A Parametric Analysis of Residential Water Heaters

by

Jeffrey Brad Maguire

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Written by Jeffrey Brad Maguire

Has been approved for the Department of Mechanical Engineering

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Moncef Krarti, PhD, PE

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Xia Fang, MS, PE

Date_______________

The final copy of this thesis has been examined by the signatories, and we

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Maguire, Jeffrey Brad (MS, Mechanical Engineering)

A Parametric Analysis of Residential Water Heaters

Thesis directed by Professor Moncef Krarti

Gas storage, gas tankless, condensing, electric storage, heat pump, and solar water heaters were simulated in several different climates across the US installed in both conditioned and unconditioned space and subjected to several different draw profiles. While many preexisting models were used, new models of condensing and heat pump water heaters were created specifically for this work to look at the majority of residential water heaters available on the market. The heat pump water heater model was extensively validated against both field test and lab test data and found to predict the performance of these units well in all of the situations examined. The condensing water heater model was compared to lab test data and found to provide a reasonable agreement with the measured data. A domestic hot water distribution system model was also created, validated, and used to examine the difference in distribution losses between heat pump water heaters and electric storage water heaters.

Annual simulations looked at both the energy savings potential and economics of these technologies. Heat pump water heaters were significant winners in both cost and energy savings for electric water heater and proved to be a cost effective replacement for electric storage water heaters even before incentives were considered in many cases. For gas water heaters, all technologies were able to save some energy with solar water heaters providing the most significant savings. However, none of the gas water heating options here proved to be cost effective without incentives. With incentives there were several situations where these water heaters were cost effective.

Acknowledgements

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List of Symbols and Abbreviations

A:	Area
C:	Thermal Capacitance
$Cost_{base}$	Base Case Capital Cost
$Cost_E$:	Energy Cost
$Cost_{breakeven}$:	Breakeven Cost
CFA:	Conditioned Floor Area
c_p:	Specific Heat
d:	Discount Rate
DHW:	Domestic Hot Water
DHWESG:	Domestic Hot Water Event Schedule Generator
E_{cons}:	Consumed Energy
E_{cool}:	Cooling Energy Consumption
E_{del}:	Delivered Energy
E_{elec}:	Electric Energy Consumed
E_{fan}:	Fan Energy Consumption
E_{gas}:	Energy Consumed from Natural Gas
E_{heat}:	Heating Energy Consumption
E_{HX}:	Energy Delivered by Solar Water Heater Heat Exchanger
E_{nrmlz}:	Normalization Energy
E_{WH}:	Water Heater Energy Consumption
EF:	Energy Factor
EF_{simple}:	Simplified Energy Factor
ELA:	Effective Leakage Area
F':	Solar Collector Fin Efficiency
f_{bottom}:	Fraction of Tank Area Below Lower Element
f_{top}:	Fraction of Tank Area Above Lower Element
$h_{outside}$:	Outside Heat Transfer Coefficient
HPF:	Heat Pump Fraction
HPWH:	Heat Pump Water Heater
I:	Investment Cost
LCC:	Life Cycle Cost
M_1:	Mass of Water Drawn During First Draw in Energy Factor Test
MC_{base}:	Base Case Maintenance Costs
MC_{WH}:	Water Heater Maintenance Costs
$\dot{m}$:	Mass Flow Rate
N:	Expected Water Heater Lifetime
OC_{base}:	Base Case Operating Costs
OC_{WH}:	Water Heater Operating Costs
OM&R:	Operating, Maintenance, and Repair Costs
P_r:	Rated Power
Q_{ground}:	Heat Transferred from a Home to the Ground
Q_{load}:	Thermal Load
$\dot{Q}_{rated}$:	Rated Heat Input

Q_{stdby}:	Energy Consumed During Standby
RE:	Recovery Efficiency
Repl:	Replacement Cost
Res:	Residual Value
S:	Total Absorbed Solar Radiation
SF:	Solar Fraction
SHR:	Sensible Heat Ratio
SLA:	Specific Leakage Area
t:	Time
T:	Temperature
$\bar{T}_{24}$:	Average Tank Temperature at the End of Energy Factor Test
T_{amb}:	Ambient Air Temperature
$\bar{T}_{del,1}$:	Average Delivered Temperature during First Draw in Energy Factor Test
T_g:	Ground Temperature
T_{in}:	Inlet Temperature
$\bar{T}_{in,1}$:	Average Inlet Temperature during First Draw in Energy Factor Test
T_{mains}:	Mains Water Temperature
$\bar{T}_{max}$:	Maximum Average Tank Temperature after First Draw during the Energy Factor Test
T_{node}:	Node Temperature
$\bar{T}_o$:	Average Tank Temperature at the Start of the Energy Factor Test
T_{out}:	Outlet Temperature
T_{req}:	Required Outlet Temperature
T_s:	Space Temperature (conditioned or unconditioned)
$\bar{T}_{su}$:	Maximum Average Tank Temperature after Cut-out During the Energy Factor Test
T_{tank}:	Storage Tank Temperature
U_L:	Overall Heat Loss Coefficient
UA:	Overall Heat Transfer Coefficient
UPV:	Uniform Present Value
V:	Volume of Water Drawn
$\dot{V}$:	Volumetric Flow Rate
V_{st}:	Measured Tank Volume
V_{wind}:	Wind Velocity
W:	Water Costs
y:	Length of Economic Study
ε:	Emissivity
η:	Efficiency
η_c:	Conversion Efficiency
η_r:	Recovery Efficiency
φ:	Shape Factor
ρ:	Density
τ_{stdby}:	Standby Period in Energy Factor Test
$\#_{node}$:	Number of Node

Chapter 1: Introduction and US Market Factors

1.1 Introduction

Water heating is the second largest energy use in homes in the US after space conditioning (1), accounting for 20% of the total energy consumed, or 2.12 Quads annually. Most homes in the US use either a natural gas or electric storage water heater (2), but many higher efficiency water heating options are available. These include tankless water heaters, condensing water heaters, heat pump water heaters, and solar water heaters. All of these different water heating technologies could provide energy savings to homeowners. Due to relatively short average lifespan of a water heater, these technologies are often considered as a way to reduce energy consumption in retrofit situations. However, these units are more complicated than a gas or electric storage water heater. Many factors, in particular mains temperature, the location of the water heater within the home, and the daily draw volume and profile impact the actual in use efficiency of these units.

Detailed, validated models of these different water heaters were used to provide insight into the actual in use efficiency and the impact of the aforementioned factors on the annual energy consumption of these water heaters. These results can be used to help homeowners choose the most efficient option for their particular situation. In addition to determining the most efficient option, the most cost effective option was also determined for each situation. This was done by calculating the life cycle cost of each unit as well as the breakeven cost.

While the technologies covered here (typical gas storage, typical electric storage, tankless gas, heat pump, condensing storage, and solar with both gas and electric backup) represent many of the most common water heating efficiency upgrades, there are several technologies not covered here. These include tankless electric water heaters (both large central tankless water heaters and point of use “booster” tankless water heaters), high efficiency gas (non condensing) and electric storage water heaters, a heat pump with a desuperheater, and condensing tankless water heaters. In the case of tankless electric water heaters and high efficiency gas and electric storage water heaters, these technologies were not included because they represent only a small potential savings with a modest increase in efficiency over the base case. Another potential water heating option is using a desuperheater with a heat pump to provide hot water. However, this option is not very common in the US at this time and was therefore excluded. In the case of tankless condensing water heaters, not enough information is available to create and validate a model of this technology. Future work will hopefully provide the information necessary to create this model.

The distribution system losses may also play a role in determining the overall energy consumption of water heating by a home. For mixed draws, homeowners often wait until a minimum useful temperature of water is reached, wasting water and energy as they do so. To quantify some of these effects and determine the impact of distribution system insulation on reducing this waste, a benchmark distribution system was modeled. The impact of alternate distribution system layouts, different piping materials, and recirculation systems were not examined here. In addition, the distribution system was modeled for both electric and heat pump water heaters (HPWHs) to determine how distribution losses change when moving to HPWHs.

This thesis begins by discussing the current US water heating market and providing an overview of these different water heating technologies. This is followed by a discussion of the different models employed in this work as well as validation results for any new models specifically created for this project. The distribution system model is then presented, followed by the details of the building models used in the whole home annual simulations. Finally, the results of the whole home simulations are discussed along with potential areas for future work.

1.2 Current US Water Heating Market

The residential water heater market in the United States is dominated by storage type water heaters. Gas and electric storage water heaters make up about 97% of the residential water heater market (2). Gas tankless water heaters make up the majority of the remaining market, with all other technologies only making up less than 1% of the market. 53% of homes in the US use natural gas as the primary fuel for water heating while 40% use electricity (2). The remaining homes use other fuel sources such as fuel oil, propane, wood, and solar. The distribution of water heater fuels varies by region as shown in Figure 1.

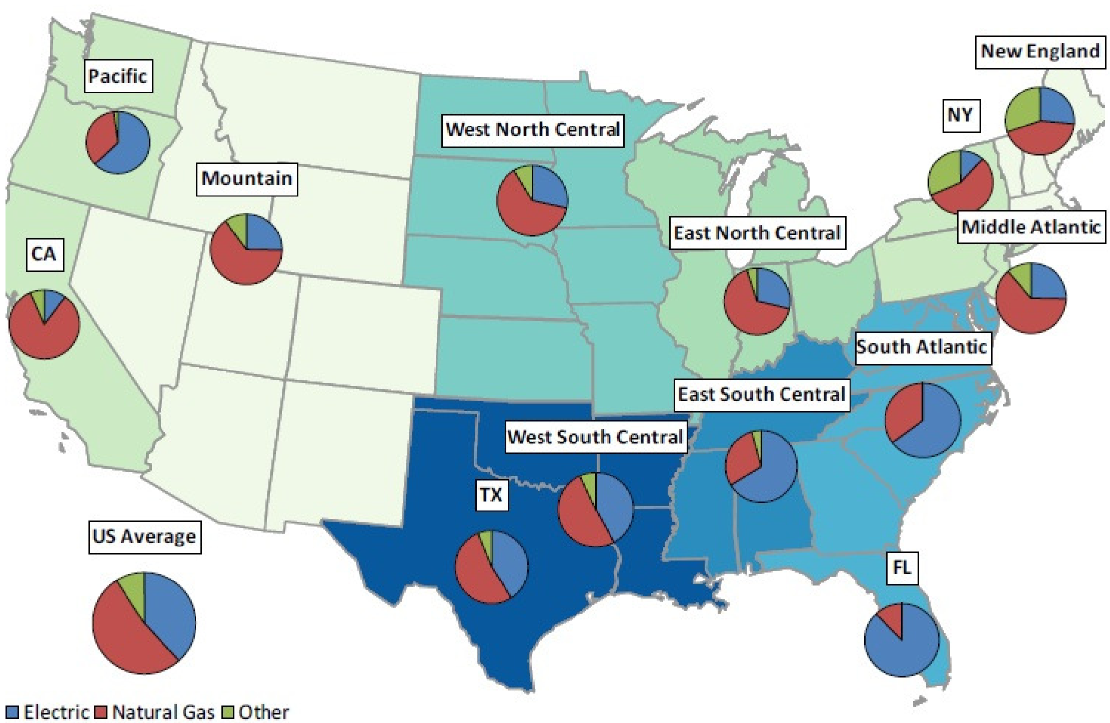


Figure 1: Distribution of fuel types for installed residential water heaters (3)

The Energy Star® branding for different water heater technologies plays a role in the US market for water heaters. Energy Star qualified units must meet certain energy efficiency requirements to be eligible for the branding. Consumers are very aware of the Energy Star program: two thirds of households can recognize the Energy Star label on sight, and over three quarters of households have at least a general understanding of the label's purpose (4). Any Energy Star qualified unit that is purchased is eligible for a tax credit, providing an incentive for consumers to seek out energy efficient products with this label. There are currently Energy Star standards for all water heater technologies except electric storage and electric tankless units. The Energy Star standards for condensing water heaters and heat pump water heaters are new standards that have been developed only in the last few years. There are currently no Energy Star certified condensing storage water heaters (5) even though there has been a standard for them over a year. Currently, Energy Star certified units make up 6.5% of total water heater sales as shown in Figure 2 (5).

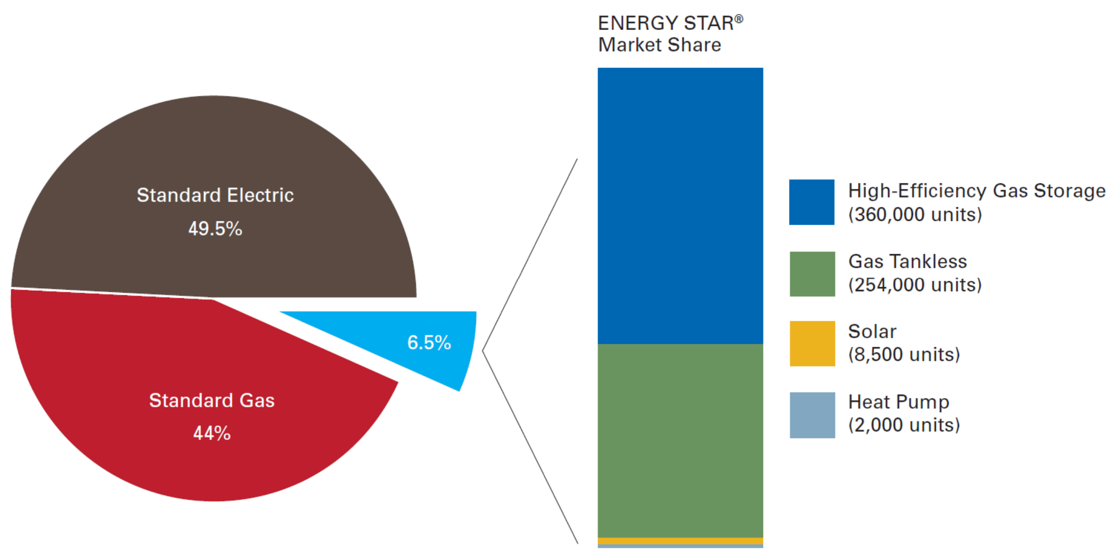


Figure 2: US residential water heater sales by technology (5)

Recently announced US energy efficiency standards for residential water heaters require gas and electric storage tanks with a capacity over 55 gallons to use either condensing (for gas) or heat pump (for electric) technologies (6). The new standard will go into effect in 2015. This new standard could be a significant driver for change, leading to a wide adoption of more energy efficient water heaters in the coming years.

1.3 Market Barriers for Efficient Water Heating Technologies

The largest market barrier for energy efficient water heaters for residential applications is the high first cost. A highly efficient water heater can cost several times what a comparable less efficient system would cost as shown in Table 1. In most cases, the energy savings will offset the higher first cost in a few years, but that does not always provide enough of an incentive. This is particularly true if the water heater is installed for someone other than the occupant, such as the owner of a rental property, who would not see any benefit from the energy savings provided by the more efficient system when utility bills are not included in rental fees. Poor customer awareness and a lack of trained installers for some water heating technologies is another common barrier (7). In retrofit situations, additional work, such as installing a larger gas line or a new electric circuit, may be required when switching to a new technology, increasing the installation cost. Finally, the often immediate need for a new water heater can be a serious obstacle to the adoption of energy efficient technologies. Thirty percent of water heaters are also purchased because the previous unit has failed catastrophically (2), in which case the customer will rarely perform an extensive search for an efficient water heater and will instead take whatever is "on the truck".

Residential Water Heating Technology	Approximate Capital Cost ($)	Average Lifetime (years)	Energy Factor
Gas Storage	620-890	13	0.58
Electric Storage	470-750	13	0.9
Gas Condensing	1300-1800	15	.8-.95
Gas Tankless*	1500-2500	20	.82-.98
Heat Pump	1300-1800	10	2-2.35
Solar	2000-3500**	20	0.5-1***

*Includes condensing and non condensing units **After Federal tax credit *** Solar Fraction

Table 1: Comparison of cost, rated efficiency, and lifetime of different water heating technologies (2)

For condensing water heaters, the high first cost largely comes from the more expensive materials required in the heat exchanger for the flue gas, which must have a high corrosion resistance (7). Recently manufactured condensing water heaters have had the high first cost in mind when designing the system, and attempts have been made to design condensing water heaters with lower first costs by using parts from water heaters currently on the market whenever possible and experimenting with different materials for the heat exchanger. An ideal market for condensing water heaters is high use residential applications (such as combined space and water heating applications) and light commercial applications, where the high first cost may be less of a factor (8).

Heat pump water heaters have also seen poor market penetration, although they have been available for several years. The main reason for this is the high first cost. They can cost 2-3 times as much as a comparable electric storage water heater, which provides a significant barrier to entry in the market. Another significant barrier to entry for heat pump water heaters is the perception that they have reliability issues (9). This perception comes from pilot programs conducted by utilities and federal agencies with some of the first residential heat pump water heaters. While the technical issues have been fixed, this perception of poor reliability remains with people who were aware of the pilot programs. Until recently, heat pump water heaters were primarily made by small manufacturers, and it is unlikely that they would have been able to meet a large surge in demand. Several large manufacturers, including General Electric, Rheem, and A. O. Smith have entered the market and currently have Energy Star qualified heat pump water heaters available. It is still unclear how large an impact these new units will have as they have only been on the market for approximately one year.

In 2006, about 8,500 new solar water heaters were sold, making up less than 1% of new water heater sales (2). The largest market barrier for solar water heaters has been the high first cost. Looking at Table 1, it becomes apparent that the solar water heater costs more than twice as much as a gas storage water heater, even after federal tax credits are included in the cost. Solar water heaters also need to be roof mounted. This requires an approximately south facing roof that is not shaded for most, if not all, of the year, which further limits who would consider installing a solar water heater.

Gas tankless water heaters also have a high first cost as the largest market barrier, particularly in the case of condensing tankless water heaters. However, they have had the highest market penetration of any of the high efficiency technologies discussed here, with 254,000 units sold in 2006 (5). In retrofit situations, the larger burner of a tankless water heater may require a larger gas line and vents to be installed, further increasing the first cost of these units. Regular maintenance may need to be performed on these units to remove any scale build up from inside the heat exchanger, particularly in areas with hard water, although there is some disagreement on how serious this issue is (10) and whether such maintenance is truly necessary.

Chapter 2: Strengths and Weaknesses of Water Heating Technologies

2.1 Gas Storage Water Heaters

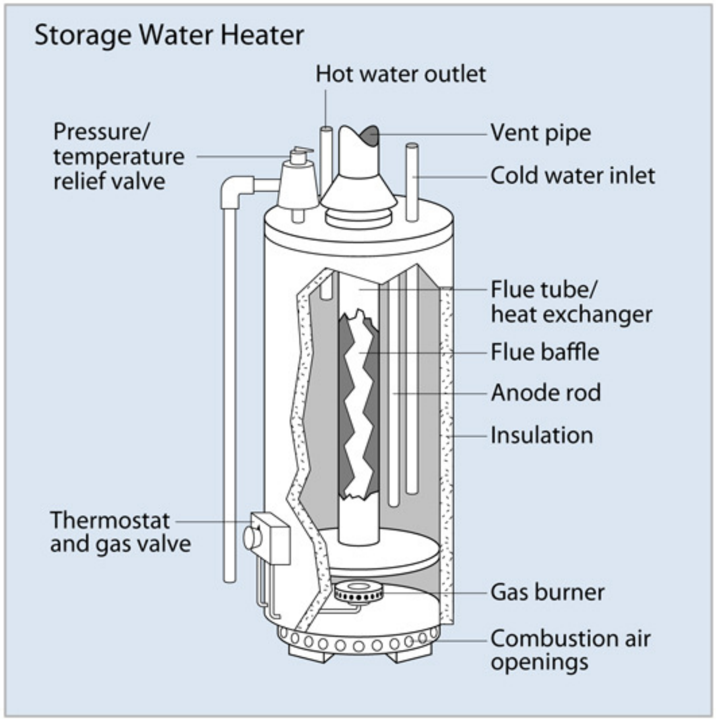


Figure 3: Gas storage water heater (11)

Gas storage water heaters (see Figure 3) are the most common type of gas water heater. They are also the least efficient (minimum EF=0.58 for a 50 gallon unit) gas water heating option. This low efficiency comes from two factors: the combustion efficiency of turning natural gas into heat and the tank losses. A large part of the low combustion efficiency is that typical gas water heaters need to vent the combustion products at a relatively high temperature. Sulfur is commonly added to natural gas as an odorizer for safety reasons, which is why a gas leak may smell like rotten eggs. If the flue gas were allowed to reduce to a temperature where the water vapor condenses out of the flow, this sulfur could combine with the water to make sulfuric acid, corroding the flue and destroying the water heater.

Tank losses are especially high in gas storage water heaters because of the central flue in most gas water heaters. Convection loops can form in this flue, further increasing the heat loss. There are some higher efficiency designs such as power venting water heaters that reduce this loss through the central flue, but these units are more expensive. High efficiency non condensing gas storage water heaters are not considered in this work as they represent only an incremental improvement in this type of water heater.

2.2 Electric Storage Water Heaters

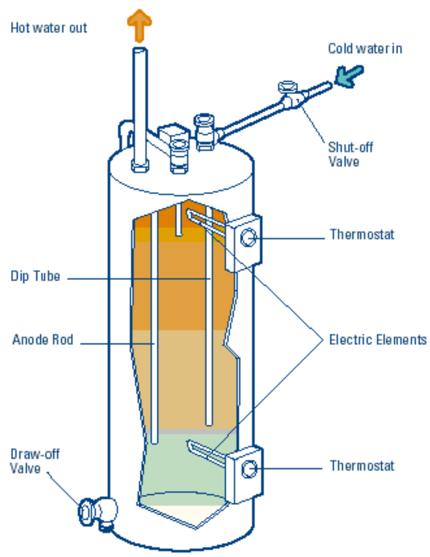


Figure 4: Electric storage water heater (12)

Electric storage water heaters (see Figure 4) are the most common electric water heating option. They are more efficient than gas storage water heaters (minimum EF=0.90 for a 50 gallon unit), but are the least efficient electric option. The conversion efficiency of electric water heaters is very close to 1, so the major source of inefficiency in electric storage water heaters is tank losses. Electric water heaters do not need a flue, which reduces their tank losses relative to a gas storage water heater. Most electric water heaters use two electric elements in a master-slave relationship with the upper element as the master to meet the load in the most efficient manner possible. Higher efficiency electric storage tank water heaters are possible by increasing the jacket insulation of the storage tank, but these savings are modest on a per unit basis. However, their national savings potential may be large if widespread adoption were to occur (13).

2.3 Gas Tankless Water Heaters

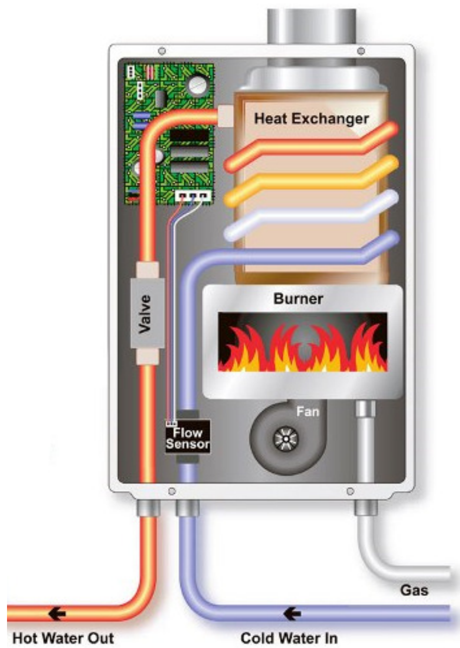


Figure 5: Tankless water heater (14)

Gas tankless water heaters (see Figure 5) improve on the efficiency of gas storage water heaters by removing the standby losses. Typical gas tankless water heaters can therefore achieve much higher efficiencies (EF=0.80) than gas storage water heaters. However, they do have some disadvantages when compared to a gas storage water heater. Tankless water heaters have a minimum flow rate that must go through the water heater before they will turn on. Additionally, once a draw begins and the burner fires, both the water and the heat exchanger have to come up to the set point temperature. As a result, during each draw there are losses associated with bringing the heat exchanger up to temperature. If a homeowner uses hot water with many short draws spread out over a day, these losses can be significant.

One other issue associated with tankless water heaters is the "cold water sandwich". The "cold water sandwich" commonly occurs between two closely spaced hot water draws (for example, two morning showers). After the first draw, there may still be hot water in the pipes but the tankless water heater will have turned itself off because no water is being drawn. The second event will start with hot water from the pipes, but will be followed by a slug of cold water right before the tankless water heater burner ignition and then hot water once the water heater fully fires. The cold water sandwich is primarily a comfort issue and does not significantly impact the efficiency of the unit. It can be avoided by installing a small buffer tank with the tankless water heater or using a control strategy designed to avoid the issue.

2.4 Condensing Storage Water Heaters

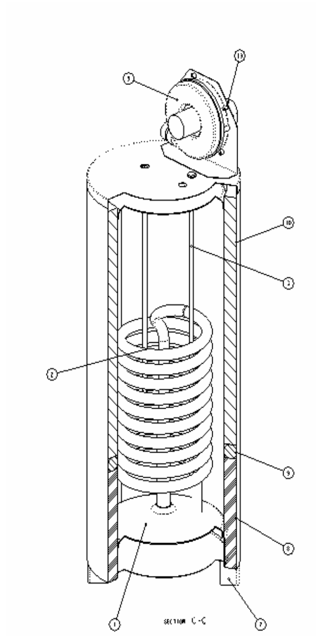


Figure 6: Condensing water heater (8)

Condensing storage water heaters (see Figure 6) improve on typical gas storage water heaters by allowing the flue gas to condense before it is vented. This condensation allows the latent heat of the water vapor to be captured, significantly increasing the efficiency (EF=0.80) of this unit. Additionally, this design typically gets rid of the central flue in favor of a helical heat exchanger in the center of the tank, which reduces the standby losses of the unit. To avoid corrosion issues associated with the acidic condensate, a corrosion resistant heat exchanger is required. This heat exchanger is typically made of either glass lined or stainless steel and represents a substantial additional cost, raising the price of these units. However, the low temperature at which it vents means the vents can be made of PVC instead of metal, reducing the installation cost in new construction (15).

While the efficiency of this unit is a large improvement over traditional gas storage water heaters, the conversion efficiency is not constant. Actual efficiency varies depending on the amount of condensation that occurs in the heat exchanger. The amount and rate of condensation is directly impacted by the temperature of water surrounding the heat exchanger and the part load (if the unit is capable of modulating).

2.5 Heat Pump Water Heaters

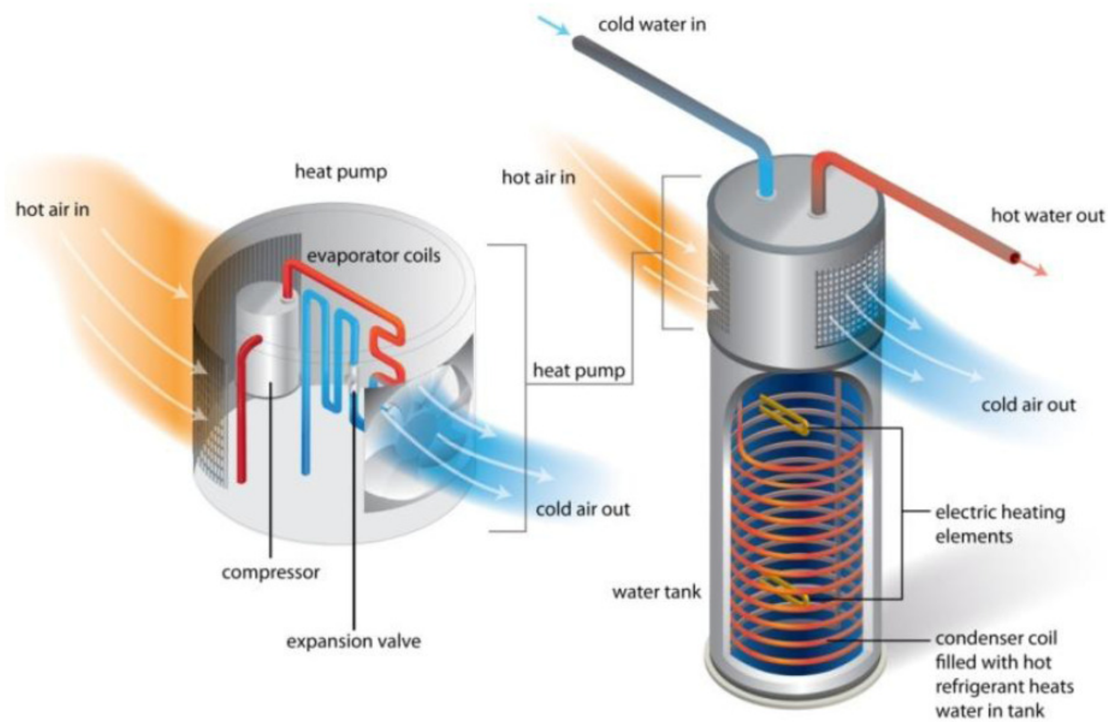


Figure 7: Heat pump water heater (16)

Heat pump water heaters (HPWHs, see Figure 7) are the highest efficiency electric water heaters available other than solar. Most HPWHs in the US are air source and operate by using a heat pump to remove heat from the ambient air and add it to the tank. However, other heat sources could be used (17). Typical COPs for these units are around 2-3 internationally, with rated COPs typically ranging from 2-2.5 in the US. Heat pump water heaters have long had a significant market share in Japan, where some HPWHs using CO_2 as the refrigerant can achieve a COP of 4 or higher (18).

