the HPWH is well mixed with air from the rest of the house. For more northerly locations the net load increase will be larger. As noted, a detailed annual simulation would be required to estimate the range of net source energy consumption impacts of locating the target HPWH in the conditioned space, and such a simulation is recommended going forward. It is assumed on a first order that the impact would be small, perhaps insignificant, in Chicago. It should also be noted that the added infiltration from the operation of the flue vent fan in the case of a tankless gas water heater would likely increase space conditioning loads in both winter and summer months. For the baseline GTWH, the vent fan airflow is estimated to be approximately 35 cfm whenever the unit is operating. It should also be noted that cold climates have the lowest ground water temperatures throughout the year. As such, this results in higher water heating loads in these climates and a more beneficial performance from HPWHs due to the cooler EWTs to the unit when meeting active house draws.

These various effects could be quantified to a large extent in a detailed annual performance TRNSYS analysis comparing advanced HPWHs to GTWHs.

7. COMPARISON OF TARGET EF HPWH TO CURRENT RESIDENTIAL ELECTRIC HPWH PRODUCTS

Electric HPWHs available in the United States as of August, 2010, include at least 28 models under 16 brands from 6 different manufacturers. Of these, 23 models under 14 brands from 5 manufacturers were listed as being Energy Star-qualified. That is, each was of integral (or "drop-in”) configuration with voltage ≤ 250 V, current ≤ 24 A, EF ≥ 2.0, first-hour rating ≥ 50 gallons, warranty ≥ 6 years on the sealed system, and meeting UL 174 and UL 1995 safety standards. Listed values for their associated EFs ranged from 2.00 to 2.51. First-hour ratings ranged from 60 to 84 gallons and rated storage volumes ranged from 50 to 80 gallons.

The target EF of 3.19 is more than 27% above the highest Energy Star listing (2.51 for an 80-gallon RSV unit). When/if an HPWH product meeting the target is available, it would use about 21% less source energy than 2.51 EF units. If we choose the most widely used RSV for residential electric water heaters (50 gallons) for comparison, the highest Energy Star-listed EF is 2.35. If the target is met for this size, then the target EF is almost 36% higher and the source energy use would be about 26% lower.

At the 50-gallon size, the target represents a 237% increase in EF relative to the new minimum standard (EF = 0.945) and a 253% increase relative to the current minimum standard (EF = 0.90). The corresponding reductions in source energy consumption are 70% and 72%, respectively.

8. COMPARISON OF TARGET EF HPWH TO JAPANESE ECOCUTE HPWH PRODUCTS

EcoCute HPWH systems employ CO_2 as the working fluid and benefit from significant government subsidies in Japan. Most such systems are of the split arrangement with separate tanks that store large amounts (much more than typical daily usage of U.S. residences) of high temperature water, with tank charging occurring primarily during the late night hours to take advantage of low off-peak electricity prices. For these units, energy efficiency has been characterized by an annual performance factor (APF) based on the Japanese Refrigeration and Air Conditioning Industry Association Standard JRA 4050 (2007R and a subsequent annex). This value is based on a number of steady-state tests with the heat pump operating at selected ambient air (dry bulb and wet bulb) and water (inlet and exit) temperatures and appears to be specifically tailored to Japanese operating conditions. It does not account for transient performance and tank standby losses and therefore cannot be directly compared to the water heater EF rating employed in the United States. Furthermore, the cost premium of EcoCute units relative to current Energy Star-listed products appears to be prohibitive.

9. EVALUATION AGAINST GATE 2 GATE PERFORMANCE CRITERIA — *HIGHLY EFFICIENT ELECTRIC HPWH FOR APPLICATION TO COLD CLIMATES* PROJECT

As called for in the statement of work for the *Highly Efficient Electric HPWH for Cold Climates* project, recommendations are provided here for the stage-gate performance energy savings criteria for an advanced electric HPWH for application to cold climate residences for passage from the project’s current stage, Applied Research, to the next stage, Exploratory Development. For the sake of completeness and consistency, the entire set of stage-gate criteria is summarized. Our recommendations are guided by the Building Technologies Program’s (BTP’s) *Stage-Gate Implementation Handbook for R&D Projects*, June 2009, Version 1.0.