HPWHs typically feature both a heat pump and electric resistance elements for heating. The electric resistance elements typically turn on if the heat pump cannot keep up with the load or if the ambient air conditions prevent the heat pump from running. Each manufacturer has their own control logic for determining when to switch to the backup electric resistance elements based on their system's design. How often the backup electric resistance elements have to be used is heavily dependent on climate and hot water use. A "heat pump fraction" metric, analogous to the solar fraction for solar water heaters can be used to approximately evaluate the performance of these units when they are located in unconditioned space. Figure 8 shows one manufacturer’s predicted heat pump fraction for an average household in different locations. Zone 1 represents a heat pump fraction of 0.9-1, Zone 2 represents a heat pump fraction of 0.6, and Zone 3 represents a heat pump fraction of 0.5.

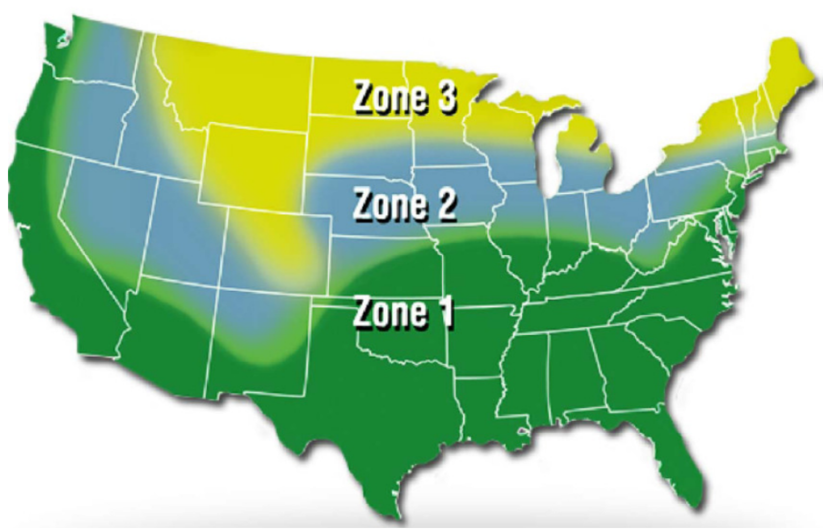


Figure 8: Manufacturer's map showing heat pump fraction (19)

The heat pump COP depends heavily on the temperature of water adjacent to the condenser and the ambient air conditions (both temperature and humidity), which can cause the actual in use efficiency of this unit to vary widely depending on where it is installed, particularly when it is installed in unconditioned space. When this unit is installed in conditioned space, it will cool and dehumidify the space it is in while it is in use, which may increase the overall energy consumption of the building. This unit must also be installed in an area with enough airflow to ensure that it does not significantly cool down the air around it, which would cause the unit to reduce its own COP. In retrofit situations, this may require the installation of a louvered door if the water heater is located in an enclosed closet. Ducting the HPWH is also a possibility, although none of the units currently available are configured for ducting.

2.6 Solar Water Heaters

Solar water heaters are the oldest of the high efficiency water technologies discussed here and have had a fairly long and interesting history in the US. The first commercially available solar water heater, the Climax, was made available in 1891 and was primarily sold in southern California. Solar water heater sales in southern California peaked at 1000 units sold in 1920, but sales rapidly declined when cheap natural gas became widely available (20). Solar water heater sales then moved to Florida in 1923, and average sales were between 4000 and 10000 units in Miami alone between 1935 and 1941. Solar water heater manufacturing stopped during World War II, and competition from electric water heaters after the war drastically reduced solar water heater sales. Sales of solar water heaters picked up again after the first Arab oil embargo of 1973, peaking at over 20 million square feet of solar thermal collectors shipped in 1980. A combination of the repeal of tax credits for solar water heaters and a bad reputation coming from some poorly manufactured solar water heaters drastically reduced solar water heater sales in the

1980s (21). Solar water heater sales today have increased compared to sales in the 1980s due to the return of federal tax credits as well as many state incentives for solar water heaters.

Solar water heaters offer an opportunity to greatly reduce the gas or electricity consumption for hot water. A solar water heater uses the sun to heat water for use in domestic hot water or hydronic space heating applications. Solar water heaters typically provide over half of the energy required by a household for water heating, while a backup water heating system provides the remaining energy (5). The backup heating system could be any residential water heater, including a condensing or heat pump water heater. There are currently over 100 different models of solar water heaters on the market in the US (2). However, solar water heaters currently make up less than 1% of the market in the US.

There are several types of collectors which can be used for solar water heating. The two most common options are a flat plate collector and an evacuated tube collector, but there are also systems with the water storage integrated into the collector (these are commonly referred to as ICS, or integrated collector storage systems). Flat plate collectors consist of a collector surface that absorbs the solar radiation, a glazing to prevent the absorber from reradiating the solar energy, a heat transfer medium (for domestic hot water applications, this is almost always water or a propylene glycol base heat transfer fluid, depending on the climate and system type), and insulation on the sides and back to prevent losses by conduction and convection (22). Evacuated tube collectors consist of several absorber surfaces with heat transfer fluid flowing through them encased in a vacuum. The vacuum around the collector minimizes the losses to the environment from the collector since it drastically reduces the conduction and convection losses from the system. However, they are typically much more complicated, and therefore more expensive, than flat plate collectors. ICS systems typically consist of a flat plate collector with a large enough volume to also act as a storage tank. Since they are especially susceptible to freezing, they are typically only used in climates where the outdoor air temperature rarely gets below freezing.

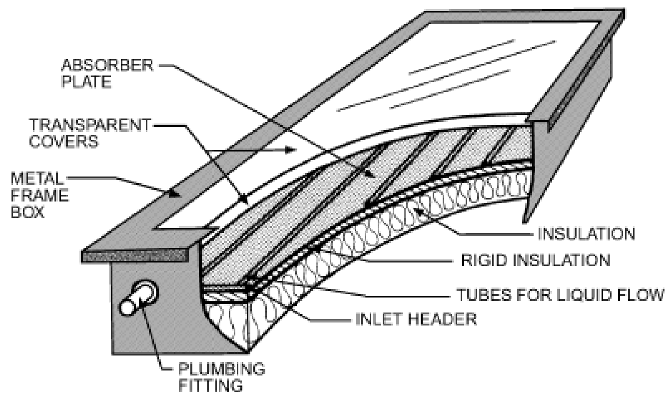


Figure 9: Typical flat plate collector (23)

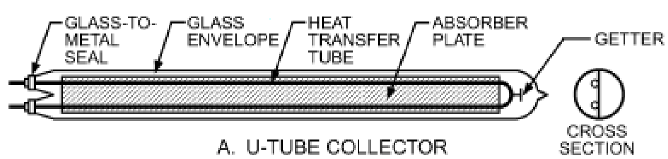


Figure 10: Typical evacuated tube collector (23)

The decision of which collector type is used typically depends on the inlet fluid parameter, which itself is dependent on the ambient temperature, the inlet fluid temperature, and the incident solar radiation intensity (24). For the temperatures used in most residential applications, flat plate collectors are more efficient, although for higher temperature applications an evacuated tube collector may be more efficient. Flat plate collectors also make up 76% of the collector sales for domestic water heating applications in the US (25), and are therefore the only solar water heating technology considered here. Evacuated tube solar water heaters are better suited to commercial or light industrial applications.

Solar collectors have been in use for a relatively long time compared to other high efficiency water heating technologies and because of this a rating procedure for collector efficiency is well established. Solar collectors in the US are rated by the Solar Rating and Certification Corporation (SRCC), which tests different collector designs under standardized conditions to characterize the thermal performance of the collector (26). The SRCC procedure takes data from standardized tests (either ASHRAE 93-77 or 96-1980) and produces predictions for the amount of energy captured by the panel per day depending on the weather and application. The SRCC test also produces an efficiency equation for the collector that provides the efficiency as a function of the inlet fluid parameter. This data is used for projection by the SRCC of energy savings for each solar water heater in a variety of locations. The SRCC also makes this data available so that it can be used for the modeling of solar water heaters.

Chapter 3: Existing Water Heater Models

Most of the exiting water heating technologies have already been modeled in the TRNSYS environment (27). Models for gas storage, electric storage, and solar water heaters have existed since the initial release of TRNSYS, although recently created models also exist that more accurately reflect the actual performance of these units by capturing effects not included in the original models (28) (29). Accurate models of gas tankless water heaters have been recently created and verified (30). However, accurate models of a HPWH and a condensing water heater needed to be created for this particular work. A description of the existing models is provided below.

3.1 Gas and Electric Storage Water Heaters

The gas and electric storage water heater models used here both use the same multi-node storage tank model (28). This model consists of a storage tank subdivided into a user defined number of vertical isothermal nodes as shown in Figure 11. 15 nodes have generally been shown to be adequate for capturing the stratification in these types of water heaters (31). An overall heat transfer coefficient (UA) is specified for each node in the water heater to allow any thermal shorts in the tank to be modeled at any location in the water heater. A flue loss coefficient can also be specified for the tank to model the losses from the central flue in a gas water heater. Heat can be added to any node in the water heater, and the tank inlet and outlet can also be located in any node.

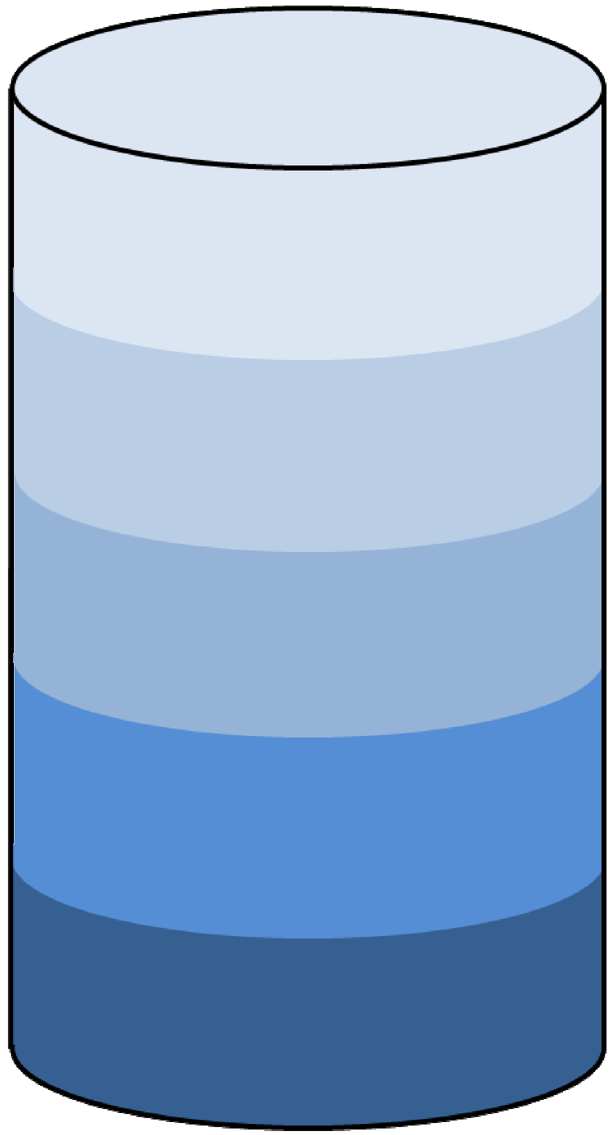


Figure 11: Stratified storage tank water heater model. Isothermal nodes are shown as varying colors.

For gas and electric water heaters, a fairly large margin of error is allowed in the actual volume of the water heater when compared to the nominal volume. For gas water heaters, the actual volume has to be the nominal volume ±5%. For electric water heaters, the actual volume can be within ±10% of the nominal volume (31). Most manufacturers can produce tanks with much less variation in the actual volume than what is required and therefore tend to produce tanks with a volume at the low end of the specified range for cost reasons.

Two parameters need to be derived to successfully model these units: an overall heat transfer coefficient (UA), which can be applied uniformly to all nodes unless there are specific thermal shorts that need to be captured, and a conversion efficiency (η_c) for the heating device. These parameters can be derived from the standard rating tests for these units using by applying an energy balance to the storage tank for the duration of the tests as shown in (31). For a gas water heater, the overall heat transfer coefficient can be expressed as:

$$UA_{gas} = \frac{\frac{RE}{EF} - 1}{(T_{tank} - T_{amb})\left((24hrs/Q_{load}) - \frac{1}{P_r \times EF}\right)} \tag{1}$$

In Equation 1, *RE* is the recovery efficiency, *EF* is the Energy Factor, T_{tank} is the average storage tank temperature over the entire test period, T_{amb} is the ambient air temperature, Q_{load} is the thermal load delivered by the water heater, and P_r is the rated power for the water heater. The conversion efficiency can be expressed as:

$$\eta_c = RE + UA\frac{(T_{tank} - T_{amb})}{P_r} \tag{2}$$

For a gas water heater, the conversion efficiency is the combustion efficiency of the water heater. A full derivation of Equations 1 and 2 can be found in (31). The tank temperature, ambient air temperature, and load delivered come directly from the conditions specified in the ratings test definition (32). No standing pilot light was modeled for this unit. The impact of a pilot light on the annual energy consumption of a gas water heater and a solar water heater with gas backup is discussed in Appendix G.

In order to accurately consider the impact of locating this water heater in conditioned space, the fraction of the heat loss that goes out the flue as opposed to going to the space it is located in and impacting the heating and cooling loads of this space needs to be determined. It has been found (33) that the installation of an electromechanical flue damper can reduce the overall heat transfer coefficient by 1/3 for a 40 gallon gas water heater. This change in heat transfer coefficient was derived assuming that the flue damper blocked 90% of the total flue area, leaving 10% still open for air flow. This work was also done for a typical 40 gallon water heater, which means the flue area and the tank surface area may change for different water heaters. However, this division of the heat loss was assumed in all non condensing gas water heaters modeled here due to a lack of data on other configurations.

In the case of an electric water heater, it is typical to see a mass of colder water below the lower element during much of the ratings test as little mixing occurs between this water and the water above the lower element (34). An electric water heater tank temperature profile is shown in Figure 12 to illustrate this phenomenon.

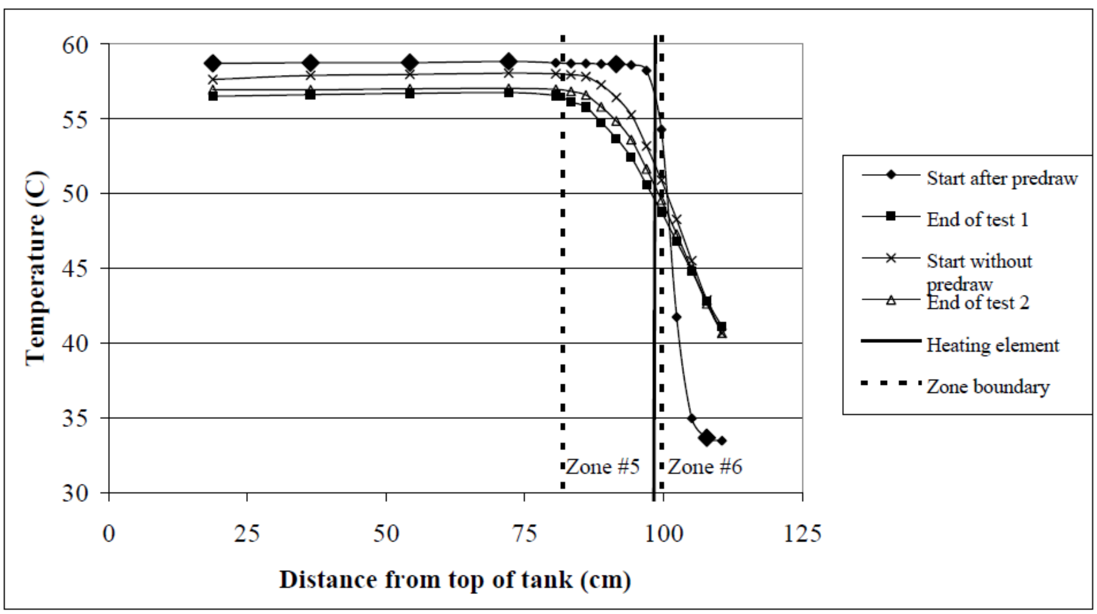


Figure 12: Temperature profile in an electric storage water heater during Energy Factor Testing (34). A significant thermocline occurs in the region below the lower heating element

In this case, the model parameters can still be derived from the ratings test but the tank can no longer be considered isothermal, so a more complicated derivation is required (35). The conversion efficiency for an electric water heater is typically 1 as electric resistance elements have an efficiency very close to 1. The overall heat transfer coefficient can be expressed as:

$$UA = \frac{\frac{Q_{load}}{EF - 1}}{(T_{tank} - T_{amb})(24hrs)\left(f_{top} + f_{bottom}\frac{T_{mains} - T_{amb}}{T_{tank} - T_{amb}}\right)} \tag{3}$$

Most variables in Equation 3 are the same as those in Equation 1. f_{top} is the fraction of the tank area above the lower element while f_{bottom} is the fraction below. The temperature of water in the lower section of colder water is assumed to be midway between the tank set point temperature and the ambient air temperature.

The controls for both of these units also need to be modeled. For a gas water heater, there is typically only 1 temperature measurement taken near the bottom of the tank as heat is input at the bottom of the tank and there is rarely significant stratification within the tank. For an electric water heater, two electric elements, both with their own thermocouple to measure tank water temperature near the element, are included and both try to maintain the set point temperature near them. These elements typically have a master-slave control strategy with the top element as the master. For the gas water heaters, a dead band of ±5 °F was used for the controls. For the electric water heater, a dead band of ±5 °F was used to control the upper element, while ±10 °F was used for the lower element, as the temperature near the bottom of the tank has less of an impact on the delivered temperature unless a very large draw that nearly depletes the tank occurs. The modeling parameters used for the gas and electric water heater are given in Table 2.

	Electric Water Heater	Gas Water Heater
Tank Volume	45.5 gallons	47.6 gallons
Tank Height	46 inches	43 inches
Tank Loss Coefficient	0.161 Btu/hr-ft^2-F	0.46 Btu/hr-ft^2-F
Rated Power	15335 Btu/hr	50000 Btu/hr
Conversion Efficiency	100%	82.3%
Number of Nodes	15	15
Location of Upper Control Temperature	Node 4	NA
Location of Lower Control Temperature 2	Node 13	Node 13

Table 2: Model parameters for gas and electric storage water heaters

3.2 Gas Tankless Water Heaters

The gas tankless water heater model used here is a multiple node gas tankless water heater model. This model was created based on extensive lab testing of the unit (30). The TRNSYS model used here has parameter values that are specific to the particular model of tankless water heater used during lab testing. The model subdivides the heat exchanger of the gas tankless water heater into multiple nodes and performs an energy balance on each node. Multiple nodes are required to capture the dynamics of the unit heating and cooling for each draw. This particular water heater is capable of modulating gas flow into the unit to fire at the minimum rate necessary to meet the load. The governing equation for each node in this model (30) is:

$$\frac{C}{\#_{node}}\frac{dT_{node}}{dt} = \frac{\eta_{ss}\gamma\dot{Q}_{rated}}{\#_{node}} - \dot{m}_{fluid}c_p\left(T_{out,node} - T_{in,node}\right) - \frac{UA(T_{node}-T_{amb})}{\#_{node}} \tag{4}$$

In Equation 4 C is the capacitance of the tankless water heater heat exchanger, #node is the number of nodes used in the model, T_{node} is the temperature of the tankless water heater node, η_{ss} is the steady state conversion efficiency of the tankless water heater, γ is the control signal, $\dot{Q}_{rated}$ is the rated heat input into the tankless water heater, $\dot{m}_{fluid}$ is the flow rate of water through the water heater, c_p is the specific heat of water, $T_{out,node}$ is the temperature of water leaving the node, $T_{in,node}$ is the temperature of water entering the node, UA is the overall heat transfer coefficient of the water heater, and T_{amb} is the ambient air temperature. This equation is solved for each node at every timestep to determine the temperature of each node and fully capture the performance of the tankless water heater.

For a tankless water heater, the key model parameters (overall heat transfer coefficient, capacitance, and steady state efficiency) cannot be derived from the standard rating tests. Instead, lab testing must be performed to accurately determine the model parameters. The aforementioned lab testing was designed to derive these key parameters and the results were used to determine their values. In addition, the heat exchanger geometry and electricity consumption during operation and standby was determined from this testing.

To correctly model the performance of this unit, sub timesteps may be required to model different modes of operation and delays in response time if the simulation timestep is large. Tankless water heaters have two different delay times: there is a delay at start up between when the unit detects a flow of water through it and when the burner turns on and during operation between when the unit detects a change in water flow rate and the burner modulating to the correct burn rate. Testing determined that both of these delays are approximately 3 seconds. However, this model appeared to have difficulty correctly modeling performance when sub timesteps were used. To avoid issues with sub timesteps, the delay in the unit turning on was set to 6 seconds, while the controls during operation were allowed to instantly modulate the gas flow rate. These assumptions should have a minimal impact on simulation results during annual simulation (36).

Two important parameters not determined during lab testing are the freeze protection electricity consumption and the amount of heat that is lost out of the flue. When this unit is installed in unconditioned space, there is a potential for the water entrained in the heat exchanger to freeze if the unit has been idle for a long time. Since any freezing of this water could result in catastrophic failure of the water heater, most tankless water heaters have an electric resistance heater that will turn on if an ambient temperature close to freezing is detected by the unit. If the electric heaters are unable to heat the unit sufficiently to prevent freezing, the gas burner may fire to ensure freezing does not occur.

The freeze protection algorithm here was provided by the manufacturer of this particular tankless water heater and was not validated by lab testing. To determine if freezing is likely to occur, the temperature at the inlet and outlet of the tankless water heater is monitored. If either of these temperatures drops below 38 °F, several electric heaters distributed across the heat exchanger will turn on and remain on until both temperatures are above 53 °F. These heaters draw a total of 100 W. To model the several different electric heaters distributed throughout the unit in TRNSYS, the 100 W drawn by the heaters was evenly distributed across the whole heat exchanger. If either of the measured temperatures drops below 35 °F (which can occur if the ambient temperature is so low that the electric heaters do not provide enough heat to prevent freezing), the gas burner will fire at full capacity until both measured temperatures are above 53 °F. It should be noted that there is also a variety of this particular tankless water heater that is designed to be located in conditioned space which has a different freeze protection algorithm. However, this algorithm is not modeled here as units in conditioned space never required any freeze protection energy use regardless of which algorithm is used.

Due to a lack of data on how much of the heat losses from the tankless water heater go out the flue compared to how much goes to the surrounding space, the same split that was used for gas storage water heaters (2/3 to the surrounding space, 1/3 out the flue) was used for the tankless water heater.

3.3 Solar Water Heaters

The solar water heater model used here consists of several different TRNSYS components connected together. The system is made up of a storage tank, a flat plate collector, a pump, a controller, and pipes connecting the flat plate collector to the storage tank. Two different types of solar water heating systems are considered. For the case where the backup fuel source is electric, a single storage tank with an electric resistance element in the upper half of the tank is used. For the case where gas is used as the backup fuel source, a dual tank system consisting of a solar storage tank with a separate gas storage water heater installed in line after the storage tank is used. These systems were chosen as they are the most common type of solar water heating systems in the US for each backup fuel source. A schematic of the solar water heating system used here is given in Figure 13 and Figure 14.

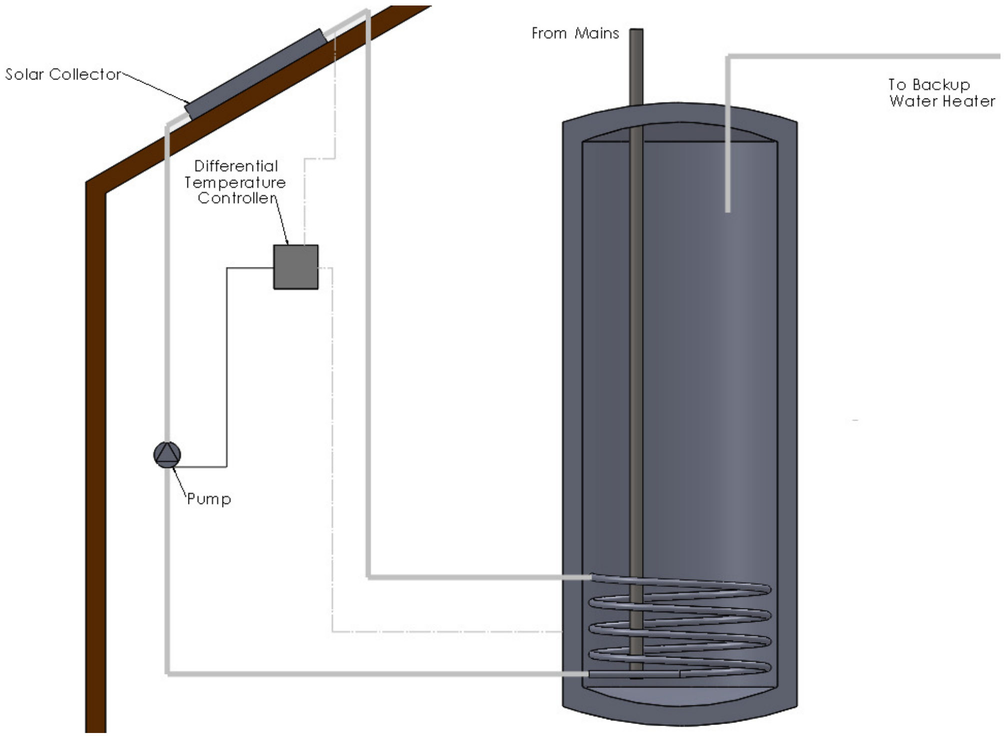


Figure 13: Schematic of solar water heating system with gas backup.

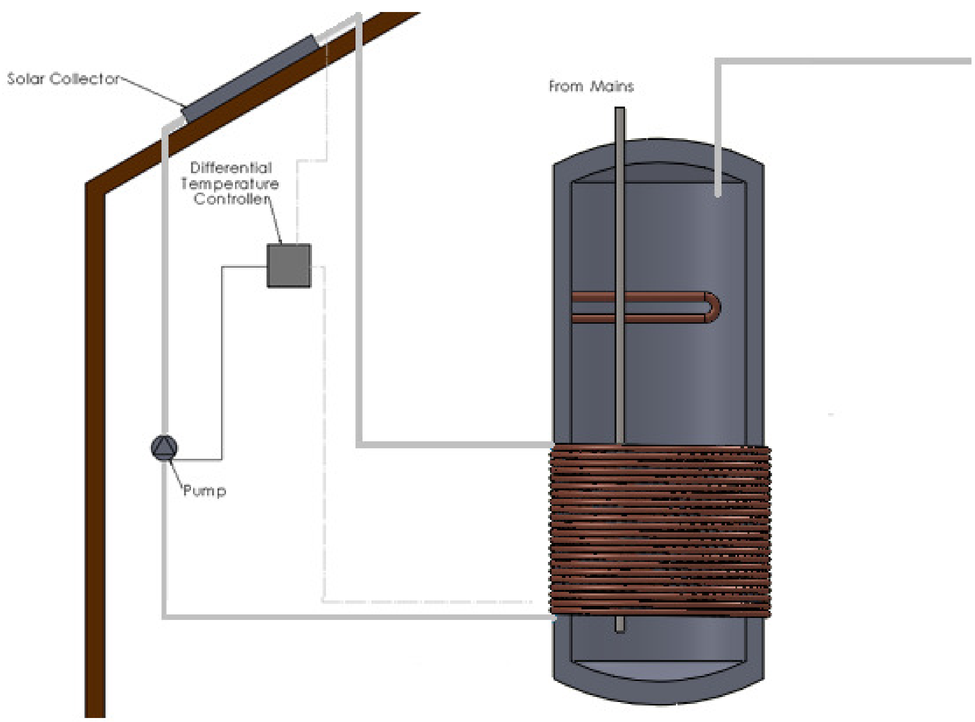


Figure 14: Schematic of solar water heating system with electric backup

The flat plate collector model used here is a modification of the original solar flat plate collector model in TRNSYS that includes the capacitance of the unit (29). Including the capacitance of the water heater allows transient effects associated with the heating and cooling of the collector to be considered in the model. This model is physics based as opposed to relying on a performance map. Collector model parameters were taken directly from Solar Rating and Certification Corporation (SRCC) test results (26). The SRCC tests are designed to capture all of the parameters necessary to model a solar collector in TRNSYS, as TRNSYS is used by the SRCC to provide a solar fraction for rated systems at different locations in the US (37). These test results are used by TRNSYS to derive several of the physical properties of the collector. For all cases, 2 collectors with a total area of 64 ft^2 were used. This is the most common collector area installed in the US (38). In general, smaller collector areas are only used for passive (thermosiphon) systems, which use water in the collector instead of a heat transfer fluid. These types of systems can only be used in areas where there is no risk of freezing (for example, in southern Florida and Hawaii) and are therefore rare in the locations considered in this study (39). Each collector is subdivided into 20 nodes, which allows the fluid heating and heat loss from the collector to be more accurately calculated. All collectors were oriented due south with a slope of 26.57°, which corresponds to a 6/12 roof pitch. Collector parameters used in this model are given in Table 3.

Number in series	2
Collector area	2.973 m^2
Fluid specific heat	3.559 kJ/kg-C
Collector test mode	1
Intercept efficiency (a0)	0.691
1st order efficiency coefficient (a1)	12.128
2nd order efficiency coefficient (a2)	0.0708
Tested flow rate per unit area	49.046 kg/hr-m^2
Fluid specific heat at test conditions	4.18
1st-order Incidence Angle Modifier coefficient	-0.194
2nd-order Incidence Angle Modifier coefficient	0.006
Minimum Flow rate	0 kg/hr
Maximum Flow rate	10000 kg/hr
Capacitance of Collector	17.729 kJ/kg
Number of Nodes	20

Table 3: Parameters used in the solar collector model (Type 539)

The governing equation for this model is (29):

$$C\frac{dT}{dt} = F'\big(S - AU_L(T - T_{amb})\big) - \dot{m}c_p(T - T_{in}) \quad (5)$$

In equation 4, C is the thermal capacitance of the collector, F' is the collector fin efficiency, S is the total absorbed solar radiation, A is the gross collector area, U_L is the overall heat loss coefficient of the collector, T is the collector fluid temperature, T_{amb} is the ambient temperature, $\dot{m}$ is the mass flow rate of fluid through the collector, c_p is the specific heat of the collector fluid, and T_{in} is the temperature of fluid entering the collector. S is calculated based on incidence angle modifiers which come from SRCC test results, and A is reported in the test results as well. F' and U_L can also be derived from the collector efficiency equations reported in the test results as shown in (29). C is not reported in the test results, and was instead derived based on the manufacturer's specifications for this collector. The capacitance of the collector includes the capacitance of both the copper collector pipes and the entrained water.

The solar storage tank uses the same model as the electric and gas tank water heaters

(28), but for solar water heaters a heat exchanger is also modeled. For the case of a gas water heater, the heat exchanger is modeled as an immersed helical heat exchanger in the bottom of the tank. For an electric water heater, the heat exchanger is modeled as a wrap around heat exchanger wrapped around the bottom half of the tank. The storage tank used here is an 80 gallon model, sized based on a rule of thumb for solar water heaters to provide 20 gallons per occupant, which would correspond to a 3 bedroom home (21). The actual tank volume is assumed to be 10% less than the nominal volume to be consistent with tank sizing regulations for electric water heaters. It is assumed that since most manufacturers offer a tank with backup electric resistance element for use in a single tank system, the actual size would be consistent with electric water heater sizing (which is typically 10% less than the nominal volume) as opposed to gas water heater sizing (which is 5% less than the nominal volume). The dip tube shown in Figure 13 is not included in the model. Instead, all flow is assumed to come into the bottom of the storage tank. The tank overall heat loss coefficient is taken from a study of a different solar water heater (40) which determined the overall heat transfer coefficient of the particular storage tank while installed in the field. The R value from that tank is applied to this water heater to get a comparable overall heat transfer coefficient while taking into account differences in storage volume (and therefore surface area) between the two storage tanks.

For all solar water heaters, 50 feet of copper piping is assumed to connect the solar water heater to the storage tank, split evenly between going into the collector and leaving the collector. This piping assumed to have 3/4" thick pipe insulation with an R value per inch of 3.97 ft^2-hr-°F/Btu-in along its entire length, the minimum insulation required for the piping to and from the collector (37). The heat loss from these pipes is calculated in the same way as heat loss for the DHW distribution system as described in Chapter 6 of this thesis. The pipe specifications used here are consistent with SRCC guidelines for solar water heating systems (37). The first 20 feet of pipe entering and leaving the storage tank are generally assumed to be in the same location as the water heater, while the remaining 5 feet are assumed to be outside. In the case of solar water heaters in a basement, the first 20 feet are assumed to be in conditioned space as the pipes still have to reach the roof so the majority of their length will be in conditioned space. This allows the first 20 feet to interact with the space the water heater is located in as all heat loss from this section will go the space, influencing the heating and cooling loads.

The pump used for this system has a maximum power draw of 0.04 horsepower (29.8 W) and a maximum flow rate of 2 gallons/minute. 5% of the energy used by this pump is assumed to be turned to heat, which is transferred to the fluid flowing through the pump. The pump is controlled by a differential temperature controller which turns on if the temperature at the collector outlet is 10°C higher than the temperature of water in the storage tank at the same node as the heat exchanger outlet. The pump turns off if the temperature in the tank is 2°C lower than the temperature at the collector outlet. Pump energy consumption was taken into account when considering the overall efficiency of the solar water heating system.

Chapter 4: Heat Pump Water Heater Model

Extensive laboratory testing of several different heat pump water heaters (HPWHs) was performed at NREL (41). Lab testing involved creating performance curves (see Figure 15), operating mode tests, tests with the fan partially blocked to determine the impact of reduced airflow, DOE standard rating tests (Energy Factor and First Hour Rating), and draw profile tests. The results of these tests were used to validate the HPWH model created here. It was found that the existing HPWH model in TRNSYS was not able to fully capture the behavior of this unit, so modifications were made to create a model that could better simulate the actual performance of these units. The model described here is based on only one of the units (see Figure 16) that was tested so that validation could be done of the model. The chosen unit has a rated efficiency (Energy Factor) of 2.35 and features a 700 W compressor as well as two 4.5 kW electric resistance elements. There are differences in the controls, condenser design, choice of refrigerant, and other factors between the different units tested. The unit modeled here, while fairly typical of the HPWHs currently available on the market, cannot be considered representative of every HPWH on the market due to these differences.

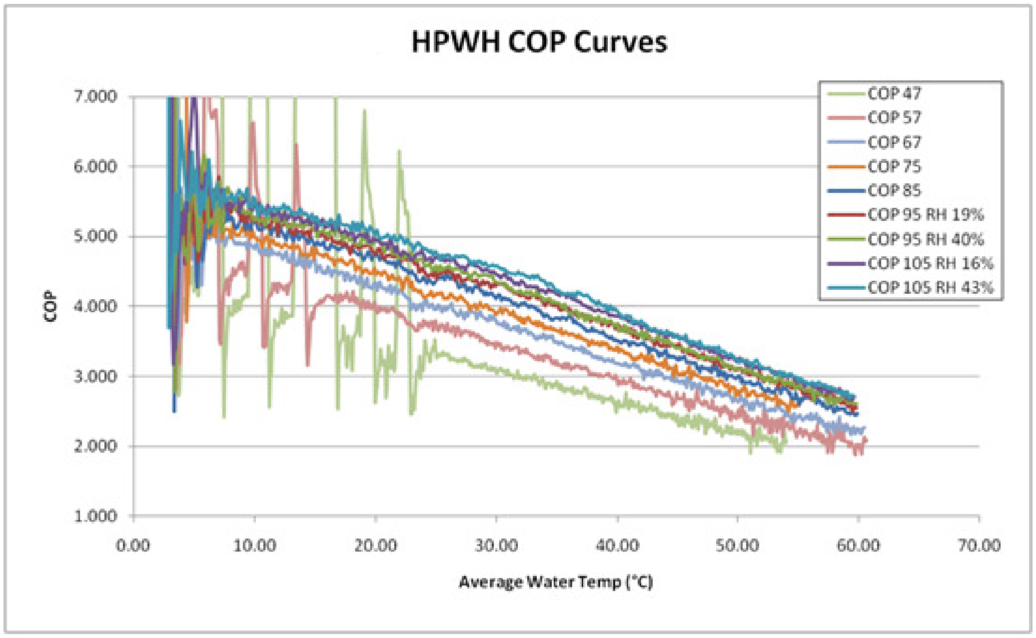


Figure 15: COP curves for the tested HPWH (41)

Figure 16: The modeled water heater during lab testing (41)

4.1 Modifications to the Existing Heat Pump Water Heater Model

The new HPWH model developed here was based on an existing HPWH model (42), combined with a stratified storage tank model previously described in Section 3.1. The existing model is a performance map based model that requires the compressor power, total and sensible cooling rate of incoming air, and rate of heat rejection from the heat pump to the water as a function of water temperature, ambient air temperature, and ambient humidity. The existing model assumes that a fluid (typically water) is being pumped through the unit continuously from either the storage tank or the condenser and that all heat rejected by the unit goes into that fluid. The model also assumes that the HPWH has a single speed fan.

The HPWH model described here is a modification of the existing HPWH model. It used a similar performance map to the existing HPWH, but only mapped the performance to the wet bulb temperature and the tank temperature adjacent to the condenser instead of mapping to both dry bulb temperature and humidity. This modification was done because the lab testing did not fully explore the impact of humidity and primarily looked at the impact of wet bulb temperature. The performance map used here takes a list of points of the HPWH's performance at different water temperatures and ambient wet bulb temperatures and linearly interpolates between these points. To develop this performance map, the COP curves shown in Figure 15 were divided into a series of discreet points. The average heat pump performance was calculated for every 5 °C change in water temperature. A schematic of this model is give in Figure 17. The performance map of COP as a function of water temperature and wet bulb temperature is shown in Figure 18.

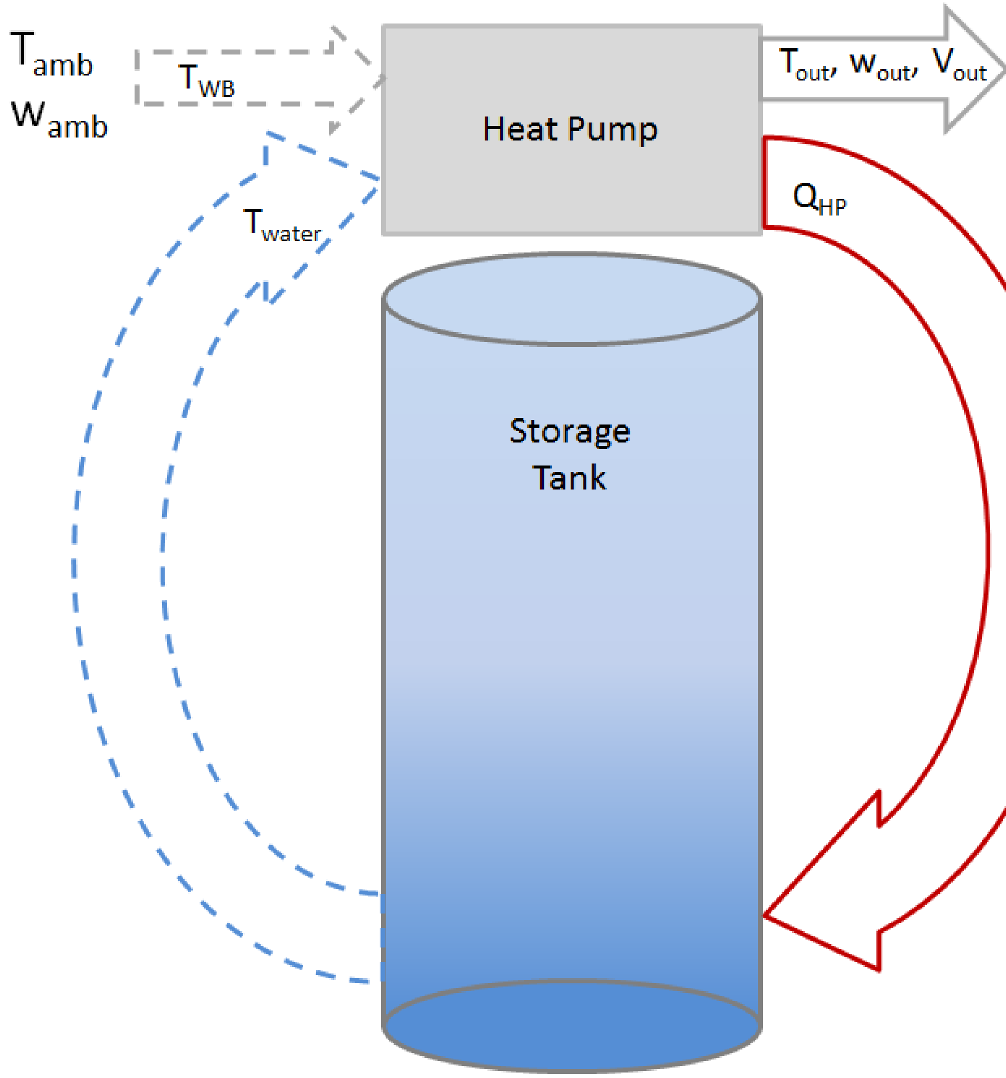


Figure 17: Schematic of the HPWH model

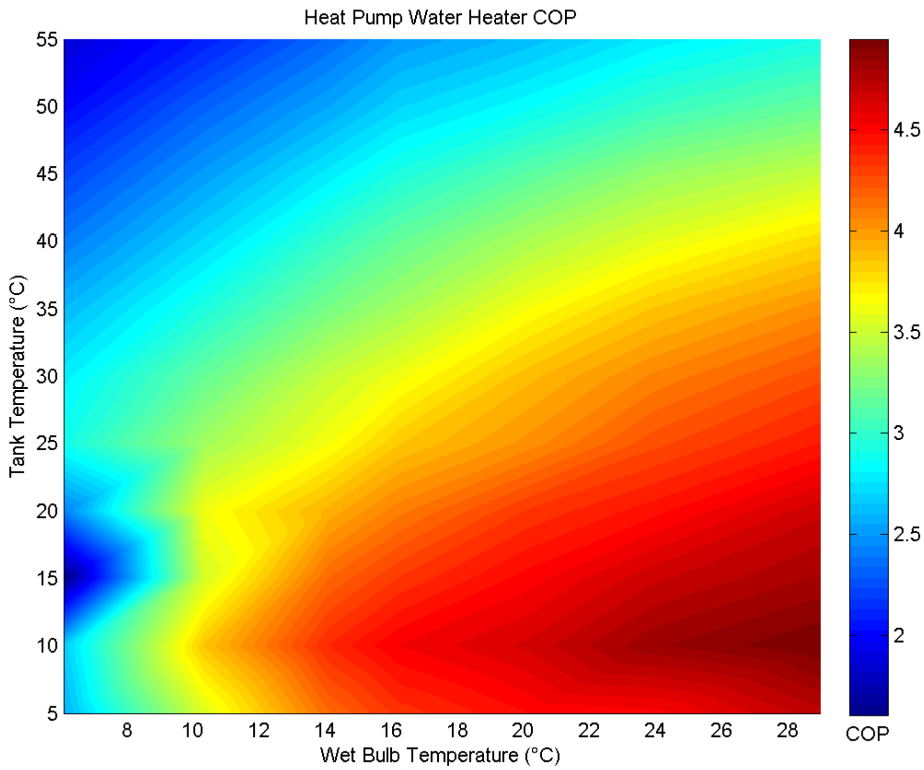


Figure 18: HPWH COP performance map

This HPWH can vary the fan speed based on ambient air conditions and tank temperature. Fan speed was added to the performance map, but fan power could not be added to the performance map because of limitations of the TRNSYS framework (only 5 variables can be the output of the performance map). Based on the COP testing, a correlation between fan speed and fan power was developed. This correlation was used to determine the fan power based on the fan speed calculated from the performance map. This correlation was "hard coded" into the model and would need to be changed to model other HPWHs. This correlation is shown in Figure 19.

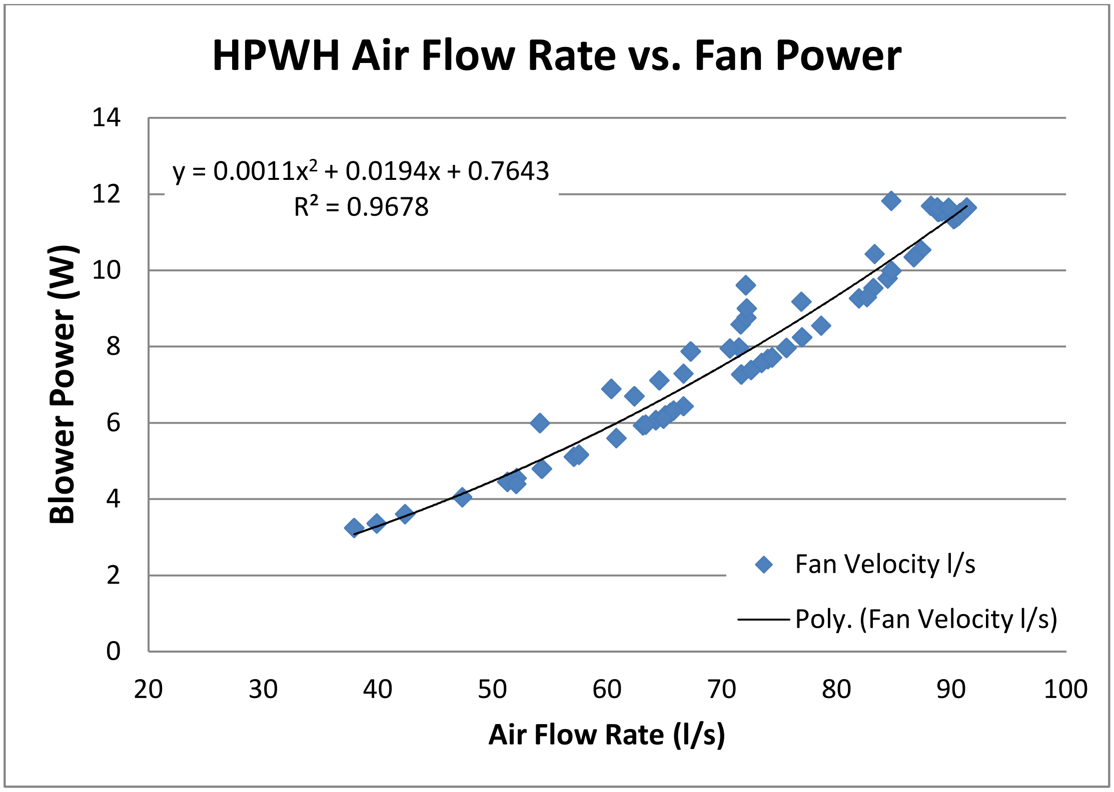


Figure 19: Relationship between fan speed and power for the modeled HPWH

The actual HPWH that was tested does not have water pumped through it, but instead has the condenser wrapped around the tank (although there are some HPWHs on the market that do have water pumped through the heat pump, they are not the typical design). To simulate this, all the heat that would have been transferred to the fluid flowing through the unit was instead added to the tank. The heat was distributed through all the nodes that were thought to be adjacent to the condenser. The exact condenser location could not be determined during testing and was considered proprietary by the manufacturer, so the location used here is approximate. The heat was distributed evenly into the tank across the nodes adjacent to the condenser. In reality, slightly more heat should be transferred to water at the condenser inlet than the outlet as the refrigerant (R-134a in this case) is warmer at the inlet. However, for this HPWH it is unknown whether the condenser inlet is at the top or bottom of the condenser. The distribution of heat is shown in Figure 20.

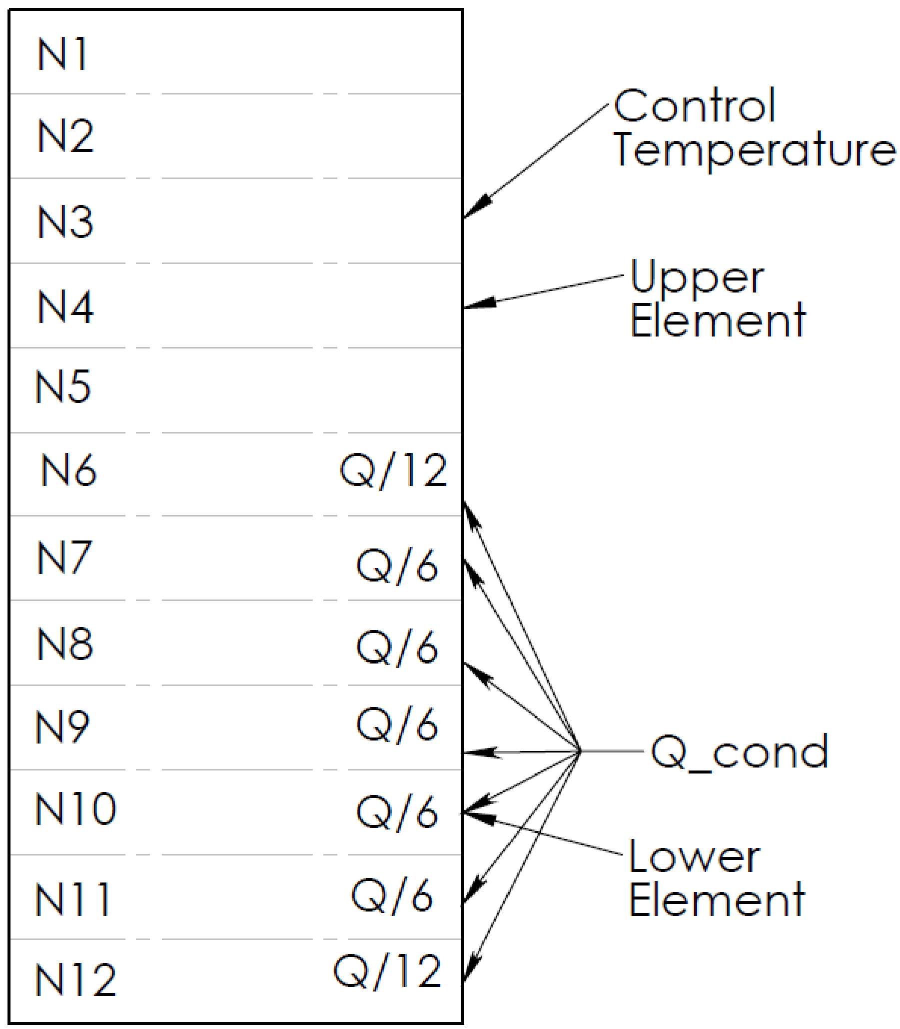


Figure 20: Distribution of heat from the heat pump to the storage tank

4.2 Derivation of Model Parameters

Several parameters for this model needed to be derived from the lab testing. In particular, the overall heat transfer coefficient (UA) of the tank needed to be derived. This parameter was derived from the Energy Factor test performed at NREL (41). This test features six successive draws spaced at one an hour apart, followed by 18 hours of standby in an attempt to model a homeowner's water use. This standby period was used to derive the UA value of the tank.

As part of the calculations to determine the Energy Factor of the tank, the UA value of the tank is calculated. In the standard test procedure, the equations necessary to calculate UA are:

$$UA = \left(\frac{Q_{stby} - \frac{V_{st}\rho C_p(\bar{T}_{24} - \bar{T}_{su})}{\eta_r}}{\tau_{stby}} \right) \Big/ \left(\bar{T}_{t,stby} - \bar{T}_{a,stby} \right) \tag{6}$$

$$\eta_r = \frac{M_1 C_p\left(\bar{T}_{del,1} - \bar{T}_{in,1}\right)}{Q_r} + \frac{V_{st}\rho C_p(\bar{T}_{max} - \bar{T}_o)}{Q_r} \tag{7}$$

Variable	Definition
Q_{stby}	Energy consumed during standby
V_{st}	Measured tank volume
ρ	Density of water
C_p	Specific heat of water
$\bar{T}_{24}$	Average tank temperature at the end of the test
$\bar{T}_{su}$	Maximum average tank temperature after cut-out
η_r	Recovery efficiency
τ_{stdby}	Standby period
$\bar{T}_{t,stby}$	Average tank temperature during standby
$\bar{T}_{a,stby}$	Average ambient temperature during standby
M_1	Mass of water drawn during first draw
$\bar{T}_{del,1}$	Average delivered temperature during first draw
$\bar{T}_{in,1}$	Average inlet temperature during first draw
Q_r	Energy consumed during first draw
$\bar{T}_{max}$	Maximum average tank temperature after first draw
$\bar{T}_o$	Average tank temperature at the start of the test

Table 4: Definition of variables used in equations 6-7

However, these equations are inaccurate for a HPWH. The recovery efficiency term should not be part of the calculation in equation 6. Instead, the overall heat transfer coefficient was calculated based on the period of standby between recovery from the last draw and the first time the heat pump needed to turn on during standby. It was assumed that this tank is likely to have a slug of cold water below the heat pump, which is similar to what is seen in electric water heaters (see Section 3.2 on existing water heater models for more details on this phenomenon). As a result, the tank was divided into two separate isothermal nodes, one above the condenser at the average measured temperature and one below the condenser at a temperature midway between. The energy balance for this case can then be written as:

$$\rho V c_p\left(T_f - T_i\right) = \frac{UA}{\varphi}\left(\bar{T}_{tank} - T_{amb}\right)t \tag{8}$$

Where:

$$\varphi \equiv f_t + f_{bot}\left(\frac{T_{bot} - T_{amb}}{T_{top} - T_{amb}}\right) \tag{9}$$

In the laboratory testing, the average tank temperature during this period was 59.5 °C, the average ambient temperature was 20 °C, the average tank temperature dropped 2.76 °C over 6.88 hours. This gives an overall heat transfer coefficient of 4.10 kJ/hr-°C.

The tank volume was based on the measured volume, not the nominal 50 gallon volume. As was previously mentioned in Section 3.2, water heater manufacturers are allowed ±5% in the tank volume for gas water heaters and ±10% for electric water heaters. Most manufacturers size their tank to be just within the lower end of this range. For this HPWH, the water heater volume was found to be 45.6 gallons, about 10% less than the nominal volume**.** The controls for this unit were found to consume 3 W on average when no heat source was in use. This electrical energy consumption was added to the model to capture the actual energy use.

4.3 HPWH Controls

The controls for this unit were very complex and difficult to determine during testing. The exact controls are proprietary, so the controls used for this model were derived from the HPWH testing, in particular the draw profile testing, which was designed to simulate a typical home's hot water use. The controls presented here are for "hybrid mode", the factory default heating mode which tried to balance using the heat pump with the electric elements to provide both energy savings and comfort. This unit also has a heat pump only mode, an electric element only mode, and a "high demand" mode that uses the electric elements more frequently than hybrid mode. The majority of the lab testing focused on hybrid mode, although some testing was done of the unit in other operating modes. The testing of electric only mode was used to determine the controls for when the HPWH is subjected to ambient air conditions that prevent the heat pump from operating, but no attempt was made to model the controls of the other operating modes.

While the exact controls are proprietary, the manufacturer did provide some insight into how the controls for this unit work. The controller for this HPWH monitors the temperature near the third node in this model and attempts to determine how quickly water is being drawn by monitoring this one temperature, possibly with some form of derivative control. It is not possible to determine the exact controls used by the manufacturer, in particular to verify any derivative component in the controls, without their input. As a result, the controls used here are based on set point temperatures only. This approach worked well both with the lab testing data as shown in Figure 27 in Section 4.4 and with field test data as shown in Section 4.6.

. Several unique behaviors of this unit were observed during laboratory testing. First, only one heat source (the heat pump, upper electric element, or lower electric element) would turn on at a time. The heat pump would turn on first as long as the ambient air temperature was in range for the heat pump to operate, followed by the lower electric element if the heat pump could not meet demand. If the temperature at the thermistor continued to drop while the lower electric element was on, the upper electric element would turn on. Once one of the electric elements turned on, the elements would remain on until the set point temperature was reached. If the temperature dropped enough to turn on the upper element, the upper element would not bring the tank all the way back up to set point, but would bring the temperature to within a few degrees of the set point (for fast recovery), then switch to the lower element to provide the full capacity of the tank. The unit is assumed to operate in electric element only mode if the ambient air temperature is out of range for the heat pump to function. The control logic for this unit during normal operation is shown in Figure 21. The control logic for electric mode is shown in Figure 22.