Including the required, corporate must-meet criteria (on page 33 of the stage-gate handbook), the following are our recommendations for the full set of gate criteria.

Corporate Must-Meet Criteria:

1. There is a unique federal role that is clearly defined and there is the potential to partner with other organizations in the technology's development. ***Response:*** *It is considered that absent a Federal RD&D involvement, a push to much higher electric HPWH efficiencies as would be needed for this development to succeed is unlikely. Soliciting specific manufacturer partner(s) was not within the scope of this initial analysis project. But we have good relationships with current U.S. WH industry players and these will be approached to join us in the effort should the project pass the Gate 2 review and go forward.*
2. The intended customers and the technology's competitive advantages for those customers have been identified. ***Response:*** *Analysis of housing and residential energy consumption data shows that about one third of all U.S. housing units are located in cold climate regions where the advanced HPWH technology would be of significant energy savings benefit to the nation (estimated total annual savings of ~0.13 quads with 100% market penetration) and these consumers. A detailed business case and payback assessment was beyond the scope of this current project, but the advanced HPWH technologies examined are expected to be similar in configuration to current integral HPWH products that carry retail prices in the $1400-$1600 range. A first order estimate of retail price for the advanced system (assuming similar production quantities and similar but more efficient components) is 20-30% over that of current HPWHs, or a range of $1700 to $2100. Taking the median of that range and assuming installation costs are $450, the installed cost is estimated at $2350. By comparison, retail prices quoted for residential GTWH products vary widely, ranging from under $1000 to over$3000 based on several sources we checked. The installed cost for the GTWH used by BEopt™ is $1182. Based on that figure, the estimated cost premium of the advanced electric HPWH is well within the $2000 limit desired.*
3. The technology has the potential for significant energy savings compared to alternatives for the targeted application. ***Response:*** *National annual energy savings estimates range from ~0.13 quads (with 100% replacement of both gas and electric WH stock in cold climate areas of interest) to ~0.07 quads (30% replacement of gas WH stock and 100% replacement of electric WH stock assuming future minimum EF regulatory actions requiring heat pump technology for almost all residential electric WH applications).*
4. There is reasonable likelihood that the result can be achieved within the time frames required to meet BTP goals. ***Response:*** *Should be met assuming that an aggressive CRADA partnership with a committed manufacturer partner can be arranged.*

Project-Specific Should-Meet Criteria:

The BTP Statement of Needs (SON) for the HVAC/WH program element specifically expressed a need for advanced water heating systems for cold climate locations that achieve 30% source energy savings vs. a GTWH baseline system. This suggests the following should-meet criterion:

1. The projected energy savings should closely approach or preferably exceed 30% source energy savings compared to the baseline system, a GTWH configuration as implemented into the National Renewable Energy Laboratory BEopt™ model version 0.8.7 as specified in the project SOW. ***Response:*** *Project analyses reasonably prove the technical feasibility of achieving an electric HPWH with a target EF of at least 3.19, the minimum needed to meet the energy savings criteria.*
2. The value of these savings should outweigh the projected additional first cost, if any, of the HPWH over the baseline GTWH. ***Response:*** *Compared to the incremental cost target specified in the DOE/BTP FY10 SON ($2000), the incremental costs we have been able to preliminarily estimate for the advanced electric HPWH option vs. GTWHs are*

much lower (~$1168 compared to the GTWH unit cost of $1182 used by BEopt™). So from this standpoint, this criterion has definitely been met.