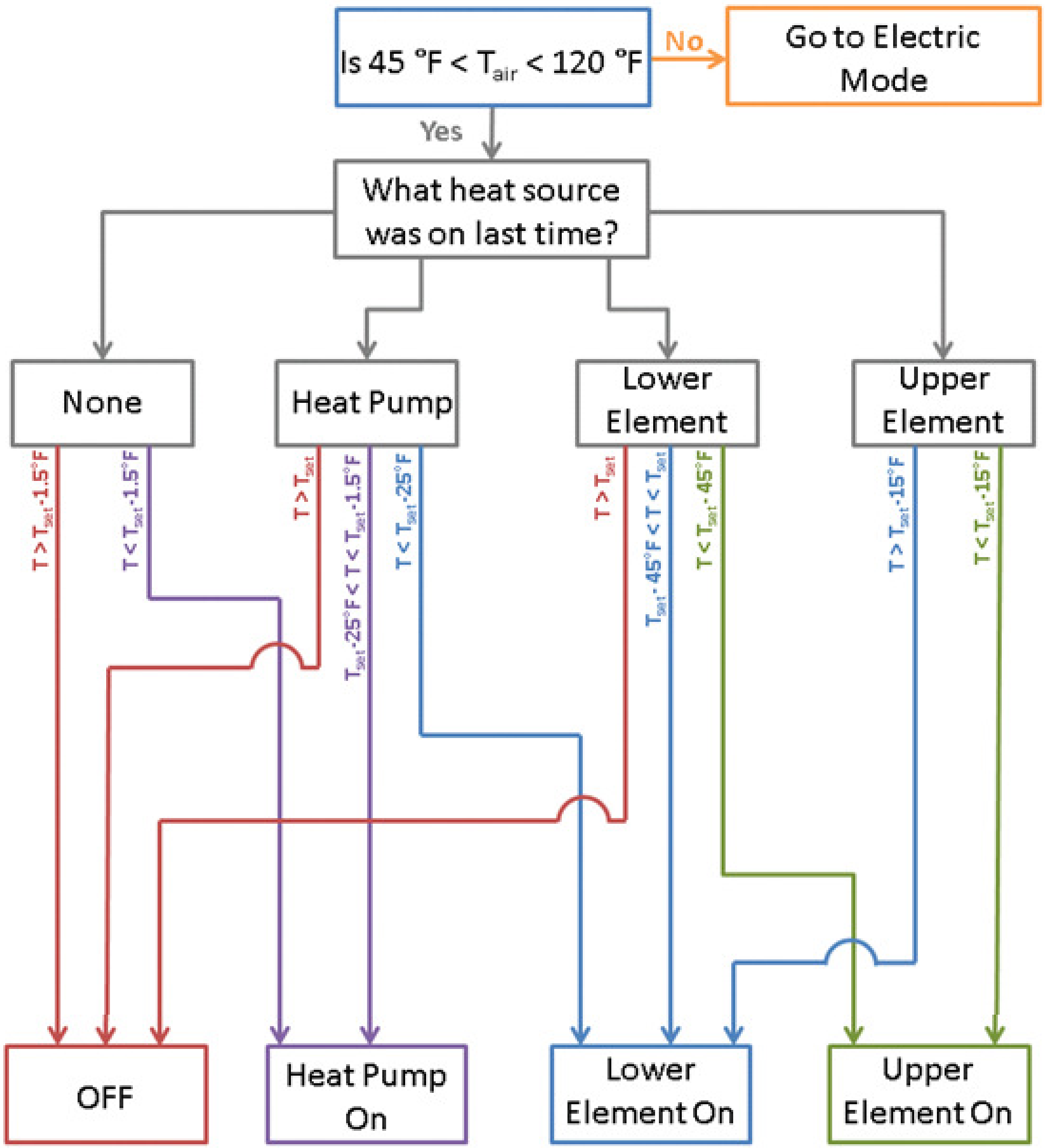


Figure 21: Control logic for the HPWH in hybrid mode

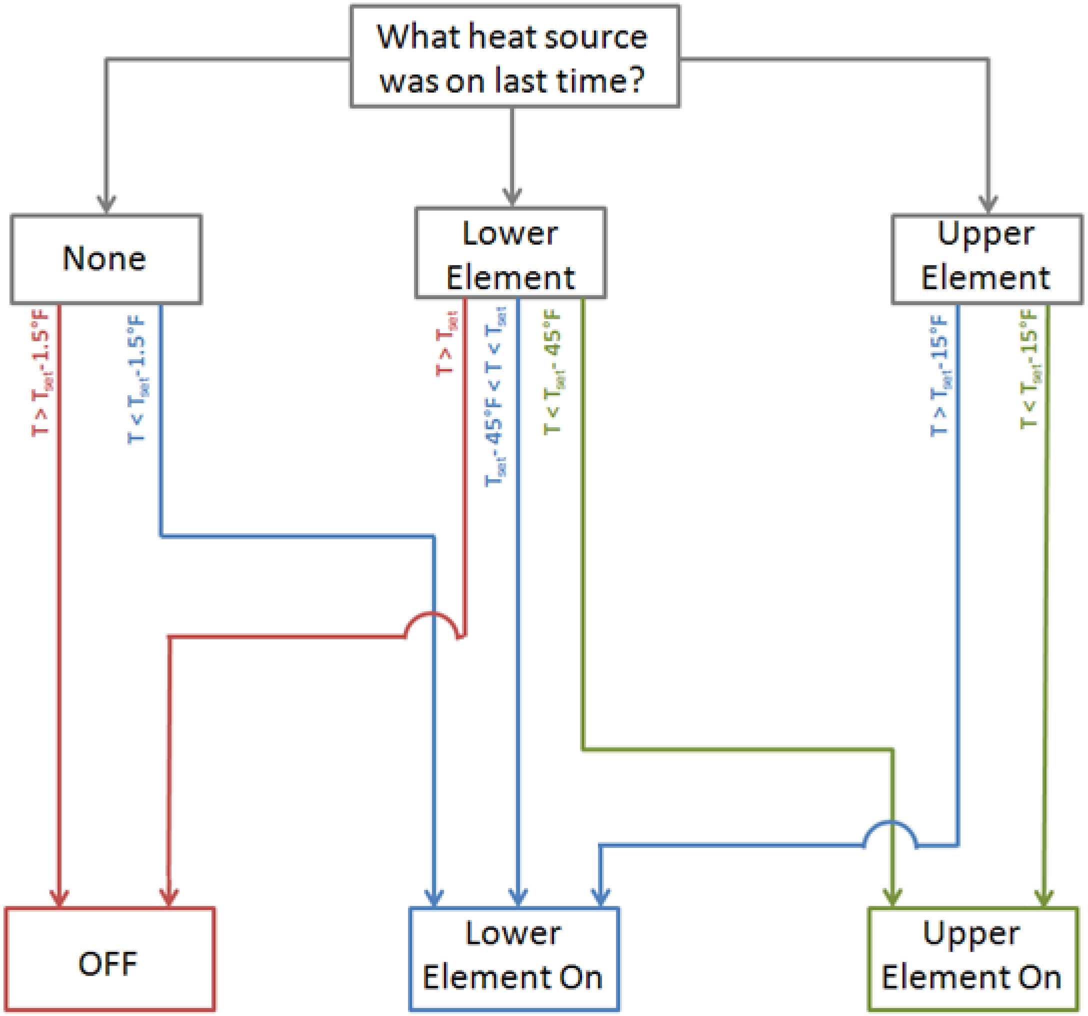


Figure 22: Control logic for the HPWH in hybrid mode if ambient air temperature is out of heat pump operating range or the HPWH is set to electric mode

4.4 Validation of the new HPWH model

To ensure that the performance map was working correctly, a simulation was performed on one of the COP tests (at a wet bulb temperature of 14.1 °C) and the simulation results were compared to the actual test results. The comparison of tank temperature is given in Figure 23. Most temperatures agreed very well, although there was a larger difference in the lowest measured temperature (T_6) than with any of the other temperatures. This is largely due to uncertainty in the exact location of the condenser: the sixth thermocouple was very close to the bottom of the condenser, which makes it difficult to determine exactly how much heat should be going to this location. In addition, since the model had 12 nodes, the measured temperature was compared to the average temperature of the two adjacent nodes. This causes some error when looking at the lowest temperature as the area below the condenser usually has a steep thermocline and average temperatures cannot accurately capture this phenomenon.

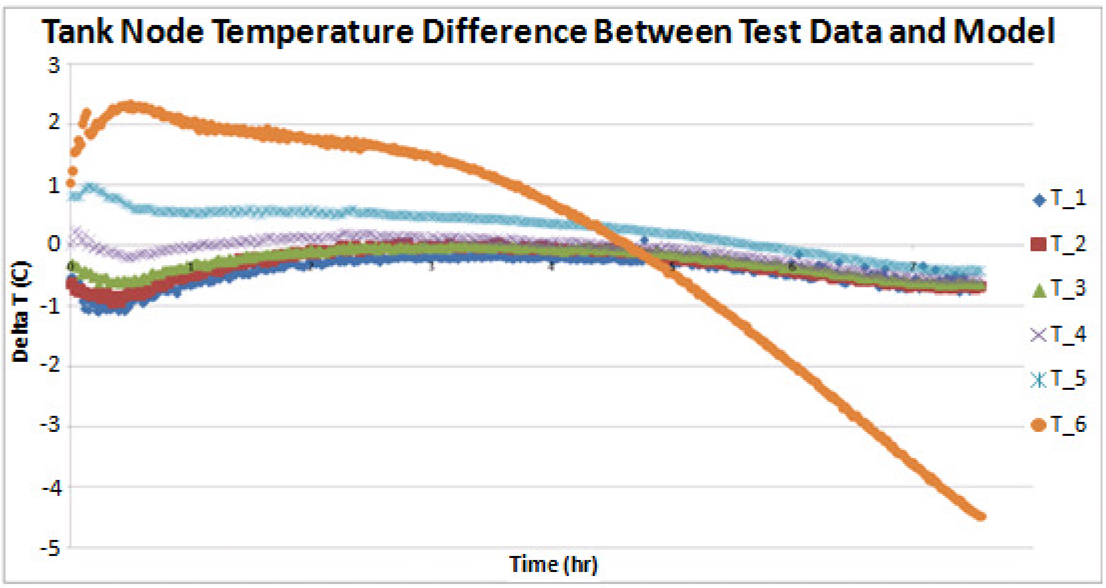


Figure 23: Comparison of modeled and measured tank temperature for the HPWH

The HPWH outlet air temperature and humidity ratio comparison between the model and the measured data is shown in Figure 24. Both of these quantities closely matched the measured values and were largely within the accuracy of the measurements (±0.5 °C for air temperature, humidity was calculated from temperature and dew point measured via a chilled mirror hygrometer with an accuracy of ±0.3 °C). There were some differences in the beginning as the modeling and testing was intended to capture steady state operation and does not fully capture variations in the heat pump performance that occur during start up.

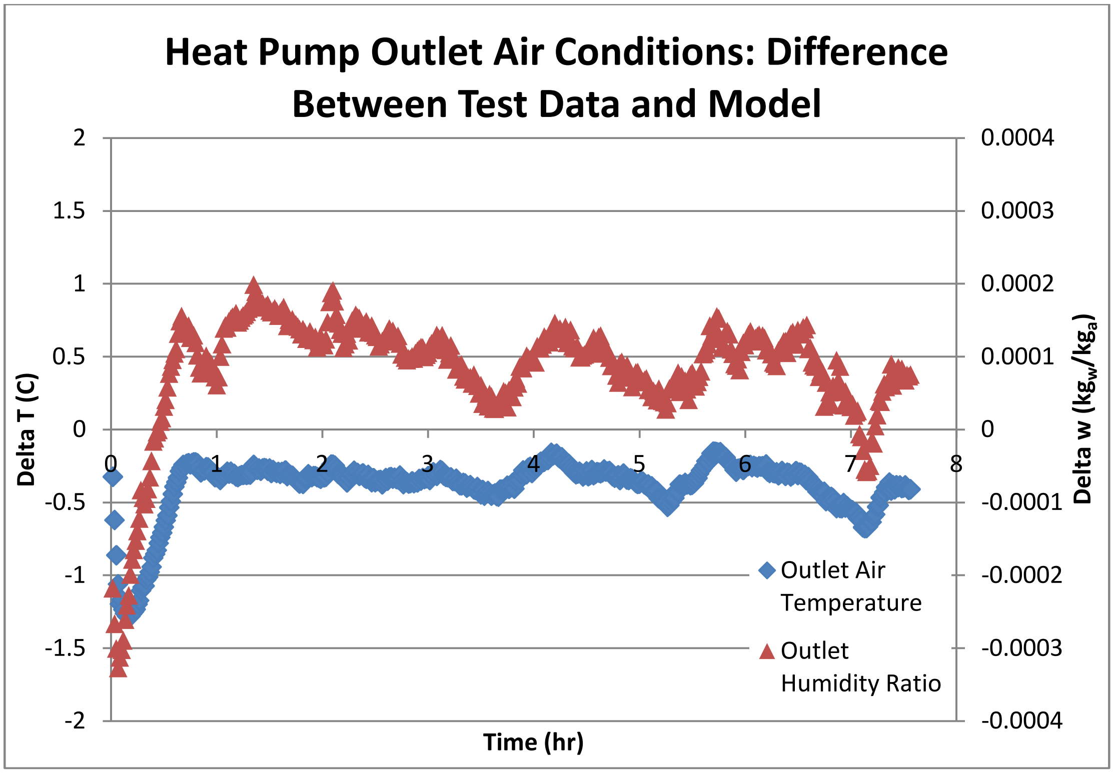


Figure 24: Comparison of modeled and measured outlet air temperature and humidity for the HPWH

A sensitivity analysis was done to ensure that the 12 node tank model shown in Figure 23 was sufficient to capture the stratification of the tank observed during testing. An 18 node and 24 node tank were also simulated and the temperature of the tank was compared to the measured tank temperature as shown in Figure 25 and Figure 26. It was impossible to get the same heat distribution in the 18 node tank as the 12 or 24 node tank due to the fact that in an 18 node tank what was originally 2 nodes became 3 nodes, which caused some differences. Overall, the difference between the models was largely contained in the bottom node, which had larger uncertianty than any other nodes because of the uncertianty in the exact location of the condenser. It was therefore determined that a 12 node tank was sufficient to accurately capture the stratification seen during testing. The root mean squared error in the temperatures for each tank model are given in Table 5.

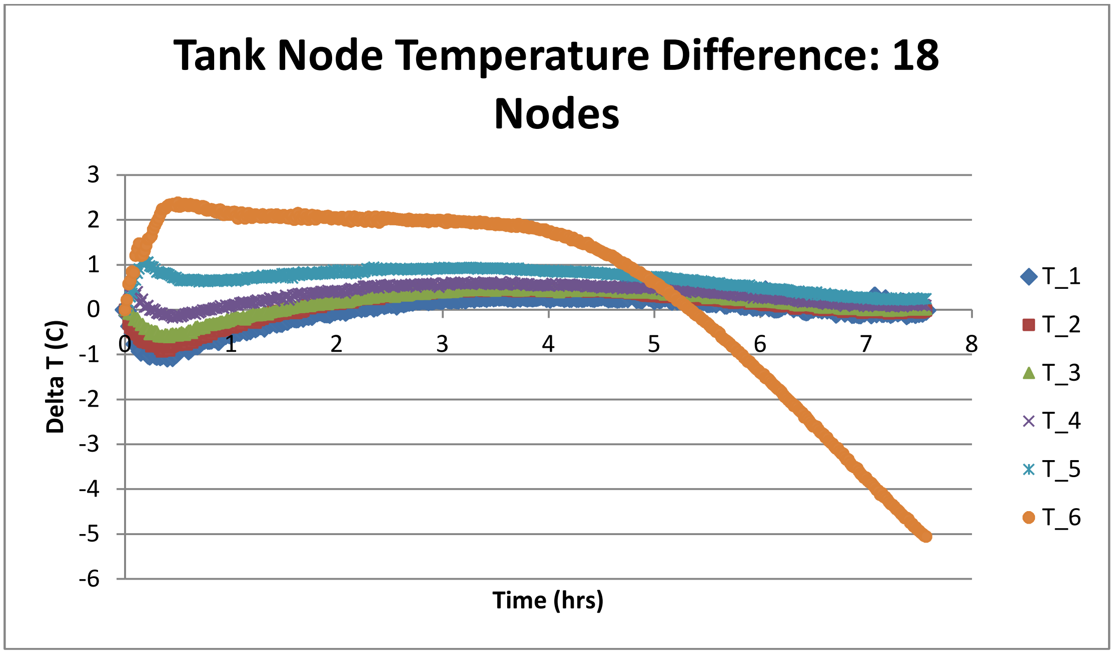


Figure 25: Difference in measured and modeled tank node temperature for a 18 node tank model

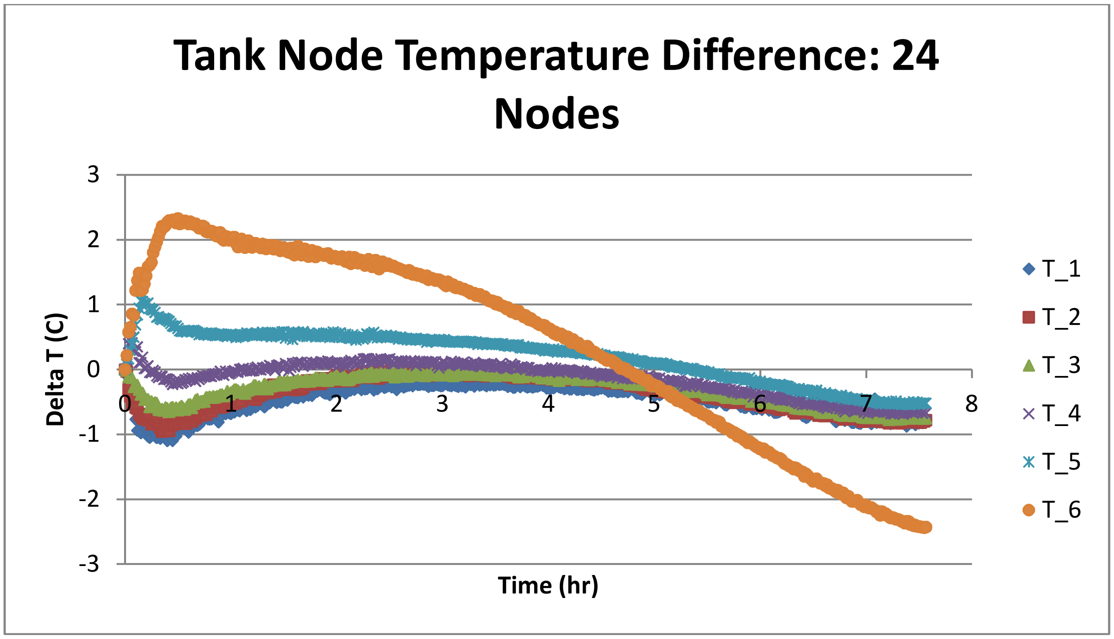


Figure 26: Difference in measured and modeled tank node temperature for a 24 node tank model

	T1	T2	T3	T4	T5	T6	Total
12 Nodes	0.0323	0.0288	0.0256	0.0203	0.0207	0.1617	0.0896
18 Nodes	0.0167	0.0170	0.0152	0.0185	0.0338	0.1030	0.0753
24 Nodes	0.0250	0.0219	0.0183	0.0138	0.0207	0.0711	0.0689

Table 5: Root mean squared error in tank temperature for multinode tank models

The controls (as described in the preceeding section) were also compared to the measure data as shown in Figure 27. This was done by running the model under the conditions of a draw profile test which was simulating typical water heater use. The modeled controls did a fairly good job of determining when the heat pump and different elements would turn on and off, with less than a 2% difference in the measured and modeled power consumption. In addition, the change from running on the heat pump, which consumes about 500 W during this test, to using the electric elements, which consume about 4500 W, was captured by the model. The change from using the upper element to the lower element (which occurs about 1 hour and 45 minutes into the test) was also captured by the model, as can be seen by the sudden jump in the lower element temperature and gradual decay in the upper element temperature at this time. The TRNSYS modeled temperature was generally lower than the measured temperature, especially for the temperature near the lower element. However, this difference in lower temperature would only affect very large draws that used the majority of the water in the tank before recovery could occur. There is a discrepency in the modeled upper temperature at about 2 hours into the test where the measured temperature dropped sharply. This could be due to mixing caused by conduction through the dip tube or a plume of warm water coming off of the lower electric element, neither of which was not included in the model. However, the exact cause of this sudden dip in temperature is unknown and was not fully explored here.

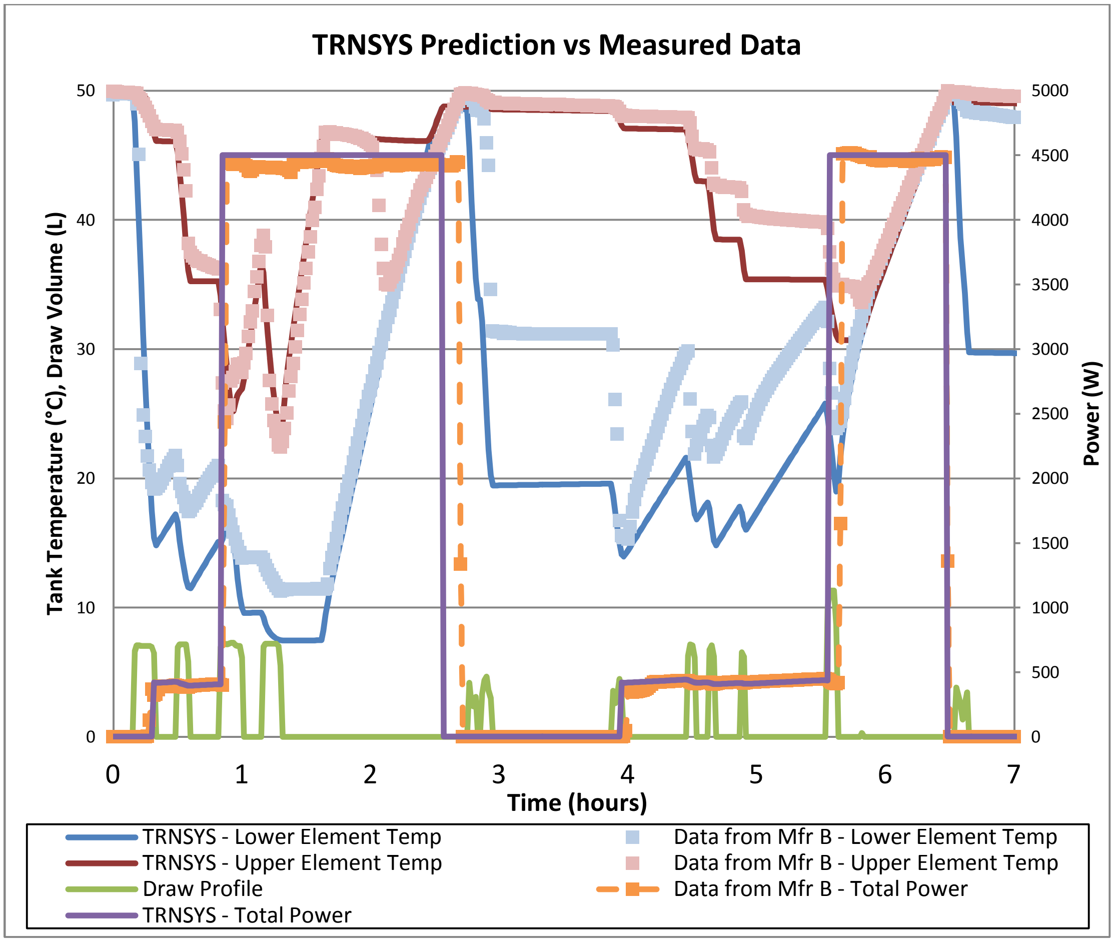


Figure 27: Comparison of HPWH model to the measured data for testing with a realistic draw profile

The controls used for electric only mode were also compared to measured data as shown in Figure 28. The model does not work as well in this case, with the modeled energy consumption being 13% larger than the measured energy consumpation. In particular, there as a heating event that occured in the first few minutes of testing that was not caputred in the model. The measured control temperature here is very close to the set point temperature (within the accuracy of the thermocouple used for this measurement), implying that the measured temperature is either not the same as the control temperature or that the controls are based on more than the temperature such as using the derivative of temperature. The model also did not alway exactly get the off and on times for the elements correct, which can be seen around hour 4 in Figure 28 below. However, these controls were considered sufficient as the water heater only rarely goes into electric mode in most locations when installed in unconditioned space and should never go into electric mode if it is installed in conditioned space unless a homeowner were to manually switch the unit's operating mode.

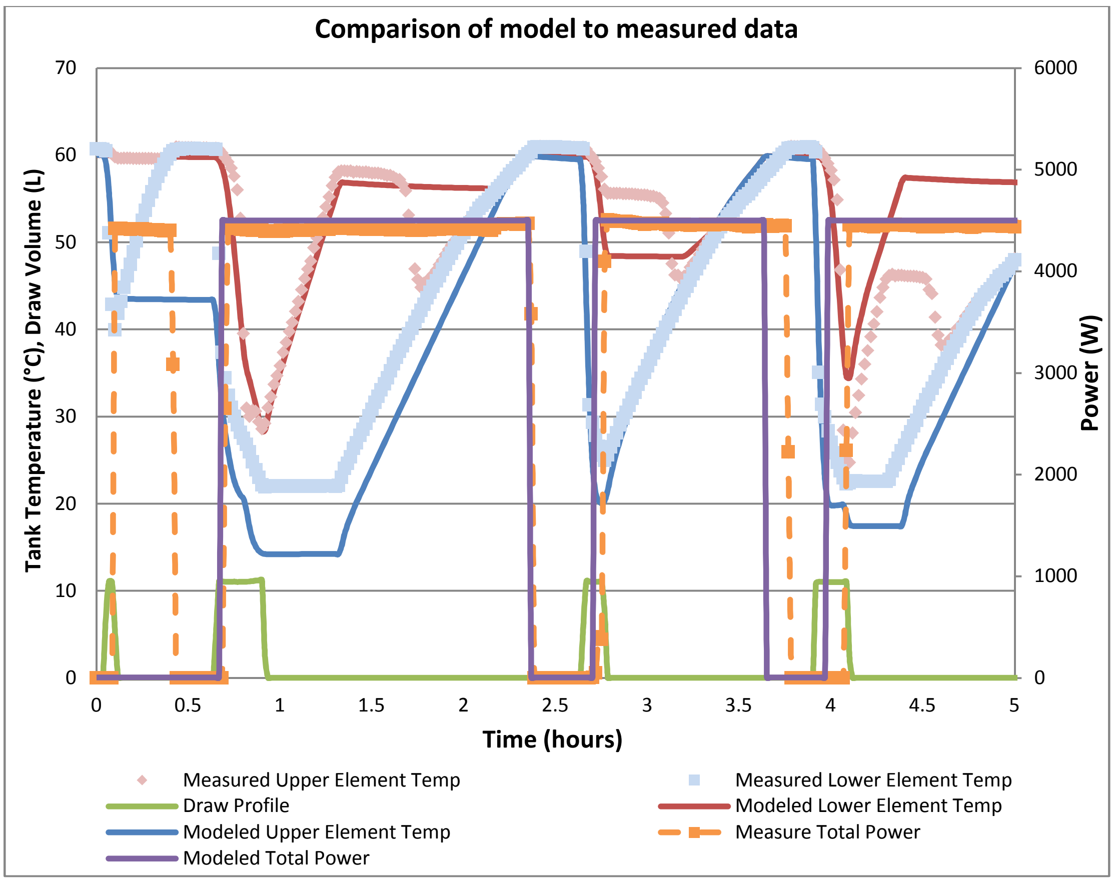


Figure 28: Comparison of HPWH model to the measured data for testing in electric mode with a realistic draw profile

4.5 Extrapolation of Performance Data

While the laboratory testing covered a wide range of conditions, it did not fully examine all of the conditions that could occur during simulations and actual operation. The testing covered a range of wet bulb temperatures between 6.1°C and 29 °C. However, in simulations it was found that the ambient wet bulb temperature could be outside of this range for fairly large portions of the simulations (up to 10% of the time). In TRNSYS, no extrapolation is performed by default: if the model is called when the conditions are outside of those provided by the performance map, performance is simply the same as it would be if the model was operating at the closest rated conditions. To deal with this issue, extrapolation was performed on the data and a performance map featuring this extrapolation was used in TRNSYS.

To cover the full range of conditions that are likely to be encountered during simulations, the performance map was expanded to cover all wet bulb temperatures between 0 °C and 35 °C. Extrapolation must be performed on all values which are retrieved from the performance map. For the heat pump water heater model, this is the total cooling, sensible cooling, compressor power, heat rejected to the tank, and the fan power. In general, a quadratic provided a good fit to the available data and was used for extrapolation. The data and associated regression for performance as a function of wet bulb temperature is provided at one tank temperature (10 °C) in Figure 29 through Figure 33.

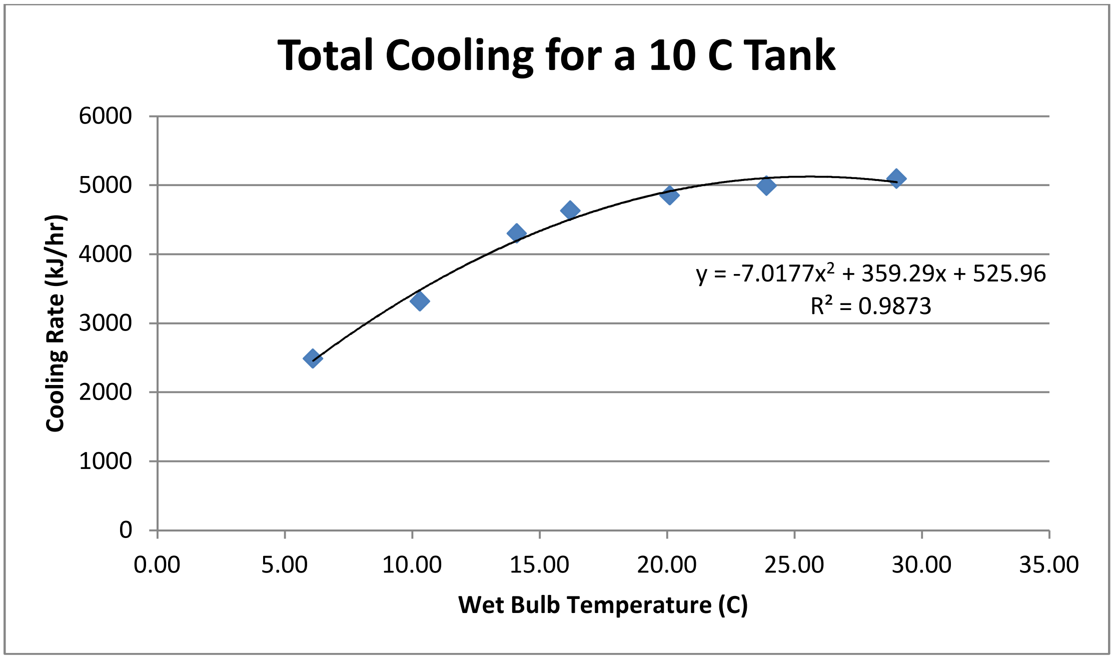


Figure 29: Total cooling rates for a 10 °C tank

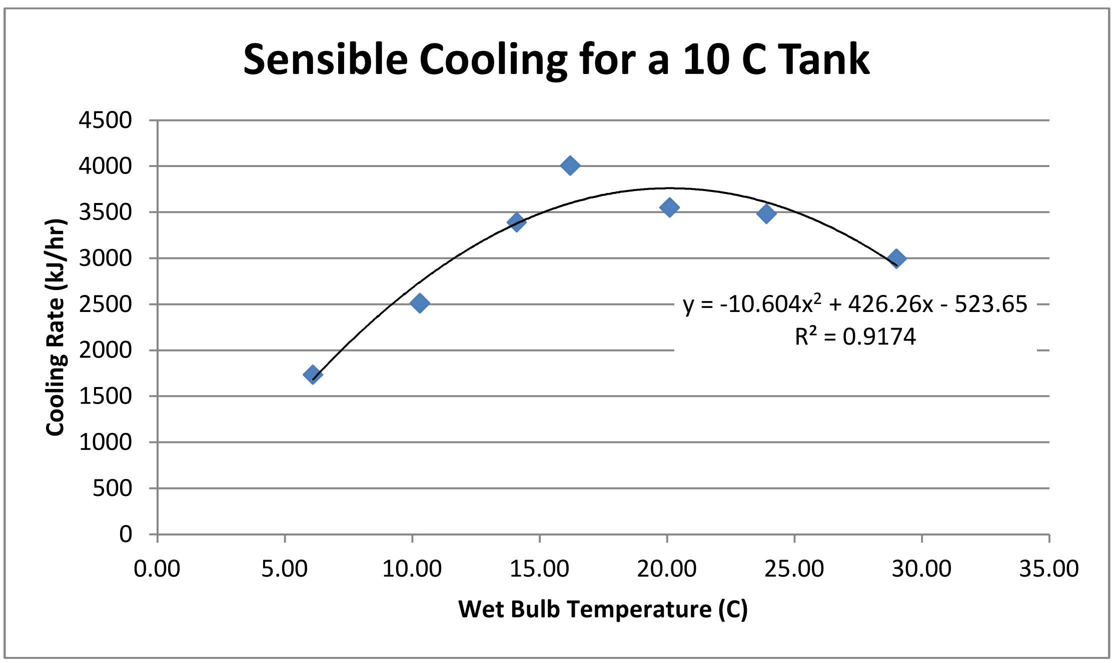


Figure 30: Sensible cooling rates for a 10 °C tank

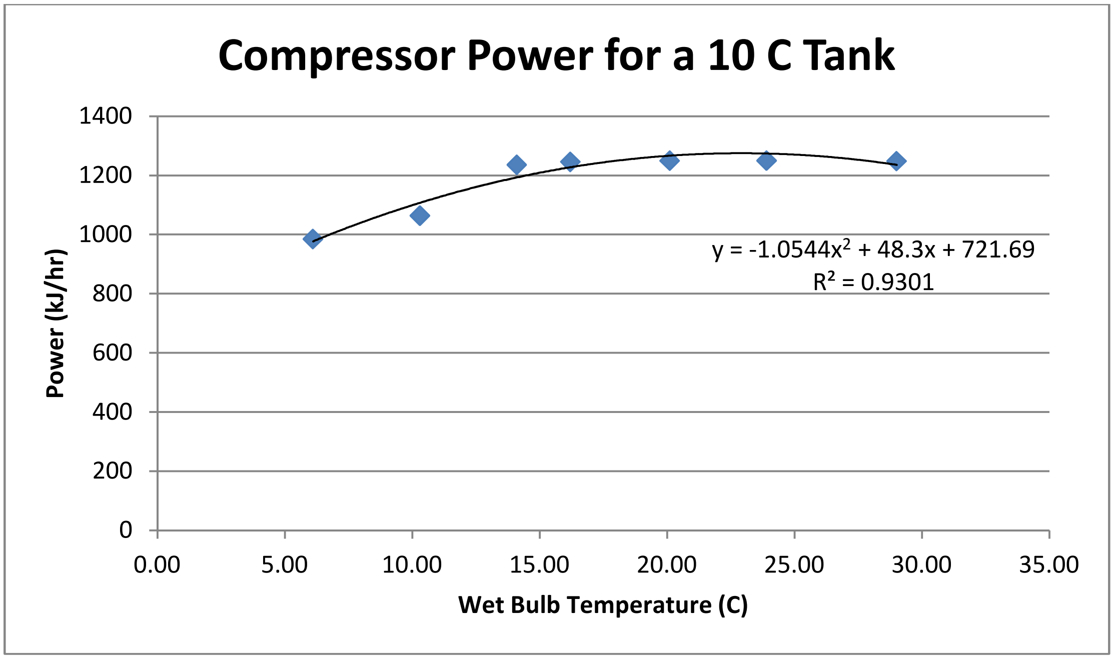


Figure 31: Compressor power for a 10 °C tank

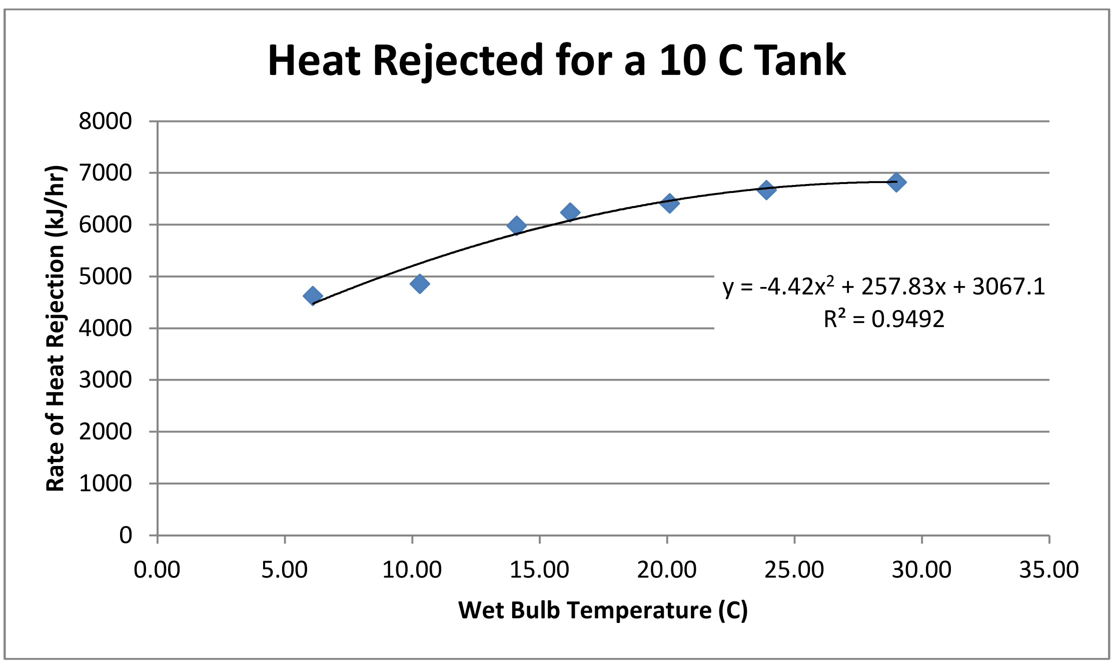


Figure 32: Heat rejected for a 10 °C tank

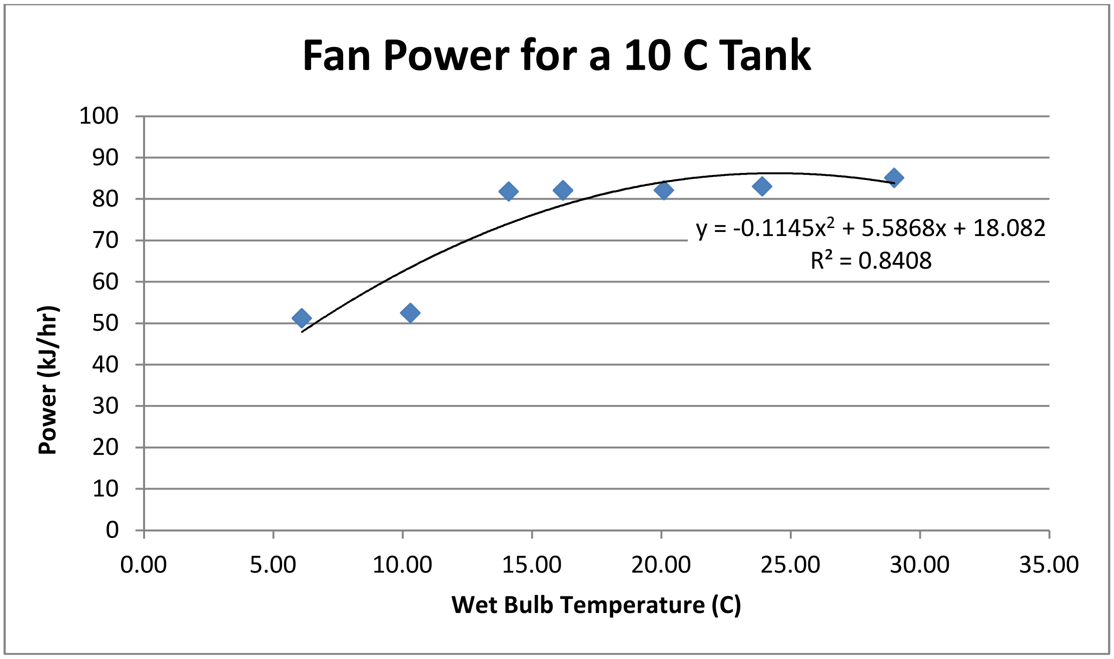


Figure 33: Fan power for a 10 °C tank

One problem arises from using a curve fit as can be seen in Figure 30: the sensible cooling rate can go to zero or even negative if just the regression analysis is used. To compensate for this, in cases where extrapolation was performed on the sensible cooling rate below the conditions explored during lab testing, the sensible heat ratio (SHR, defined as the sensible heat removed divided by the total heat removed by the heat pump) was kept constant at whatever SHR was observed during the test with the lowest wet bulb temperature. The sensible cooling curve is used by TRNSYS primarily to determine the SHR and does not impact the COP of the heat pump during operation. These conditions can also only occur when the water heater is placed in unconditioned space, where it has less of an impact on the building's heating and cooling loads, so this assumption has a minimal impact on annual simulation results.

The coefficient of determination for each regression is proved in Table 6. In general, a good fit is provided by this quadratic regression. The average coefficient of determination is

0.9307 for all regressions. While this extrapolation is considered to be sufficient for this work, ideally future lab testing could determine the performance of the HPWH in the conditions for which extrapolation was required.

Tank Temp	Total Cooling	Sensible Cooling	Compressor Power	Heat Rejected	Fan Power
5	$R^2 = 0.988$	$R^2 = 0.9751$	$R^2 = 0.8962$	$R^2 = 0.9653$	$R^2 = 0.8543$
10	$R^2 = 0.9873$	$R^2 = 0.9174$	$R^2 = 0.9301$	$R^2 = 0.9492$	$R^2 = 0.8408$
15	$R^2 = 0.9893$	$R^2 = 0.9485$	$R^2 = 0.9646$	$R^2 = 0.3936$	$R^2 = 0.9253$
20	$R^2 = 0.9973$	$R^2 = 0.9435$	$R^2 = 0.9886$	$R^2 = 0.9968$	$R^2 = 0.7395$
25	$R^2 = 0.9936$	$R^2 = 0.9431$	$R^2 = 0.9956$	$R^2 = 0.9986$	$R^2 = 0.7363$
30	$R^2 = 0.9941$	$R^2 = 0.9561$	$R^2 = 0.9917$	$R^2 = 0.998$	$R^2 = 0.3977$
35	$R^2 = 0.9841$	$R^2 = 0.9289$	$R^2 = 0.9989$	$R^2 = 0.999$	$R^2 = 0.5476$
40	$R^2 = 0.9842$	$R^2 = 0.9317$	$R^2 = 0.9977$	$R^2 = 0.9981$	$R^2 =0. 9153$
45	$R^2 = 0.9832$	$R^2 = 0.9557$	$R^2 = 0.9985$	$R^2 = 0.9982$	$R^2 = 0.9943$
50	$R^2 = 0.9814$	$R^2 = 0.9589$	$R^2 = 0.997$	$R^2 = 0.9973$	$R^2 = 0.9509$
55	$R^2 = 0.9689$	$R^2 = 0.9601$	$R^2 = 0.9988$	$R^2 = 0.9961$	$R^2 = 0.9705$

Table 6: Coefficients of determination for HPWH performance map extrapolation

4.6 Comparison to Field Test Data

As a further validation of the HPWH model presented here, a comparison was performed between the model and field test data. Data was gathered by Steven Winter Associates from 14 different sites which have a HPWH installed (43). The sites are all located in New England in either Massachusetts or Rhode Island. Data was available for most sites from late November of 2010 until August of 2011 and was gathered in 15 minute intervals. The field monitoring gathered inlet water and air temperature, outlet water temperature, ambient relative humidity, and electricity consumption by the heat pump, elements, and whole water heater. Of these 14 sites, 10 had the same brand of HPWH as was modeled in this work. Only these sites are compared to the model as several factors, such as the heat pump performance, controls, and tank losses are specific to this particular brand of HPWH and not generally applicable to all of the models of

HPWHs currently available. A summary table of the sites covered is provided in Table 7.

Site	HPWH	Set Point Temp (°F)	Average Air Temp (°F)	Average %RH	System COP	Electric HPF
1	AO Smith 80 gallon	120	-	-	-	-
2	Stiebel Eltron	140	-	-	-	-
3	General Electric	125	64.1	37.9%	1.672	0.479
4	AO Smith 60 gallon	120	-	-	-	-
5	General Electric	129	51.8	62.1%	0.942	0.242
6	General Electric	122	61.0	52.6%	2.102	0.939
7	General Electric	125	65.3	47.1%	1.845	0.891
8	General Electric	125	64.8	44.7%	2.018	0.805
9	General Electric	120	61.4	48.8%	1.985	0.752
10	Stiebel Eltron	140	-	-	-	-
11	General Electric	140	76.6	32.3%	1.698	0.454
12	General Electric	130	71.4	44.1%	2.102	0.791
13	General Electric	130	68.1	57.7%	1.422	0.829
14	General Electric	120	60.7	51.0%	1.850	0.797

Table 7: Summary of HPWH field test data

Of note in Table 7 is that the heat pump fraction (HPF) is the electric heat pump fraction. In the simulations performed here, the heat pump fraction is usually defined as the amount of heat that comes from the heat pump to the tank divided by the total amount of heat added to the tank by both the heat pump and the elements. This definition makes HPF analogous to solar fraction (SF) for solar water heaters and provides some general information about how efficient a HPWH may be. In this field test data, the electricity consumption of the heat pump and the elements was monitored as opposed to the heat added to the tank by the heat pump. This definition means that the heat pump fraction from field test data is not the same as the thermal HPF. The electric HPF was calculated for all TRNSYS runs for validation of the field test data to provide an accurate comparison.

One other key factor of note is the set point temperatures of the different HPWHs. Most units (excluding the Stiebel Eltron) have a set point temperature that can be varied by the user. The default factory set point for both the AO Smith and General Electric HPWHs is 120 °F. However, the General Electric HPWH often has issues delivering water at its set point temperature due to poor controls and the fact that the heat pump can generally only supply heat at 2-2.5 kW while the electric resistance elements are 4.5 kW. For occupants, this means that users can either change the control strategy (the General Electric HPWH also has an all electric mode and a high demand mode, which uses the elements sooner than the factor default efficiency mode), change their behavior, or raise the set point temperature of the water heater so that the sag in outlet temperature does not drop below what would be a useful temperature. For example, consider a case where the outlet temperature has sagged 20 °F below its set point temperature, which can happen during normal operation. If an occupant is taking a shower and would like 105 °F water during their shower, they wouldn't be satisfied if the water heater was set to 120 °F, but would be satisfied if they set their water heater to 125 °F or above.

The data here suggest that this temperature sag issue will be dealt with by raising the set point temperature of the HPWH. However, this is only a small sample of homes in one region of the country where the HPWHs are likely installed by the same few plumbers. It is unknown if the set point temperature was adjusted by the plumber (either to reduce callbacks complaining about the unit or by the plumber making sure the new water heater has the same set point temperature as the old water heater) or the homeowners in response to thermal comfort issues. Future field test data and surveys of HPWH owners would needs to be performed and analyzed to determine what typical ways of dealing with this issue are. A further discussion of this outlet temperature sag, including how to account for it when comparing different types of water heaters, is included in the results section of this paper.

In this field testing, there were errors in the setup of the equipment at some sites that caused the power meter on the upper electric element to read very low power draws during all of the power draws. As all of the measurement equipment was located in a small enclosure, the most likely cause of this faulty power measurement is electromagnetic interference from some of the nearby equipment. These faulty measurements were removed from the analysis by assuming that any very small power draws (below 200 W) by the upper element are due to interference.

To compare the model to the field test data, simulations were run using the measured ambient air temperature, humidity, mains water temperature, and hot water flow rate as inputs and the calculated system COP and HPF for the entire test period was compared to the measured results. A table of the comparison of these performance metrics is provided in Table 8 while plots of both measured and calculated system COP and electric HPF are provided in Figure 34 and Figure 35 respectively. Plots of the daily system COP and HPF for each site are provided in

Appendix C.

Site	COP TRNSYS	COP Measured	HPF TRNSYS	HPF Measured	% Diff COP	%Diff HPF
3	1.931	1.672	0.715	0.479	15.5%	49.4%
5	1.328	0.942	0.452	0.242	41.0%	86.7%
6	2.093	2.135	0.955	0.938	-0.4%	1.7%
7	1.866	1.877	0.902	0.889	1.2%	1.2%
8	1.791	2.018	0.784	0.805	-11.3%	-2.6%
9	2.007	2.015	0.806	0.748	1.1%	7.2%
11	1.583	1.698	0.403	0.454	-6.7%	-11.4%
12	1.986	2.153	0.783	0.786	-5.5%	-1.1%
13	1.519	1.422	0.557	0.829	6.8%	-32.9%
14	2.042	1.850	0.934	0.797	10.3%	17.2%

Table 8: Comparison of measured and calculated HPWH performance

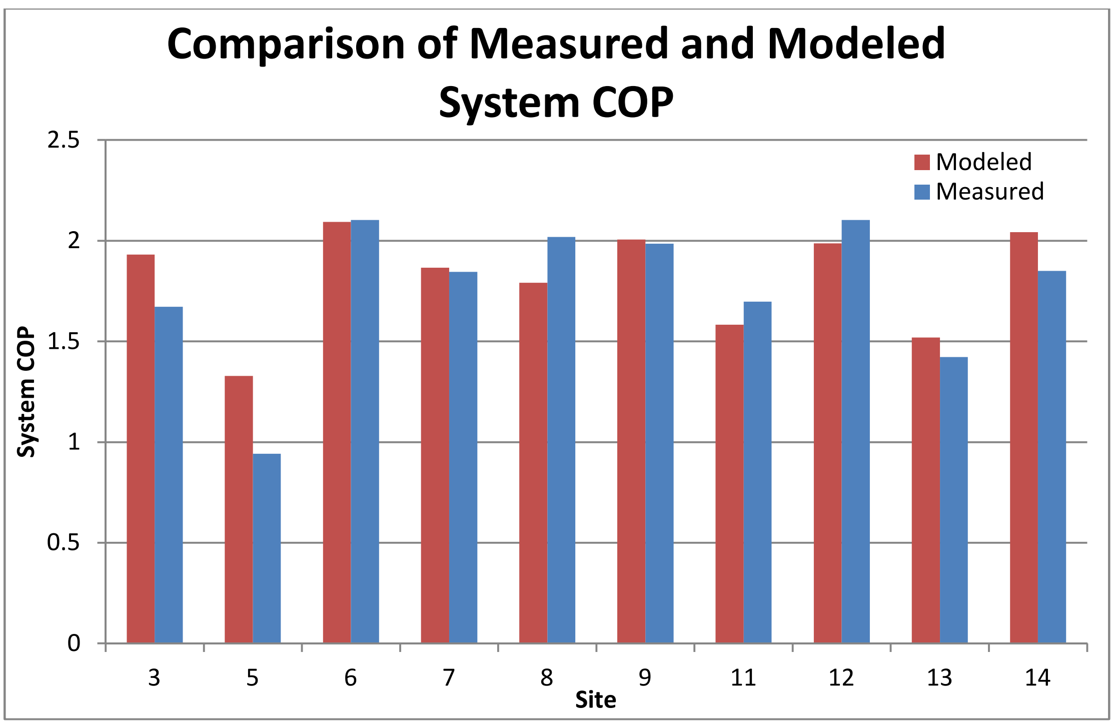


Figure 34: Comparison of calculated system COP to field data

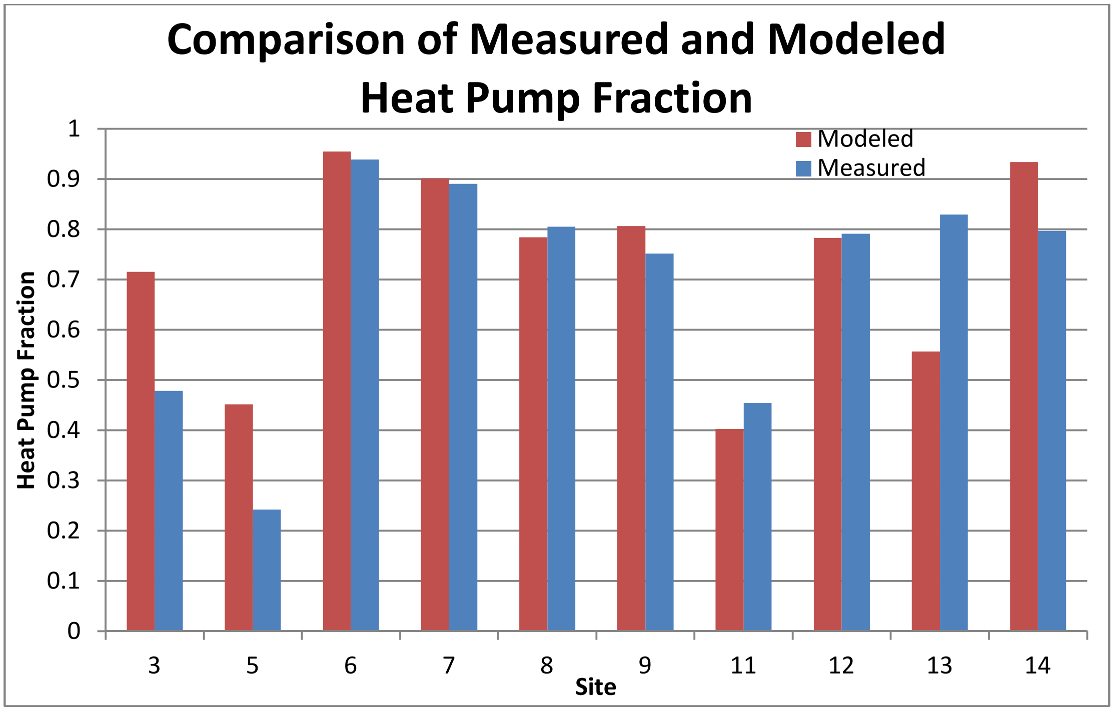


Figure 35: Comparison of calculated HPF to field data

Of these 10 sites, 3 had particularly large discrepancies between the measured and modeled performance: Sites 3, 5, and 13. At sites 3 and 5 the measured system COP is significantly lower than the modeled system COP, indicating that the model had significantly over predicted the energy savings that could be achieved by installing a HPWH at these locations. The most likely cause of this discrepancy is icing of the evaporator. Icing is not fully captured in the TRNSYS model as is discussed in the next section and in situations where icing occurs the model over predicts energy savings. To determine the impact icing may have on the results, the potential for icing and difference in system COP was compared on a daily basis for these sites. The potential for icing was defined as the fraction of the day where ambient conditions could potentially lead to icing of the evaporator. From lab testing, icing was found to occur in some situations when the tank temperature was below 15 °C and the ambient wet bulb temperature was below 10 °C. Since tank temperature was not measured, incoming water temperature was used instead. As can be seen in Figure 36 and Figure 37, the model tends to over predict performance more (a positive difference between measured and modeled system COP) during periods where icing is likely to occur.