Viewed from the consumer economics standpoint, the results are mixed. As noted above, a detailed business case and consumer payback analysis was beyond the scope of the project. However, a preliminary payback estimate of an advanced HPWH (3.19 EF) over the baseline 0.81 EF GTWH (effective site EF=0.71) was made for a Chicago location (with estimated WH load of 3850 kWh or 13.14 million Btu) using recent IL natural gas and electricity prices available from EIA (July 2011 Natural Gas Monthly, Table 18, http://www.eia.gov/pub/oil_gas/natural_gas/data_publications/natural_gas_monthly/current/pdf/table_18.pdf; and August 2011 Electric Power Monthly Table 5.6.B, http://www.eia.gov/cneaf/electricity/epm/chap5.pdf, 2011 year to date average price data through May; \$7.79/1000 ft^3 for gas and 11.46 ¢/kWh for electricity; publications accessed August 25, 2011). Based on this information the estimated annual energy cost savings from switching to the advanced HPWH is about \$2. Using the preliminary estimates of the HPWH average installed cost (\$2350) and the GTWH installed cost used in BEopt™, this yields a simple payback period of >500 years. Similar estimates for Boston and Syracuse (with same annual WH load, gas prices of \$13.26/1000 ft^3 (est.) and \$12.69/1000 ft^3 , and electricity prices of 14.73 ¢/kWh and 17.64 ¢/kWh , respectively) gave annual cost savings and payback periods of about \$61 and \$17 and ~19 and ~70 years, respectively. Absent some utility (or other) incentive program to reduce cost of the target EF HPWH to the consumer the likelihood is that this technology will not significantly displace GTWHs where they are an option (e.g., natural gas is available to the home owner). It should be noted that estimated consumer payback for the target 3.19 EF HPWH vs. a 2.2 EF HPWH in these same three cities is ~4-6 years, so the advanced electric HPWH is a more attractive option where natural gas is not available.

One final point: As noted, the advanced electric HPWH technology options evaluated under this project are largely extensions of existing HPWH product configurations. As such, they should be equally applicable to both retrofit and new construction markets.

10. ORNL RECOMMENDATION

Based on the above, ORNL recommends that the project pass the Gate 2 review and proceed to Stage 2, Exploratory Development, at least through lab proof of concept breadboard testing. If no private sector partner can be interested in collaborating with us to take the development on to the advanced Stages, it should probably go no further than Stage 2.

Appendix A
Takagi TK1 GTWH

Takagi TK1 Tankless Gas Water Heater

Posted by **Richard**
February 21, 2010

Takagi TK1 Tankless Water Heater

Most efficient water heater:

- Electronic Ignition, No Standing Pilot Light
- Continuous Hot Water
- Saves Space
- Lowers Fuel Costs
- Computer Controls
- Power Vented
- Indoor & Outdoor Models

The Takagi Flash TK1 is an on demand tankless gas water heater. This space saving,
highly energy efficient water heater can deliver over 200 gallons of hot water
every hour and since there is no tank to run out, it supplies Hot Water endlessly.

Since there is no storage tank to keep heated all day and no pilot light, the
Takagi TK1 Tankless Water Heater only burns gas when you need hot water. This eliminates standby heat loss
which can be as high as 3 – 4 % every hour for storage tank type water heater.

The Takagi Flash TK1 can save as much as fifty percent on your fuel cost.

The Takagi TK1 Gas Water Heater with its convenient design and compact size is made to go anywhere you need hot water. The Takagi TK1 Hot Water Heater can free up valuable space in your home or other places. The Takagi Flash TK1 will deliver convenience, space savings, and endless hot water for all your residential and commercial needs.

Specifications:
Input BTU Rate: Natural Gas Min. 37,000 Btu – Max. 165,000 Btu; L.P. Gas Min. 37,000 Btu – Max. 165,000 Btu

Thermal Efficiency: Natural Gas Max. 82.3%; L.P. Gas Max 84.7%

Energy Factor: Natural Gas 0.81; L.P. Gas 0.84

First Hour Rating (DOE): 216 Gal/Hr

Gas Connection: ¾" NPT

Water Connection: ¾" NPT

Water Pressure: Min. 15 psi – Max 150 psi

Gas Pressure: Natural Gas Min. 5 WC – Max. 10.5 WC; L.P. Gas Min. 11 WC – Max. 14 WC

Shipping Weight: 74 lbs.

Dimensions (H x W x D): 24.5"x16.5"x8.3"

Ignition: Electronic Ignition

Electrical Supply: AC 120 V, 0.8 A

Vent: 4 in.

Buy

Tankless Hot Water Heaterhot water heater, Takagi TK1, Tankless Gas Water Heater

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