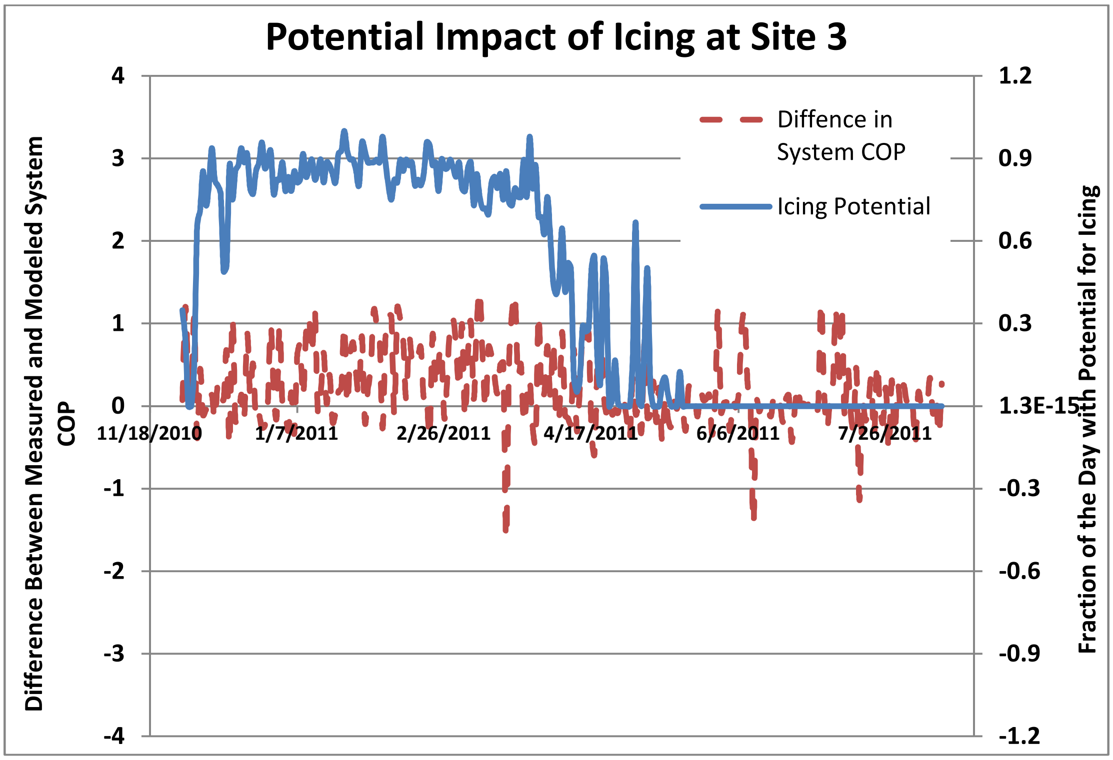


Figure 36: Potential for icing at Site 3 and corresponding difference in system COP

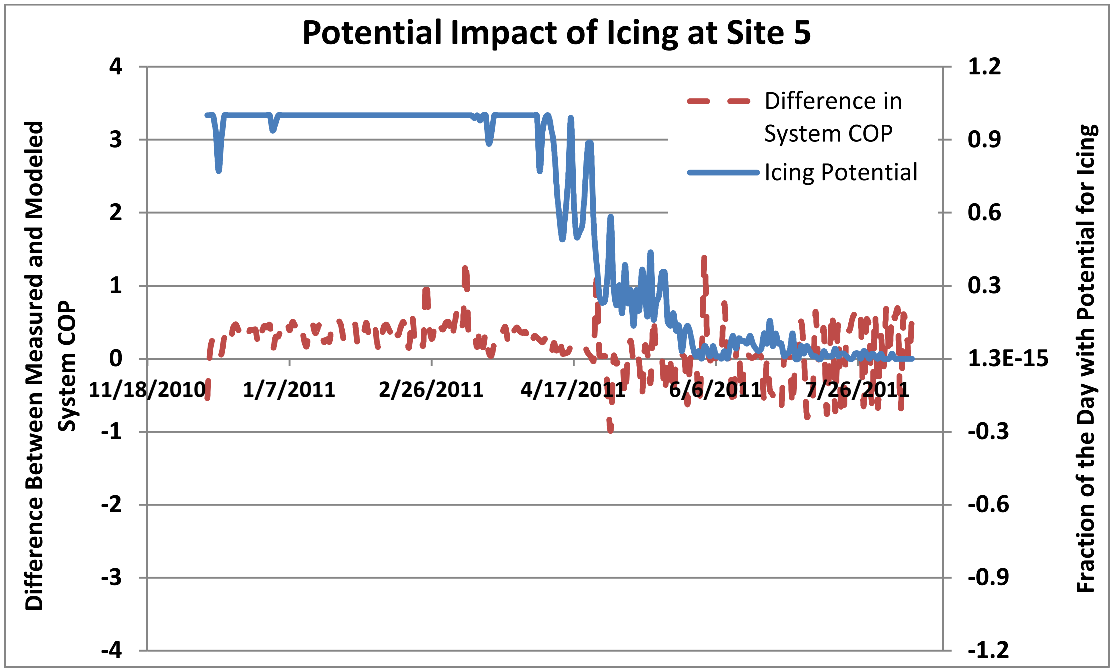


Figure 37: Potential for icing at Site 5 and corresponding difference in system COP

For Site 13, there are two potential causes for the discrepancy. One potential cause is that a different thermistor was used at this site to measure the outlet temperature. At all of the other sites, a 4.5 inch immersion thermistor was installed at the outlet of the HPWH to measure the outlet temperature. However, at this site a 2 inch thermistor was used. This may cause the outlet temperature measurements at this site to be less accurate, likely leading to the recorded outlet temperatures being lower than the actual outlet temperatures. The other potential cause for discrepancies is that this unit was the first installed and the last to get monitoring equipment, meaning it had been running for 6 months prior to the beginning of the monitoring period. While monitoring equipment was installed, the filter was observed to be very dirty and was not cleaned (43).

Other potential causes of variation in the measured and modeled HPWH performance is the large time step over which data was sampled and the accuracy of the measured data. A propagation of error analysis was carried out to determine if the model predictions were within the accuracy of the measured values for heat pump fraction and system COP. For this field test data, temperature was measured by thermocouples with an accuracy of ±0.2 °C, flow rate was measured by a turbine flow meter with an accuracy of ±1% and power was measured by a wattmeter with an accuracy of ±0.5% up to ±0.5 Wh. The results of this analysis are presented in Table 9. Only the COP at sites 6, 7, and 9 were within the bounds of the measurement accuracy. However, there are other factors that introduce some inaccuracy into the model.

The model developed here is designed to be used with short time steps (one minute or less), while the measured data has a resolution of 15 minutes. Past work examining the impact of large timesteps has shown that the system COP and HPF can vary by as much as 10% when comparing a simulation run using 6 second timesteps with the same draw profile averaged into hourly draws. These are the likely causes of discrepancies in the sites which had a calculated system COP that was reasonably close to the measured system COP.

Site	COP Error	HPF Error	Within Error: COP	Within Error: HPF
3	±0.053	±0.000445	No	No
5	±0.059	±0.001872	No	No
6	±0.096	±0.001582	**Yes**	No
7	±0.079	±0.008975	**Yes**	No
8	±0.065	±0.002497	No	No
9	±0.046	±0.019806	**Yes**	No
11	±0.017	±0.000665	No	No
12	±0.029	±0.000327	No	No
13	±0.056	±0.011426	No	No
14	±0.085	±0.021551	No	No

Table 9: Error analysis results for HPWH field test data

4.7 Model Weaknesses and Future Work

While this model performs well when compared to laboratory testing, it does have some shortcomings. Since it is a performance-based model, it can only be validated for the range of conditions under which it was tested. Some extrapolation is possible and was performed here, but the results of this extrapolation have not been fully validated and care must be taken in using these results. In addition, a new performance map has to be generated for every HPWH that is modeled. Additional performance maps can be generated from the lab testing of other units to model these units. The fan speed as a function of fan power would also need to be updated for any new unit and input into the source code of the model. Icing of the evaporator is also not fully captured since the physics of this situation cannot be easily modeled in a performance map based model. Instead, conditions under which icing is likely to occur can be tracked in any simulations. For the annual simulations run here, it was found that icing only occured fairly rarely over the course of a year, but it could be much more frequent in other locations. In particular, it was found to be a problem when comparing the model to some of the field test data.

Chapter 5: Condensing Water Heater Model

Condensing water heaters are similar to traditional gas storage water heaters but feature a heat exchanger that allows the combustion products of the burner to condense. This condensation process provides additional heat to the water in the tank that would otherwise be vented outside. To model a condensing water heater, the conversion efficiency (essentially the combustion efficiency in this case) of the water heater needs to be allowed to vary with the part load ratio of the water heater as well as the average tank temperature next to the condensing heat exchanger. The model developed here captures these effects. However, there is very limited data on the exact efficiency of a condensing water heater as a function of tank temperature and part load ratio. This model is based on a manufacturer's performance map and test results available in the literature. Detailed testing designed to develop a performance map of these units is required to create a fully validated model.

5.1 TRNSYS Model Description

The condensing water heater model used here is based on the standard storage water heater model used for gas and electric water heaters previously described in Section 3.1. However, a new model was used along with the storage tank model to capture the impact tank temperature and part load has on the overall efficiency of the water heater. This new model is based on an existing model (44) which was designed to function as an external heating device for a water heater. The existing model would calculate how much energy was used and how much of that energy went into the tank for a gas or electric storage water heater with a constant efficiency. The new model uses a user provided performance map to determine what the efficiency of the unit is based on a tank temperature from the storage tank model and a part load ratio from a temperature controller or a user specified equation. This allows the model to be used with any potential control strategy a manufacturer may implement that can be modeled within the TRNSYS environment. Since this model is external to the tank model, it can be used for both tankless condensing water heaters and tank condensing water heaters if sufficient data is available to develop a performance map. The impact of part load ratio can also be removed from the performance map if required (for example, for a unit that does not modulate).

The model developed here is based on one particular model of condensing water heater. This particular unit was chosen because the manufacturer has chosen to publish a performance map for their water heater and there is some test data on this unit available in the literature. The manufacturer provided performance map used is shown in Figure 38 and the performance map used by TRNSYS, created from this manufacturer's performance map, is shown in Figure 39.

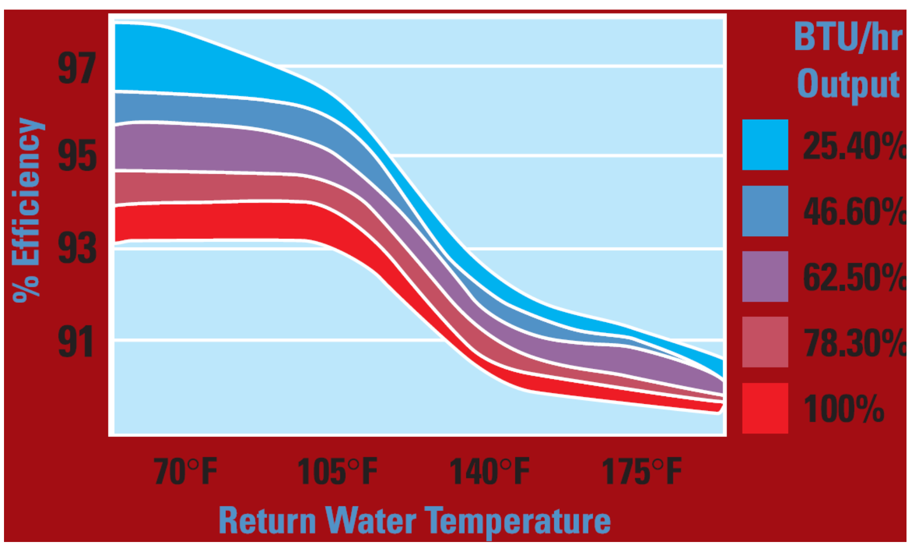


Figure 38: Manufacturer provided performance map for this condensing water heater (45)

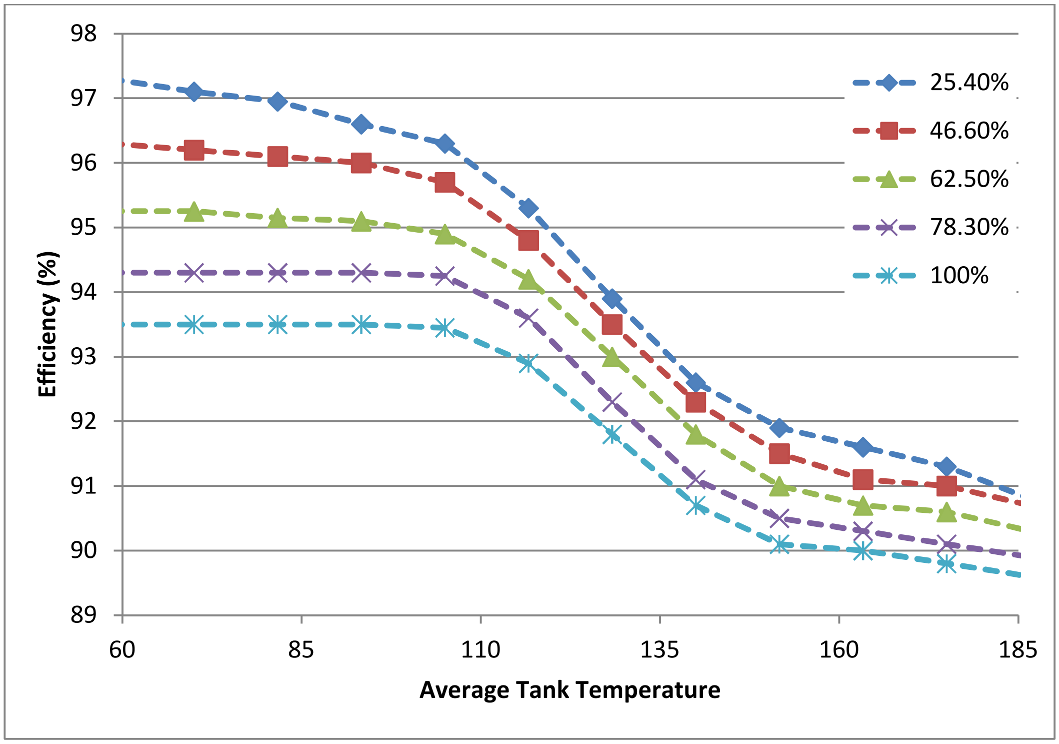


Figure 39: TRNSYS performance map for condensing water heater

The TRNSYS performance map linearly interpolates between the closest data points provided to determine the actual performance of the unit. If performance is needed at any point outside of the performance map, the performance at the closest point is used: no extrapolation is done. Therefore, care needs to be taken to ensure that the performance map includes all available points that could occur during simulations. In this case, the map is complete enough to ensure that the performance at any point that could reasonable occur during simulations is captured. The full TRNSYS performance map is provided in Appendix D and the code for this new TRNSYS component is provided in Appendix E.

5.2 Condensing Water Heater Model Description

This particular condensing water heater has a very large (170,000 Btu/hr) burner size. It is designed to be used for both space and water heating in light commercial and residential applications. This unit has an attached air handler for any residential heating applications that it may be used in as shown in Figure 40 as well as an additional set of inlets and outlets so that it could be used in hydronic heating applications. Since this is much larger than the typical residential water heater burner (usually 30,000-50,000 Btu/hr) and no information on the controls and how the unit modulates was available, the burner was assumed to always fire at 50% of full capacity (85,000 Btu/hr). This is still quite large for residential applications, but much more reasonable than having using the full firing rate. This is also fairly typical of gas condensing water heaters, which generally have a firing rate of 75,000 Btu/hr or more. As more data on condensing water heater performance becomes available, this model can be updated to include the actual controls or models of units which do not modulate may be developed.

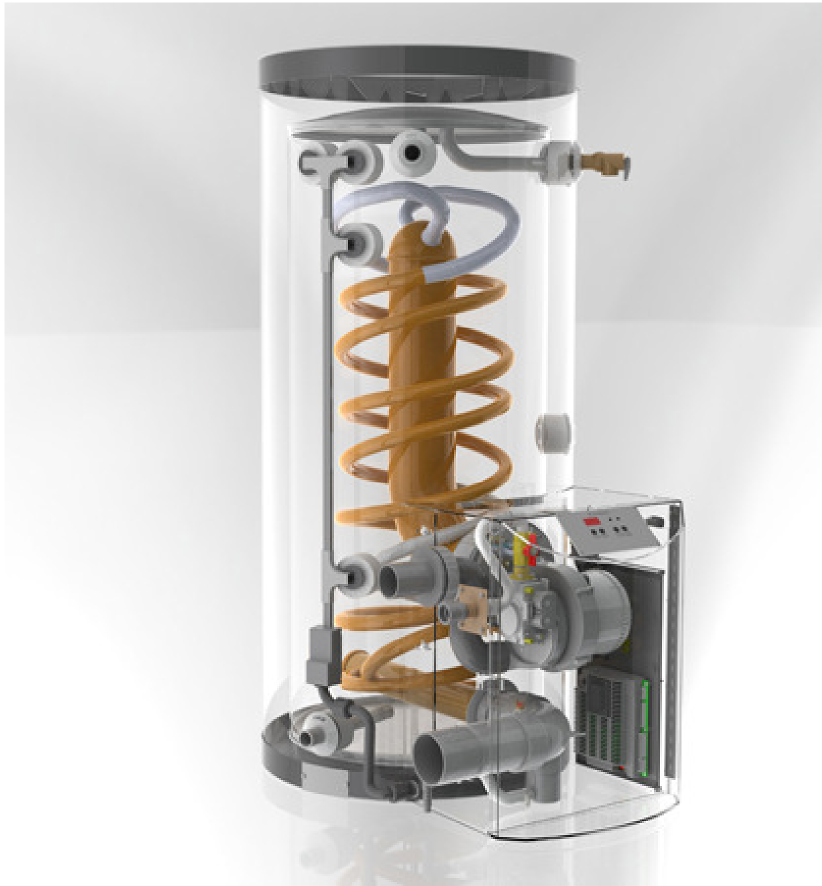


Figure 40: The selected condensing water heater (45)

One other factor of importance to capture in the model is the location of the heat exchanger (to determine which node temperatures in the tank impact the condensing heat exchanger performance) and how much heat goes into each node from the heat exchanger. This tank was subdivided into 15 nodes based on the recommendations made for typical gas and electric storage water heaters. As is shown in Figure 40, the condensing heat exchanger goes almost the entire length of the tank. Based on this, it was assumed that the heat exchanger was present in every node of the tank. To determine the distribution of heat in the tank from the condensing heat exchanger the performance map was utilized. It is assumed that at the lowest efficiency conditions (the highest return water temperature and part load), little condensing is occurring. As the performance gets more efficient, more condensation occurs and there is more heat transferred through the heat exchanger. As less condensation occurs, more heat comes directly from the burner as opposed to being recovered from the flue gasses. The distribution was therefore assumed to be such that the amount of heat going to the bottom node of the tank (next to the burner) is equal to the same percentage as the lowest efficiency provided by the manufacturer (89.6%). The rest of the heat is evenly distributed throughout every node next to the heat exchanger, which in this case is every other node in the tank. The tank model and heat distribution in the tank is shown in Figure 41.

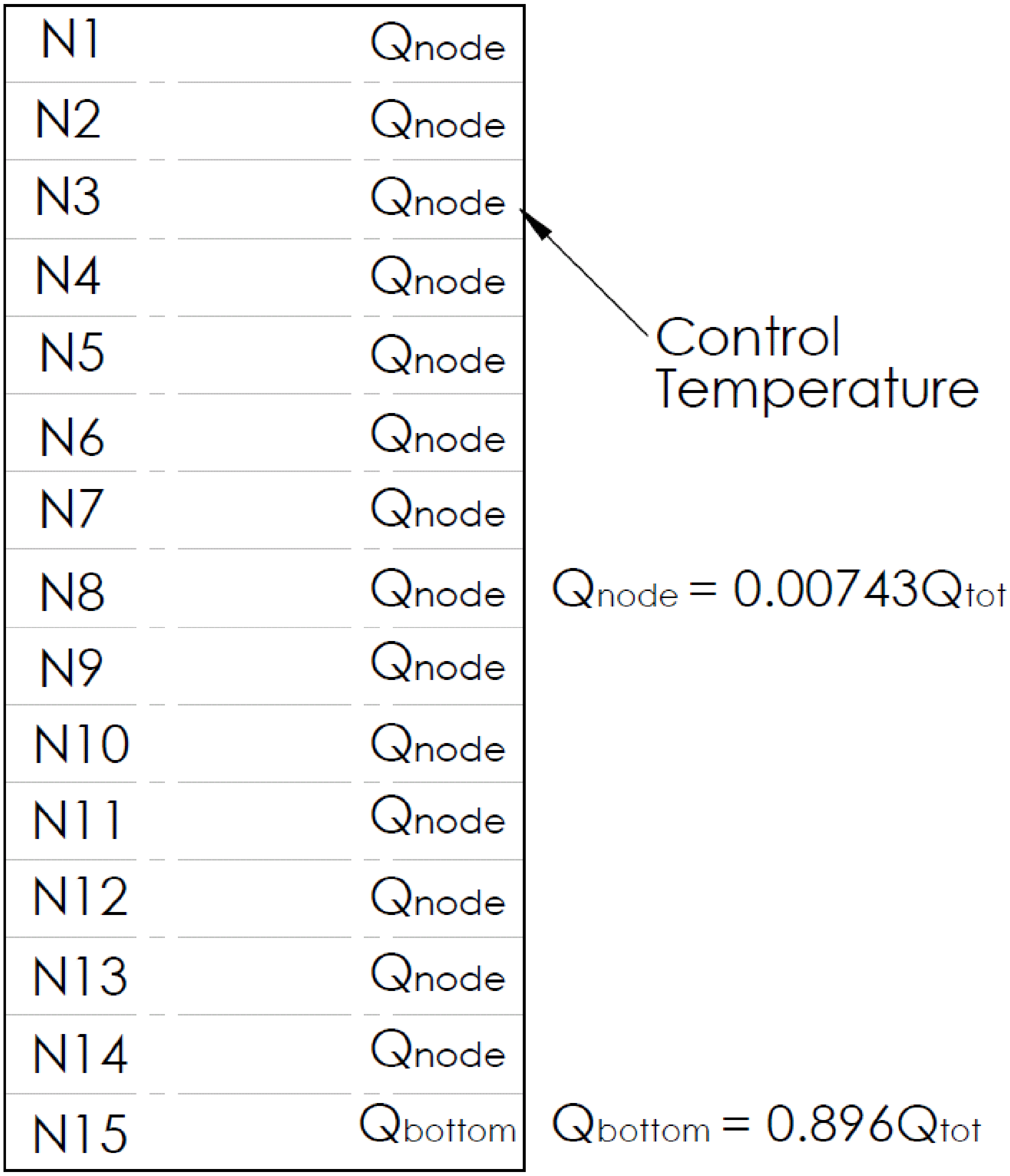


Figure 41: Distribution of heat from the condensing heat exchanger into the water heater

The aforementioned distribution of heat in the tank is an assumption as no detailed data is currently available to determine how the heat is distributed in the tank. In reality, the amount of heat that is added to the tank is probably not evenly distributed through the tank. Most of the heat will be transferred at whatever location in the heat exchanger condensation occurs, which may change with varying tank temperatures and part load ratios. This is another model parameter which would need to be derived through detailed testing designed specifically to extract such parameters.

This unit also consumes some electricity both while operating and during standby for controls and the power vent fan. Testing performed by Pacific Gas and Electric (PG&E) found that the power drawn when the burner was on is 37 W while power consumption during standby is 16 W. The testing done by PG&E is further discussed in the next section, model validation.

The final key model parameter is the tank's overall heat transfer coefficient. This parameter was derived based on the tank's rated efficiency. Since this water heater has such a large burner size, it is rated according to the standard for commercial water heaters (46). As part of the rating procedure, the amount of heat lost by the tank in an hour is reported. Assuming that both the tank and the room are isothermal at rated conditions during the test, the overall heat transfer can be calculated using the simple equation:

$$Q_{loss} = UA(T_{tank} - T_{amb}) \tag{10}$$

The total heat loss was provided by the rating results as 389 Btu/hr. According to test specifications, tank and ambient air temperatures are to be maintained at 140 °F and 65 °F respectively. This gives an overall heat transfer coefficient (UA) of 19.39 Btu/hr or 11.36 kJ/hr.

5.3 Condensing Water Heater Model Validation

Only limited test data on this particular condensing water heater is available. Pacific Gas and Electric (PG&E) performed lab testing of several different types of gas water heaters, including condensing water heaters to identify the potential savings of installing different residential gas water heaters in homes within their service territory (47). As part of their lab testing, they monitored the performance of this unit during both the DOE 24 hour simulated use test (the Energy Factor test) and several daily draw profiles designed to simulate typical occupant behavior. The Energy Factor test was repeated four times during this testing and the final results represent the average of these tests. Since full draw profiles for the simulated use tests are not available, the model was validated by running a simulation of the water heater where it is subjected to the Energy Factor test conditions and comparing the simulated results to the measured results. Comparisons were done based on a simplified Energy Factor which is simply the total energy delivered divided by the energy consumed. The official Energy Factor test procedure also includes the change in energy stored in the tank during the test procedure as the average tank temperature changes which was ignored both in the results provided by PG&E and the simulation results. A comparison of the modeled and measured energy consumed during this test period is provided in Table 10.

	Gas Consumed (therm)	Electricity Consumed (kWh)	Net Energy Consumed (therm)	Energy Factor
PG&E Test	0.505	0.401	0.519	0.833
TRNSYS Results	0.528	0.402	0.542	0.808
% Error	4.50%	0.29%	4.39%	-2.99%

Table 10: Comparison of measured and modeled energy consumption during the Energy Factor test

To determine how accurate the model is, a propagation of error analysis was performed on the PG&E test results. One issue which is a cause for concern is the fact that the flow control valves were seen to drift during this testing, although the flow rate usually stayed within the DOE specified range during the Energy Factor tests. However, the controls were based on the total volume drawn, not the flow rate, so the actual volume of water drawn should be within the specified measurement accuracy. A greater concern is that the energy delivered from this test is outside of what would be expected based on the DOE specified measurement accuracies. The simple Energy Factor, which differs from the official energy factor by not including the change in stored energy over the test period, can be written as:

$$EF_{simple} = \frac{E_{del}}{E_{cons}} = \frac{V_{tot}\rho c_p(T_{out}-T_{in})}{E_{gas}+E_{elec}} \tag{11}$$

Based on the reported energy factor and energy consumed, the delivered energy must 45,149 Btu. Based on the specified test conditions, the average tank temperature (and therefore the outlet temperature) should be 135±5 °F, the inlet temperature should be 58±2 °F, and the total volume drawn should be 64.3±1.29 gallons. These values give a delivered energy of 41,247±2884 Btu. The likely cause for this discrepancy is that stratification was observed to occur, causing the outlet temperature to exceed upper limit of the average tank temperature while keeping the average tank temperature in range. In fact, the stratification may have been quite significant: "It *(the Phoenix)* already created issues with following the DOE standard procedures, because setting the prescribed average tank temperature of 135 °F would create an unreasonably high outlet temperature." (47). However, the actual temperature the tank was set to is not provided. To ensure that the Energy Factor calculation for the error propagation analysis was the same as the measured Energy Factor, the outlet temperature was increased to 142.27 °F to ensure that the delivered energy in this calculation is consistent with the measured data.

The model created here does not significantly stratify, leading to the lower outlet temperature and consequently a lower energy factor. A propagation of error analysis shows that the Energy Factor from the TRNSYS model is outside of the range of the uncertainty of the measured Energy Factor. Note in Table 11 that the measurement uncertainty for the total amount of water drawn, inlet water temperature, and outlet water temperature are different than the allowed limits on inlet water temperature, outlet water temperature, and amount of water drawn since they are based on the prescribed measurement accuracy, which is smaller than the allowable limits. The discrepancy between these two efficiency metrics underscores the need for more performance data for these types of water heaters so that more accurate models can be created.

Variable ± Uncertainty	Partial Derivative	% of Uncertainty
EF = 83.33±1.875	-	-
E_{elec} = 0.402±0.00402	$\partial EF/\partial E_{elec}$ = -5.248	0.01%
E_{gas} = 0.528±0.00528	$\partial EF/\partial E_{gas}$ = -153.8	18.76%
m_{tot} = 536.3±10.73	$\partial EF/\partial m_{tot}$ = -0.1554	79.00%
T_{in} = 58±0.2	$\partial EF/\partial T_{in}$ = -0.9894	1.11%
T_{out} = 142.23±0.2	$\partial EF/\partial T_{out}$ = -0.9882	1.11%

Table 11: Propagation of uncertainty analysis for the condensing water heater Energy Factor test

Chapter 6: Domestic Hot Water Distribution System Model

The domestic hot water distribution system plays a role in determining the overall efficiency of the water heating system. A more efficient distribution system will waste less energy by having fewer losses during delivery of hot water and staying warmer in the pipes for longer after a draw. This hot water in the pipes after a draw can then be used in a subsequent draw if not too much time has passed between events. In addition, more distribution losses will require more water to be drawn from the water heater, increasing its energy use. To examine the impact of the distribution system, a typical domestic hot water distribution system was modeled and two different types of water heater (an electric water heater and a HPWH) were run both with and without the distribution system model to look at the impact of the distribution system and examine the differences between the distribution losses for these two different technologies. These results are presented in Section 8.4.

6.1 Distribution Losses

The developed model uses a simplified "plug flow" model of fluid flow in pipes (27) that breaks the pipes into small discrete sections. Each has its own temperature and the size of each section is determined by the flow rate in the pipes and the time step size. This model neglects axial conduction in the pipe and any mixing between sections. The thermal mass of the pipes is also neglected in this model, although a modification was made to the overall heat transfer coefficient to account for this effect between draws when the water is still and cools to the ambient temperature. To minimize the size of individual sections and model realistic hot water use, a small time step size (6 seconds) was chosen for all simulations. The heat loss from each section to the environment while water is flowing through the pipes is determined by calculating the heat transfer coefficients for the pipe inner and outer surfaces and the thermal resistance of the pipe and any insulation as shown in Equations 12-13. The thermal resistance network for this case is shown in Figure 42.

$$Q = UA\left(T_p - T_a\right) \tag{12}$$

$$R_{net} = \frac{1}{UA} = R_{c,i} + R_{cond} + \frac{R_{c,o}R_{rad}}{R_{c,o}+R_{rad}} \tag{13}$$

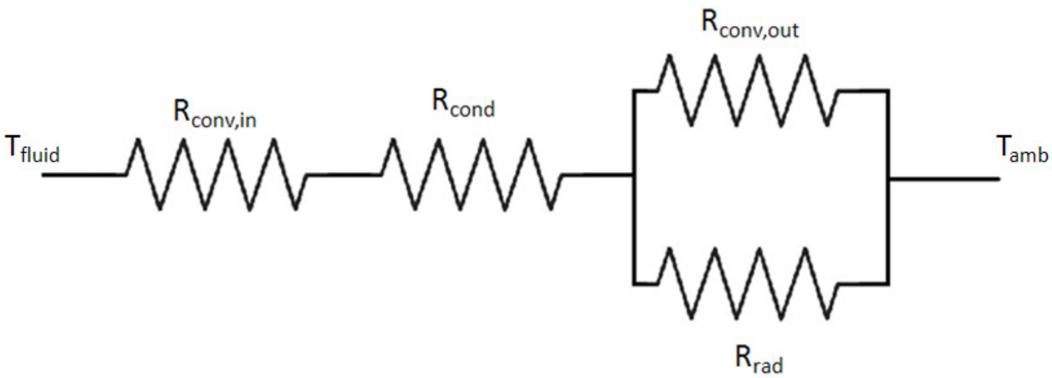


Figure 42: Thermal resistance network for DHW distribution losses

The overall heat transfer coefficient for pipes in the distribution system is determined analytically using a simple thermal resistance network (48). For the inner heat transfer coefficient, an exact solution is used for laminar flow and the Dittus-Boelter correlation for turbulent flow (48). For the outer heat transfer coefficient, radiation and natural convection are considered in parallel. The radiation heat transfer is calculated based on an emissivity (ε) of 0.9 for any insulating material and 0.6 for copper pipes. The emissivity of copper can vary significantly depending on the surface finish of the pipes and was not measured during testing. For this model, the emissivity of copper was determined by comparing the analytic model to test results and adjusting the emissivity to obtain good agreement (Table 12). For natural convection, a correlation developed by Churchill and Chu for horizontal pipes, valid under a wide range of Rayleigh numbers, was used (49). Testing showed only a slight difference between heat losses for horizontal and vertical pipes (50), so this correlation was applied to pipes in both horizontal and vertical orientations. During periods of no flow when water in the pipes is still, the pipes and the water were assumed to be at equilibrium so the same temperature was applied to both and a lumped parameter model was used. The pipe temperature was calculated according to Equation 14.

$$\frac{T-T_f}{T_i-T_f} = \exp\left(-\frac{UAt}{mc_p}\right) \tag{14}$$

To validate this assumption, the Biot number, defined as the ratio of conduction resistance inside the body to the thermal resistance at the surface of the body, was calculated. Cases with a Biot number less than 0.1 are generally considered to have a very small error associated with the lumped parameter assumption. For the worst case of uninsulated ¾ in. (19.05 mm) diameter pipes at 120°F (49°C) in air at 68°F (20°C), the Biot number was 0.106. Lower pipe temperatures, insulation, and ½ in. (38.1 mm) diameter pipe all reduce the Biot number below 0.1. The thermal mass of the insulation was neglected because it is very small compared to the thermal mass of the water and pipe. Heat loss through fittings is also neglected because the surface area of all the fittings in the distribution system was calculated to be less than 1% of the pipes' surface area.

ε	Root Mean Square Error Between Measured and Calculated UA	Average Error Between Measured and Calculated UA
0.5	0.146	5.77%
0.6	0.101	4.57%
0.7	0.167	7.69%

Table 12: Impact of copper emissivity on overall heat transfer coefficient for DHW distribution systems

6.2 Distribution System Model Layout

The prototypical distribution system modeled is based on the Building America program benchmark home (51), which reflects typical construction during the mid-1990s and serves as a baseline for model comparisons. The prototypical distribution system used in the benchmark home is based on a study of California homes that was performed to determine typical distribution system layouts for several homes of different sizes (52). The layout considered in this analysis is designed for a one story, slab on grade, 2010 ft^2 (187-m^2), three bedroom two bathroom home. The distribution system is a trunk-and-branch configuration consisting of uninsulated copper piping where the water heater and the first 10 ft of pipe are in an unconditioned garage. The Building America Benchmark calls for the water heater to be located in conditioned space, so scenarios of locating the water heater and part of the distribution system in both conditioned and unconditioned spaces are modeled.

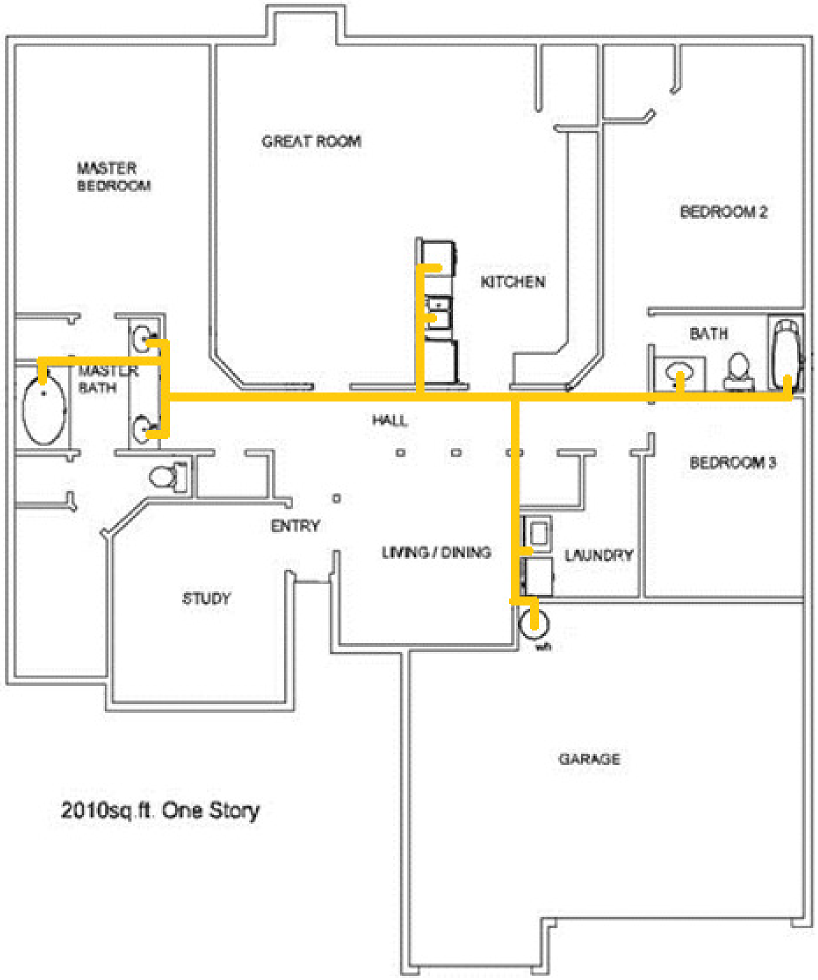


Figure 43: Benchmark domestic hot water distribution system

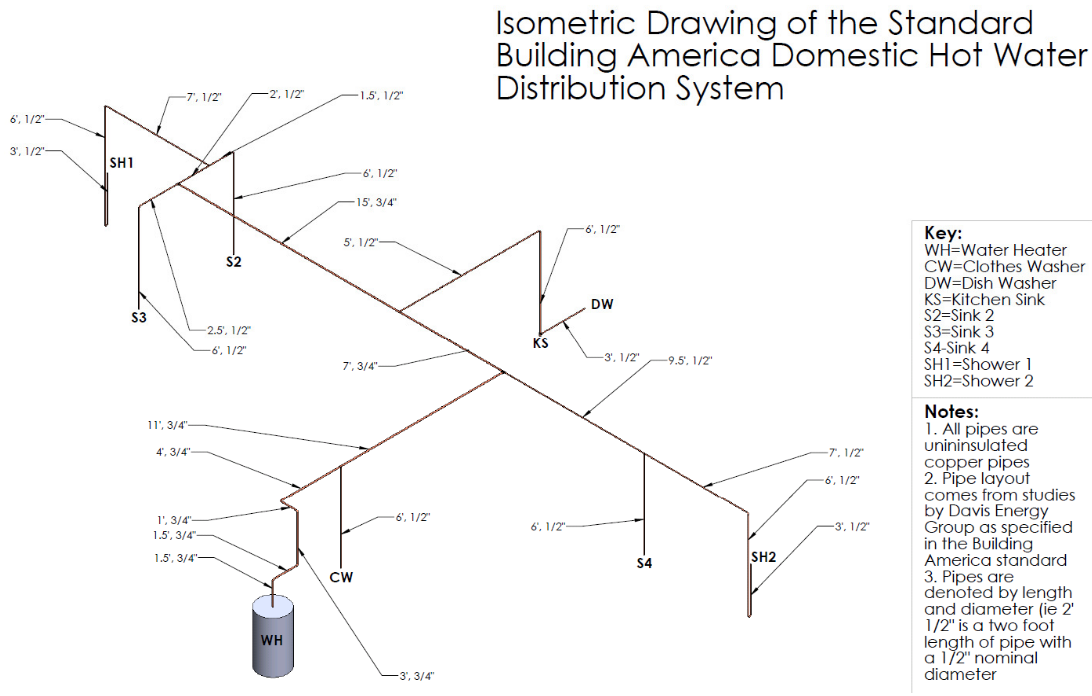


Figure 44: Isometric drawing of the DHW distribution system layout

While only one distribution system is considered in the analysis, many distribution system layouts are possible. Distribution system layouts vary significantly depending on the floor plan of the home and where the water heater is located within the home. In each home, the energy use will change from what is seen with this trunk and branch configuration depending on the distribution system layout, where the distribution system is located (in either conditioned or unconditioned space), and if a recirculation loop is used. While this layout used here is considered to be prototypical, the aforementioned factors mean the results of this study cannot be extrapolated to all homes.

Occupant behavior has a significant impact on the overall losses in a DHW distribution system: clustered events have less heat loss than spread-out events as the pipes have less time to cool to the ambient temperature. To capture the effects of occupant behavior on the distribution system, the Domestic Hot Water Event Schedule Generator was used (53). This tool is further discussed in Section 7.9. This tool uses past surveys of homes to determine the probability of hot water events associated with various end uses (sinks, showers, baths, and appliances), then generates a full year of discrete events for each fixture based on the probability distribution.

Another feature of the DHW Event Schedule Generator is that the events are more realistic if a shorter minimum duration is specified. As an example, consider an occupant using a sink for 10 seconds. If a minimum duration of 1 minute is specified, the volume drawn during this 10 second event will be spread out over 1 minute, resulting in a flow rate that is one sixth the actual event flow rate. This has two impacts on the calculation of distribution losses: the lower flow rate yields a lower heat transfer coefficient calculated for the inner surface of the pipe and the flow is modeled to have ended 50 seconds later than the actual event. This results in higher temperatures of the pipe and its entrained water at the end of the 1 minute draw than it would be for the ten second event. For all of the DHW distribution loss simulations, draw profiles with a 6 second minimum duration were used to capture actual occupant sink use. These are small enough to capture realistic hot water use, especially during sink draws, while having a reasonable run time in the distribution system model. It was also found that 6 second draw profiles show good agreement with 1 second draw profiles in terms of the distribution losses as shown in Figure 4645.

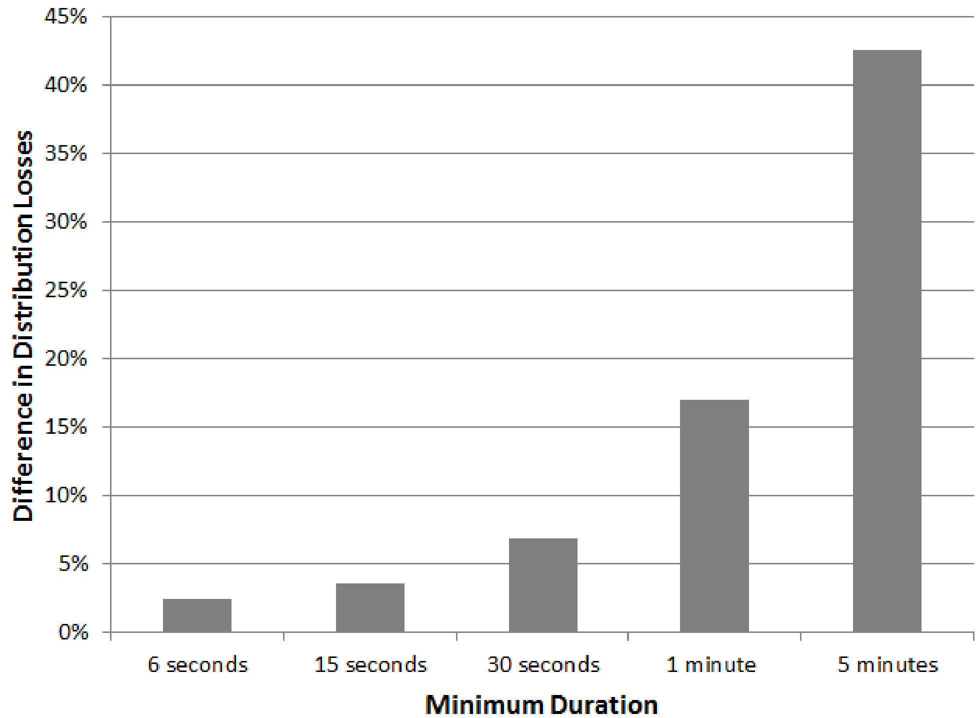


Figure 45: Comparison of the distribution system losses to a one second high use draw profile

This model also takes into account some event clustering for each water end use, as well as differences in weekday use vs. weekend use. It also includes two separate 1 week vacation periods. Any timestep size can be used and up to 6 events going to different end uses can occur simultaneously.

Two types of hot water draws were considered for this model. For sinks, showers, and baths, most occupants will wait until a minimum "useful" temperature is reached before actually using any hot water. To model this, any water drawn below a minimum useful temperature of 105°F was considered to be wasted. This wasted water is also a distribution system loss because the energy spent on heating the water from mains temperature to the water heater set point is completely lost during this "warm-up" period. For appliances (dishwashers and clothes washers), any temperature of water drawn was considered useful. A third potential use option, a so called "Btu draw" where the final temperature of all the water drawn must be above the useful temperature is possible for bath draws depending on occupant behavior, but was not modeled in this study. When analyzing the results, useful and wasted hot water draws are disaggregated and the energy associated with the wasted hot water is included in the overall distribution system losses.

6.3 Validation Analysis

The predictions of the TRNSYS model were compared against measured data. Past work had measured the heat loss for copper and PAX piping (PAX, also known as PEX-AL-PEX, consists of a thin layer of aluminum sandwiched between two layers of PEX) (50), with and without insulation, under typical DHW distribution system conditions. Tests were performed on long segments (over 80 ft) of ½ in. and ¾ in. pipe under controlled laboratory conditions for flow rates between 0 and 5 gpm. The overall heat transfer coefficient was calculated while hot water was being delivered and while the pipes cooled between draws. For the insulated pipes, tests were run with ½ in. and ¾ in. thick insulation for both ½ in. and ¾ in. diameter pipes using insulation with a rated R/in. value of 3.97 $ft^2 \cdot hr \cdot °F/Btu \cdot in$. Some unique flow phenomena, such as slip flow and stratified flow, were observed during this testing for flow rates in the transitional flow regime. These phenomena are not accounted for in the developed model.

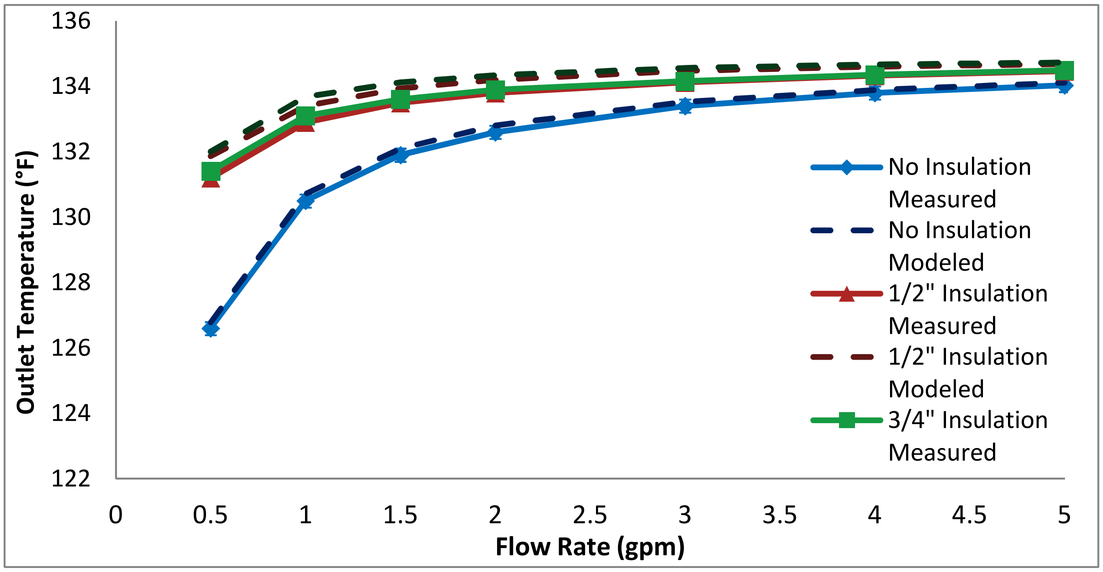


Figure 46: Temperature drop in 100 ft of ½ inch diameter copper pipe with 135 °F inlet water temperature in 67.5°F air

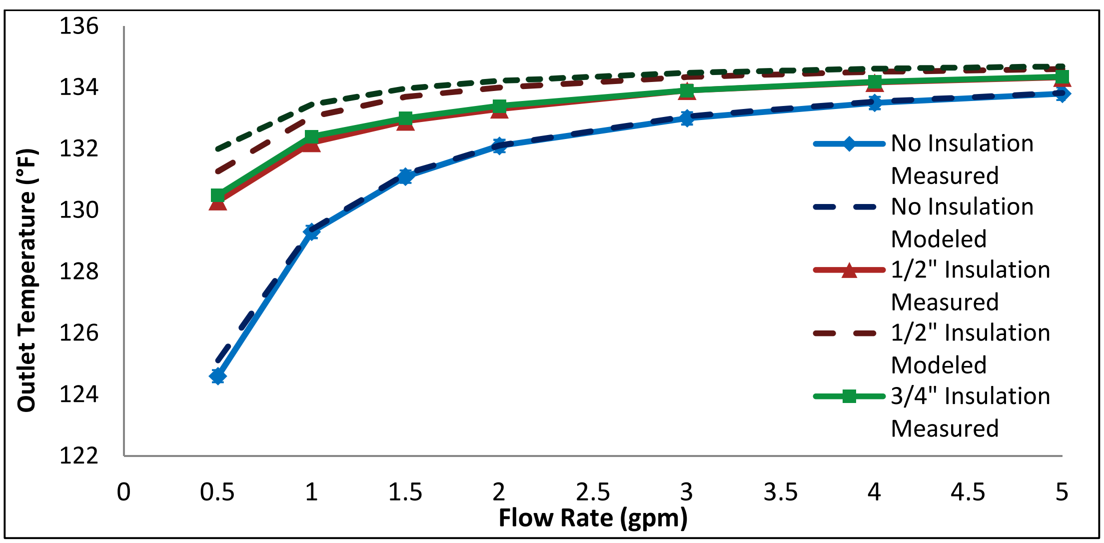


Figure 47: Temperature drop in 100 ft of ¾ inch diameter copper pipe with 135 °F inlet water temperature in 67.5°F air

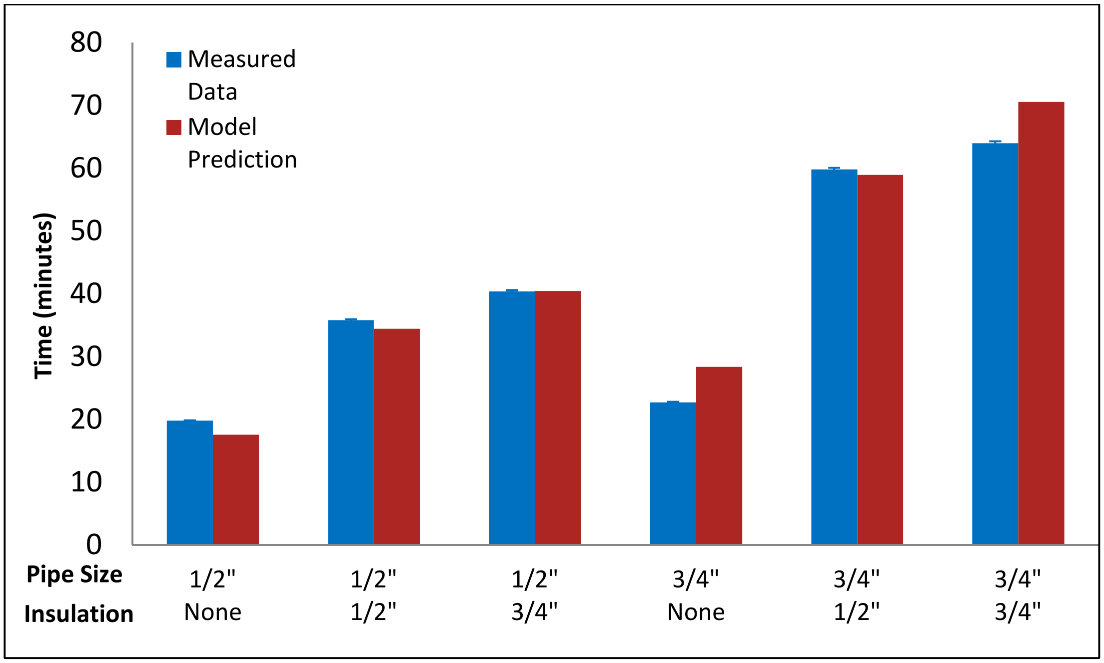


Figure 48: Time for copper pipes to cool from 135°F to 105°F in 67.5°F air

The results of comparing the model predictions against measured data are presented in Figure 46, Figure 47, and Figure 48. In general, the model more closely matches the test data at higher flow rates. At any flow rate, insulation causes a larger discrepancy than for the uninsulated case. This discrepancy increases as the thickness of the insulation increases. The larger discrepancies between the model and measured data at lower flow rates for any level of insulation occur because of the unique flow phenomena, which would be very difficult to capture in a model. The discrepancies caused by the addition of insulation could be related to nonuniform properties or underperformance of the insulation. Additionally, it is assumed that the surrounding air was still during testing and modeling, but airflow around the insulation during testing could cause a discrepancy between the measured data and the model predictions.

A comparative analysis was also performed to compare the performance of the model to the DHW distribution system modeling tool HWSIM (54). For this analysis, the benchmark distribution system was modeled using the HWSIM framework with a realistic week of draw profiles. However, the TRNSYS model and the HWSIM tool treat draw profiles somewhat differently: draw profiles in the TRNSYS model are assumed to include hot water waste for mixed events and those in the HWSIM tool do not. The TRNSYS draw profiles used for this validation are therefore specified as the draw by the water heater, while the HWSIM draw profiles are specified as the draw of hot water at the end use. This complicates a direct comparison. In an attempt to compensate for this difference, additional distribution system losses and hot water use were added to the TRNSYS model results during post-processing based on the losses observed during the useful portion of minimum temperature events. This made the comparison more direct but led to an over prediction of the losses and hot water use. The results of this comparison and the difference between the HWSIM prediction and the post-processed TRNSYS model are shown in Table 13. All other draw profiles used for modeling in TRNSYS included hot water waste, so no post-processing of this nature was required for any other simulations. To minimize the differences between the two models on simulations of a full year of draw profiles (HWSIM models one week for each month of the year while the TRNSYS model considers every day of a full year), the ambient and mains temperature for both models were kept constant year round.

	HWSIM	TRNSYS	Difference
Hot Water Use In Gal/Day (L/Day)	71.1 (269)	73.4 (228)	3.24%
Hot Water Waste in Gal/Day (L/Day)	10.1 (38.2)	10.7 (40.5)	5.55%
Distribution System Losses in kBtu/Day (kWh/day)	6.99 (2.05)	6.98 (2.04)	0.11%

Table 13: Comparison of TRNSYS and HWSIM distribution system losses

Chapter 7: Home Models

All of the water heaters modeled here were modeled as being inside a home using Type 56 in TRNSYS (55). Including a home model allowed any interactions between the space heating and cooling loads and the water heater's energy consumption to be captured. Interaction with space loads could come from tank losses from the water heater or, in the case of a HPWH, heat being removed from the space and added to the storage tank. In cases where the water heater was located in unconditioned space, the building model is necessary to accurately capture the ambient conditions (temperature and humidity) for correctly calculating the tank losses and the heat pump COP for HPWHs. A complete description of the building is provided below. While many of the building parameters were taken directly from the Building America House Simulation Protocol (51) guidelines for new construction homes, there were several deviations that prevent these buildings from being considered true Building America Benchmark buildings. A complete list of deviations from the Benchmark and a comparison of the buildings used here to Benchmark buildings as modeled in BEOpt E+ (a NREL developed building modeling software) is provided in Appendix F.

7.1 Building America Climate Zones

Climate can play an important role in determining the energy consumption and efficiency of water heaters in different locations across the United States. Water heater energy consumption varies with the incoming mains water temperature, which changes with location. In addition, the impact the water heater has on space conditioning equipment energy consumption (for water heaters in conditioned space) and the tank losses (for water heaters in unconditioned space) vary with location as well. To capture the impact of these factors, water heaters were modeled in several locations in a variety of climates across the US.

There are eight different Building America climate zones (56): Marine, Mixed-Humid, Hot-Humid, Mixed-Dry, Hot-Dry, Cold, Very Cold, and Subarctic (which only occurs in northern Alaska) as shown in Figure 49. Of these 8 climate zones, 5 are considered "major" climate zones in this study as they are the only climate zones that contain large portions of the population. These "major" climate zones are the Marine, Hot-Humid, Mixed-Humid, Hot-Dry, and Cold climates. In general, one location was chosen as representative of each of the major Building America climate zones. However, two locations (Seattle, WA and Los Angeles, CA) were chosen in the marine climate zone to capture both a warm and cold marine climates. A list of the representative cities chose for each climate zone is given in Table 14 and a map of each cities location in the climate zone is given in Figure 49. For each location chosen, a "typical" building (one that is largely consistent with the Building America House Simulation Protocol) was modeled.

Climate Zone	Representative City
Cold	Chicago, IL
Mixed-Humid	Atlanta, GA
Hot-Humid	Houston, TX
Hot-Dry	Phoenix, AZ
Marine (Warm)	Los Angeles, CA
Marine (Cold)	Seattle, WA

Table 14: Representative cities used in this study

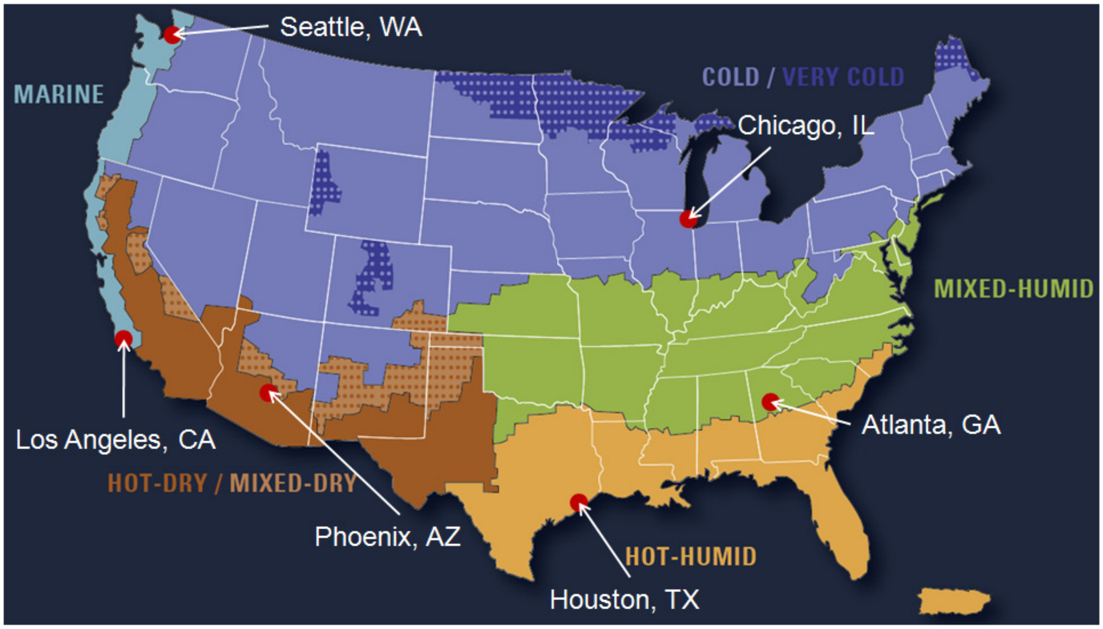


Figure 49: Building America climate zones (56) and the representative cities chosen for this study

7.2 General Building Features

In each location, a 2500 ft^2, two story home facing due south with no neighbors was modeled. The footprint of this home is 42 ft by 30 ft. This home is assumed to have 3 bedrooms and 2 bathrooms. All homes also have a 420 ft^2 (20 ft by 21 ft) garage attached to the south side of the home. All homes had a 6:12 pitched roof and an unfinished attic. For all locations, the heating set point was set to 71 °F and the cooling set point was 76 °F. There were no heating and cooling seasons (months where only the heating equipment or cooling equipment was on) accounted for in these set points, and no set back was included.

7.3 Building Envelope Description

The building envelope construction and U values used here are based on the IECC 2009 Energy Conservation Code (57). This code prescribes R values for the ceiling, walls, floor, and foundation walls depending on the location of the home as shown in Table 15. Garage walls and ceilings used the same R values as those used for the conditioned space. All garage floors consist of an uninsulated slab. In addition to the prescribed R values, a framing factor was also applied to all homes. The framing factors used are 23%, 13% and 11% for walls, floors, and ceilings respectively. Windows were also included and 18% of the total wall area was assumed to be made up of windows. These windows were evenly distributed across all of the walls. For all homes, the windows have a U value of 0.35 Btu/h·ft^2·°F and a solar heat gain coefficient of 0.35. A constant interior shading coefficient of 0.7 is also used.

Heat transfer coefficients for the inside walls are calculated internally by TRNBuild, TRNSYS's building modeling program. The heat transfer coefficient for outside walls as a function of wind speed comes from the ASHRAE fundamentals handbook (58) and is expressed as:

$$h_{outside} = 8.23 + 4V_{wind} - 0.057{V_{wind}}^2 \tag{14}$$

In the previous equation, wind speed must be given in m/s. Wind speed at the home was calculated using a power law. The shear exponent and height of the weather station are assumed to be 0.1 and 10 m respectively. The home's shear exponent and height are assumed to be 0.3 and 4.8768 m respectively. The absorptivity of exterior walls is 0.6 and the emissivity is 0.9. For roofs, the absorptivity is 0.75 and emissivity is 0.9.

Location	Ceiling R-Value	Wall-R-Value	Floor R-Value	Basement Wall R-Value	Slab R-Value and Depth
Houston Phoenix	30	13	13	0	0
Los Angles Atlanta	30	13	19	5	0
Chicago Seattle	38	13+5*	30	10	10, 2 ft

*Table 15: Building envelope R-values (57) *13+5 indicates R13 walls with R5 sheathing*

7.4 Foundations

The building foundation type used here was determined based on whatever foundation was most common in that state (59) as shown in Figure 50. Basements were used in Chicago, Atlanta, and Seattle, while a slab on grade foundation was assumed for Los Angeles, Phoenix, and Houston.

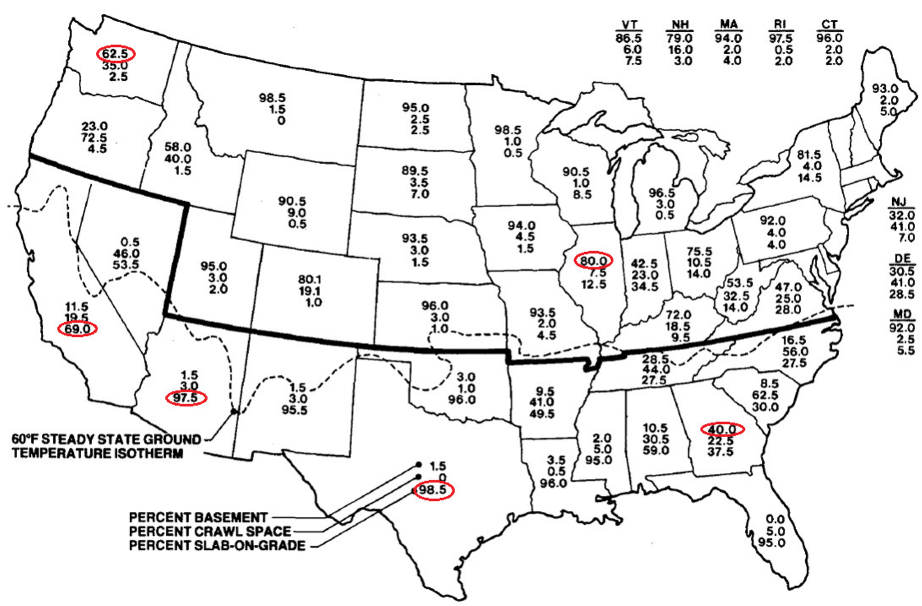


Figure 50: Foundation type by state in residential buildings (59). The foundation type used in each city for this study is circled.

Foundations were modeled using the technique describe by Winklemann for modeling foundations in DOE2 (60). The governing equation for this method is:

$$Q_{ground} = U_{eff}A\left(T_g - T_s\right) \tag{15}$$

In this equation, T_g is the ground temperature, which is calculated using the same methodology as is used in DOE2, T_s is the space temperature adjacent to the ground (either the basement temperature or conditioned space temperature depending on the foundation type), A is the surface area of the foundation, and U_{eff} is the effective heat transfer coefficient. U_{eff} is calculated based on the methodology provided by Winklemann. Depending on the foundation type, the perimeter conductance factor changes. This drives the difference in foundation heat transfer. All perimeter conductance factors used are given in Table 16. For homes with basements, heat is transferred from the conditioned space to the basement by conduction only: no air moves between conditioned and unconditioned space.

Foundation Type	City	Perimeter Conductance Factor (W/m-K)
Slab R10, uninsulated, carpeted	Los Angeles Houston Phoenix	1.30
Slab uninsulated, uncarpeted	All (garage)	1.90
Basement, 8ft, uninsulated	Atlanta	3.35
Basement, 8ft, R10 Insulation	Chicago Seattle	1.30

Table 16: Perimeter conductance factors for all foundation types used (60)

7.5 Infiltration and Ventilation

Mechanical ventilation is used in conditioned space in accordance with ASHRAE Standard 62.2 (61). Ventilation rates are 55 cubic feet per minute (cfm) for most of the day, with three changes up to 155 cfm at 6:30 am, 10:30 am, and 5:30 pm as shown in Figure 51 accounting for spot ventilation due to bathroom, kitchen and dryer vents. Ventilation rates are constant for the whole year and in all homes. No ventilation energy consumption is modeled here as the energy consumption is completely independent of the home location and water heater type. Natural ventilation is also not considered here.

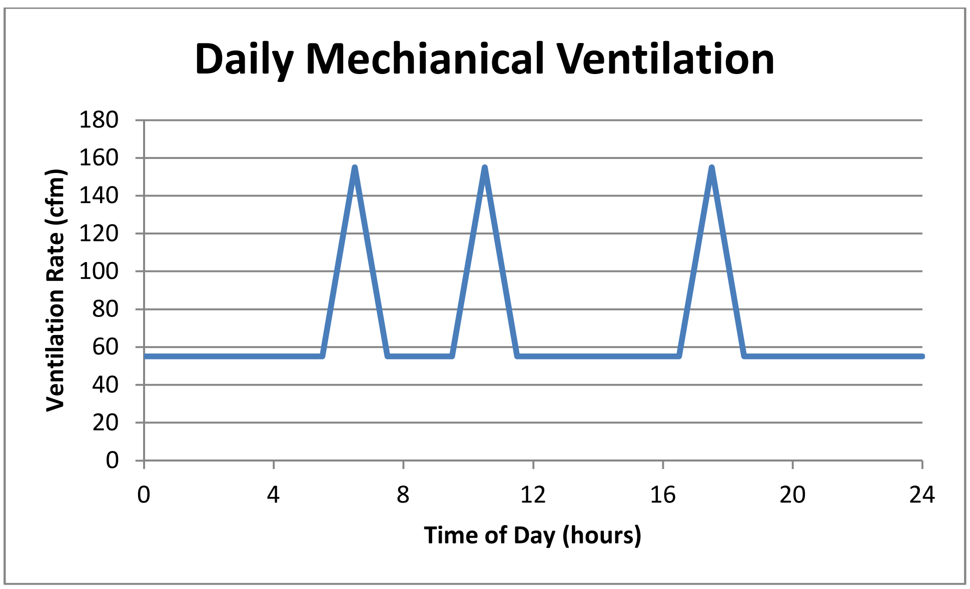


Figure 51: Mechanical ventilation rate in conditioned space

Infiltration occurs in all spaces (the conditioned space, attic, garage, and basement if applicable) of the home. The infiltration rate was calculated using one of two different models depending on how much impact infiltration has on the water heater or home energy consumption. For spaces where the impact is expected to be small, attics and basements, the Sherman-

Grimsrud model was used (58). For spaces where the impact of infiltration is significant, conditioned space and garages, the ASHRAE enhanced model (58) was used. The ASHRAE enhanced infiltration model takes longer to calculate but is more accurate and was therefore only used in spaces where the impact of infiltration is large.

Infiltration for both models is calculated using the effective leakage area (ELA), which is calculated based on the specific leakage area (SLA). Specific leakage area can be calculated as:

$$SLA = \frac{ELA}{CFA} \tag{16}$$

In the above equation, CFA is the conditioned floor area. For all areas, the specific leakage area is 0.00036. Conditioned floor area is considered to be the floor area in both conditioned and unconditioned spaces (for example, even though the attic is unconditioned, its floor area is used as the CFA in the above equation). In basements, only 1 foot of the 8 foot wall area is assumed to be above grade, so the effective leakage area used is 1/8 of what would be calculated using the above equation. Wind velocities used for infiltration calculations are the same as those used for calculating the outside heat transfer coefficient as described in Section 7.3.

7.6 Internal Gains

Internal gains come from equipment, occupants, and lighting. Equipment consists of everything in the space that provides both sensible and latent gains to the space. Occupancy gains come from people living in the space and consist of both sensible and latent gains. Lighting gains are entirely sensible gains. All gains and gain schedules come from the Building America House Simulation Protocol. The total sensible equipment gains, including appliance and miscellaneous gains, are 42851 Btu/day and latent equipment gains are 4118 Btu/day. Occupancy sensible gains are 9597 Btu/day and latent gains are 7154 Btu/day. Lighting is modeled in both the garage and the conditioned space. In conditioned space, the gain is 4627 kWh/day and in the garage the gain is 111.5 kWh/day. The daily schedules for all internal gains are constant throughout the year. The daily schedule for equipment gains, lighting gains, and occupancy gains are given in Figure 52, Figure 53, and Figure 54 respectively.

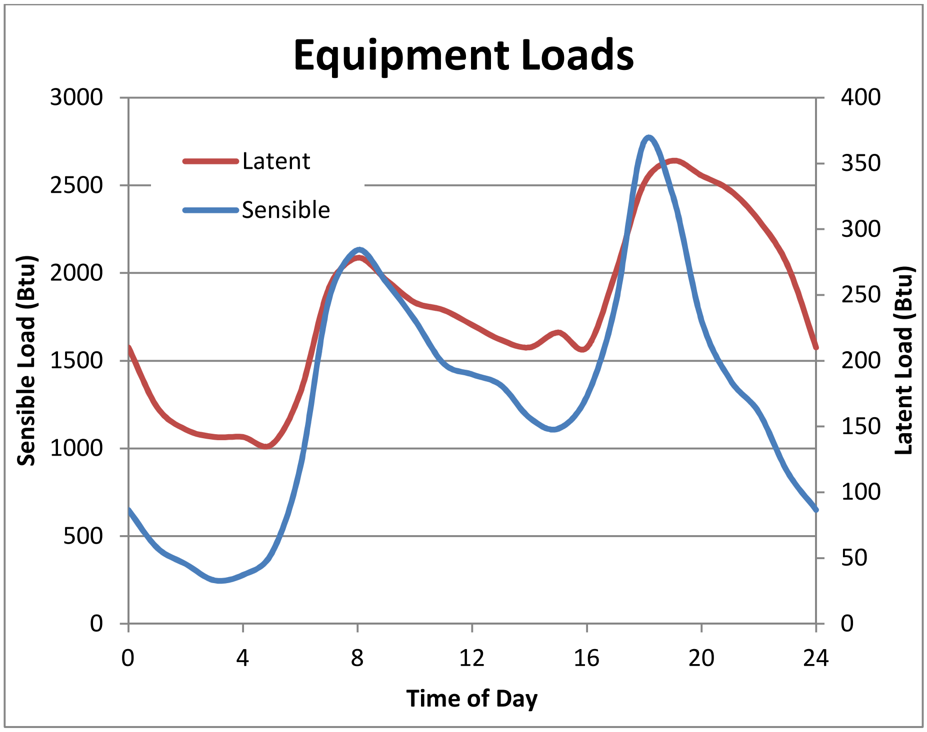


Figure 52: Daily equipment load schedule

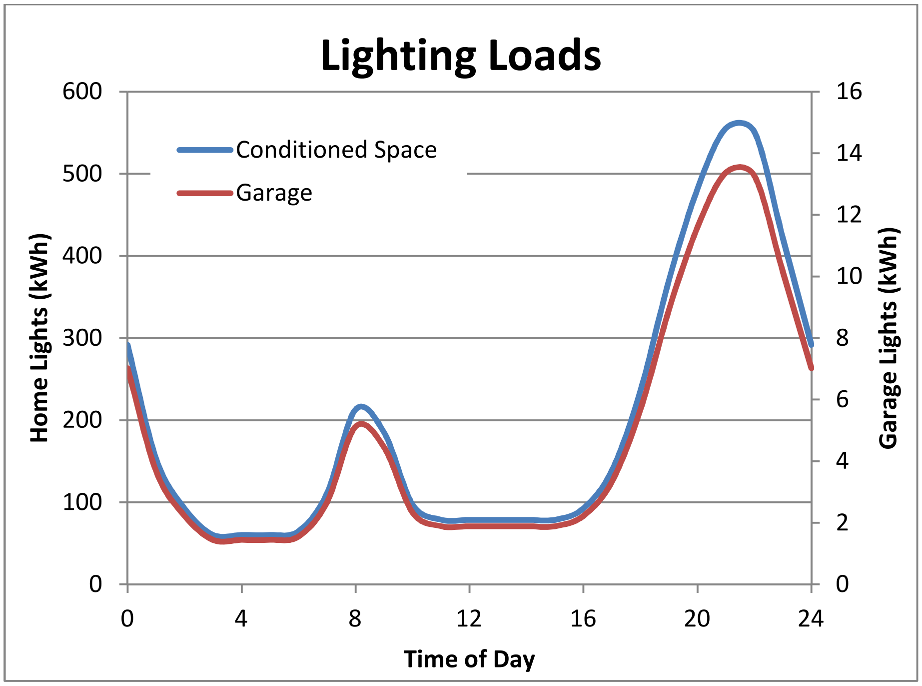


Figure 53: Daily lighting gain schedule

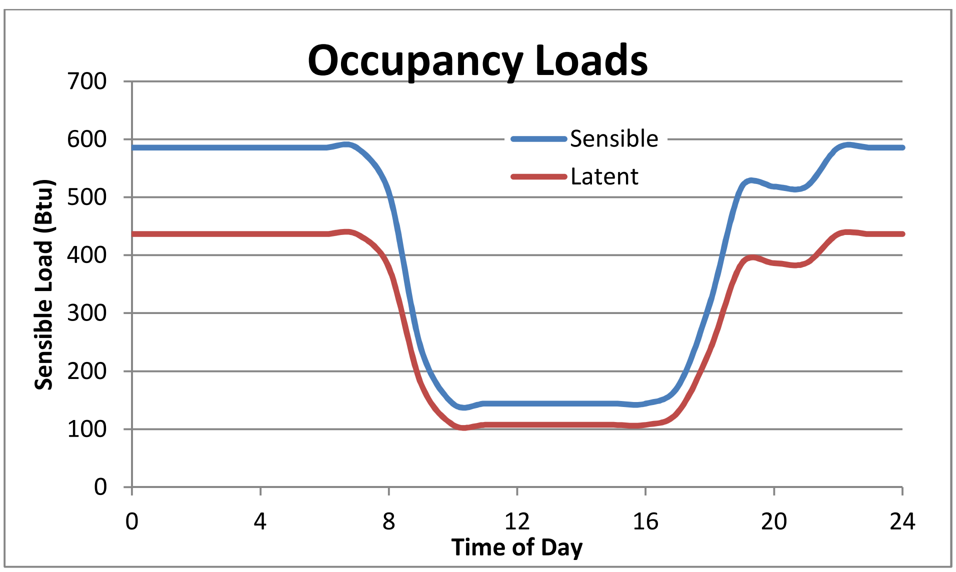


Figure 54: Daily occupancy gain schedule

7.7 Space Conditioning Equipment and Sizing

Two different types of space conditioning equipment are considered here. For homes with a gas water heater, a furnace and an air conditioner were used. For homes with an electric water heater, an air source heat pump was used. These two different types of equipment were considered since a home that has gas service is likely to use it for both water heating and space heating, while those without would need to use electricity for both. It is uncommon to see homes that use gas for water heating and electricity for space heating and vice versa, so these mixed fuel cases are not considered. In all homes, no dehumidification equipment was installed.

For homes that have gas service, both the AC and the furnace need to be correctly sized. All equipment sizing was done using BEOpt and the furnace, AC, and fan size used in each house is provided in Table 17. The furnace was modeled using TRNSYS Type 121a, a simple furnace model. The furnace was modeled as having a constant efficiency equal to the rated AFUE of a Building America Benchmark Furnace, 78%. No part load effects were taken into consideration. The AC modeled here is modeled using TRNSYS Type 921, a single speed air conditioner. For all homes, a SEER 13 unit was modeled. This particular unit requires a performance map. Performance maps for the AC were taken from past work (62) that created TRNSYS specific performance maps for a SEER 13 unit. Since not all the sizes that BEOpt prescribed had TRNSYS performance maps (in particular units that were sized with fractions of a ton) all of the AC units were modeled in TRNSYS as being 0.5 tons larger than what BEOpt had the units sized as. This is shown in Table 17.

Location	Furnace Size (kBtu/hr)	AC Size (tons)	Fan Flow Rate (cfm)
Atlanta	40	3 (2.5)	1000
Chicago	50	3 (2.5)	1000
Houston	40	3 (2.5)	1000
Los Angeles	30	2 (1.5)	600
Phoenix	30	4 (3.5)	1400
Seattle	30	2 (1.5)	600

Table 17: Furnace, AC, and fan sizes. BEOpt AC sizes are provided in parenthesis.

In addition to the furnace and AC, a fan is required to move air from the space to the conditioning equipment. The fan is modeled using TRNSYS Type 112a, a single speed fan. The fan was sized based on the AC size provided by BEOpt. For every ton of AC capacity, 400 cfm was required of the fan. The fan power is 0.00059 kW/cfm. The fan is modeled as being 90% efficient, and any losses from the fan are assumed to become heat added to the air flow through the fan.

All homes with an air source heat pump (ASHP) used TRNSYS Type 954a to model the unit. All of the units used a SEER 13 ASHP for both heating and cooling. For heating, the ASHP had 2 stage backup electric resistance heaters that turned on if the outside air temperature was too low for the ASHP to operate. The first resistance heater, with a capacity of 5 kW, turns on if the outdoor air temperature drops below 40 °F. The second heater, with a capacity of 10kW, turns on if the outdoor air temperature drops below 25 °F. A crankcase heater is also included with a power draw of 0.02 kW is used to keep the unit operating effectively if the outside air temperature drops below 50 °F and heat is required. The ASHP uses the same fan as was used in the furnace and AC case for a home in the same location.

Heating, cooling, and fan energy consumption for a home with a gas water heater and electric water heater in conditioned space using a medium draw profile are provided in Figure 55 and Figure 56 respectively.

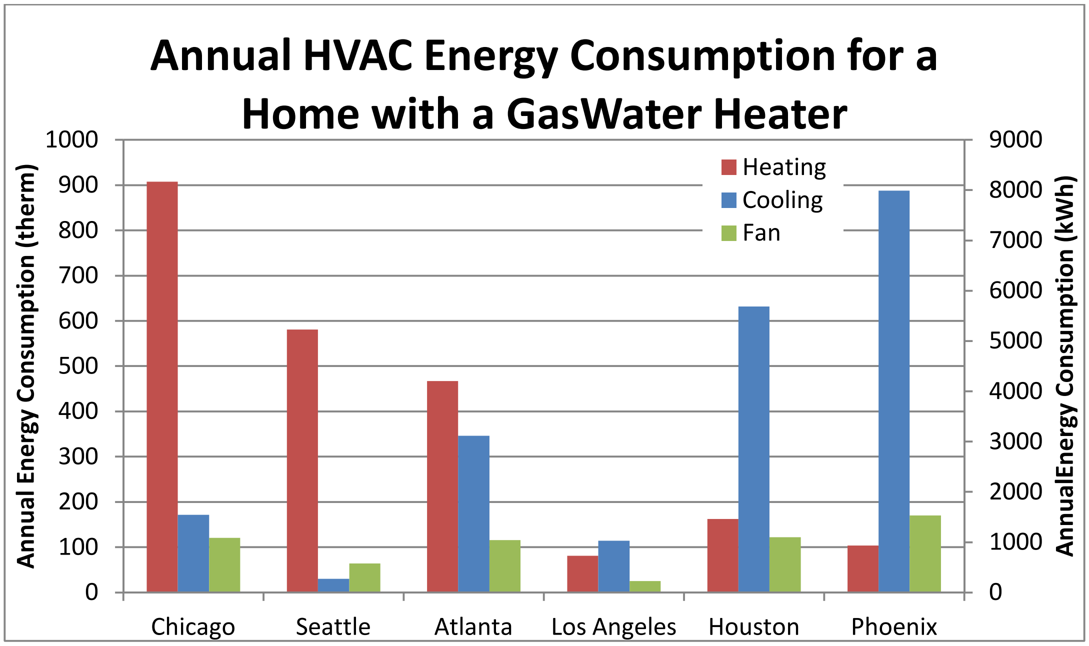


Figure 55: Heating, cooling, and fan energy consumption for a home with a gas water heater

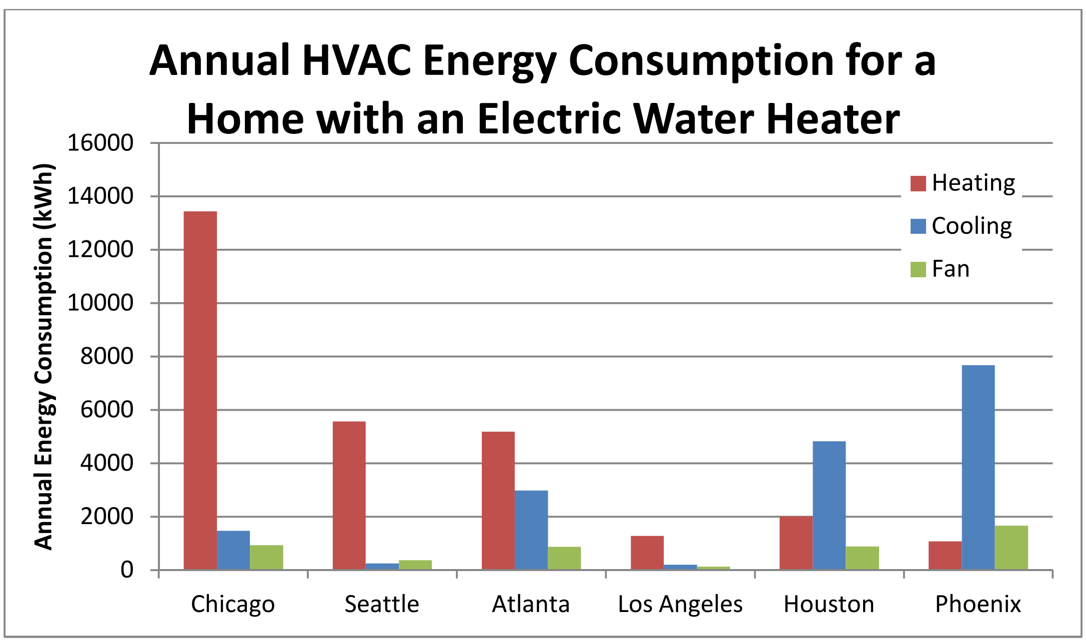


Figure 56: Heating, cooling, and fan energy consumption for a home with an electric water heater

7.8 Capacitance

Thermal capacitance exists in all spaces of the home as a result of the thermal mass of any objects and air in the space. Based on the Building America Benchmark home assumptions, a thermal capacitance of 14250 kJ/K is used in conditioned space, 382 kJ/K is used in the attic, 85.62 kJ/K is used in the garage, and 342.5kJ/K is used in the basement for homes that have one.

7.9 Domestic Hot Water Use

As water heater energy consumption was the focus of this work, a more detailed domestic hot water (DHW) schedule was used than what is prescribed in the Building America Benchmark. The Benchmark uses an hourly draw profile as shown in Figure 57. However, many water heaters, including tankless, solar, and heat pump water heaters, need a subhourly draw profile with discrete events to correctly model their performance. The Building America Domestic Hot Water Event Schedule Generator (DHWESG) was utilized to provide the necessary discrete events (53). The DHWESG is a statistical tool that generates discrete events that fit the same probability distribution as is used in the Building America Benchmark. A sample day of draws compared to the daily hourly draw profile specified in the Building America Benchmark is provided in Figure 57.

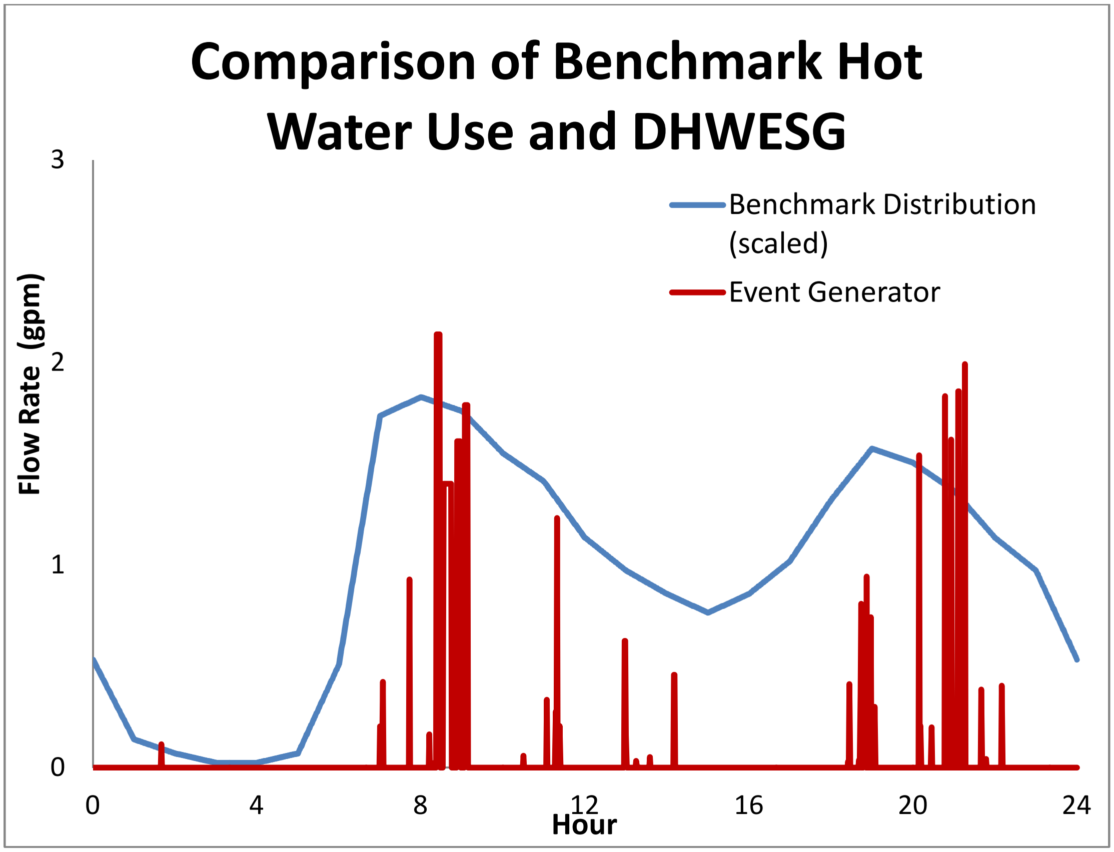


Figure 57: Comparison of benchmark domestic hot water draw profile and event schedule generator draw profile

The DHWESG creates a full year of unique draw events similar to those shown in Figure 57. Among the features of the DHWESG are realistic event clustering, separate probability distributions for weekdays and weekends, and the inclusion of vacation periods. Vacations occur for 3 days in May, one week during August, and 4 days in December. Events created by the event schedule generator have a specified mixed flow rate. The mixed flow rate is the flow rate that an occupant would actually use for mixed draw events such as sink draws, shower draws, and bath draws. For these events, a homeowner will not use purely hot water, which could scald, but instead temper the hot water with cold mains water to a useful temperature. For this work, the useful temperature is defined as 105 °F and all water heaters have a set point temperature of 120 °F. Specifying a mixed flow rate as opposed to a hot flow rate allows the amount of hot water drawn to vary with mains water temperature, which leads to different volumes of water being drawn at different locations. For appliance draws (clothes washer and dishwasher draws), the hot flow rate is specified since these devices generally do not attempt to temper the incoming hot water to any specific temperature.

The amount of domestic hot water used by a particular household is highly variable. While the volume of water drawn can roughly be tied to the number of occupants of a home (and as a result, the number of bedrooms), there is a large amount of variation in use that comes from occupant behavior. To try to capture this behavior, three different draw profiles were created and used in this work. These draw profiles correspond to a one bedroom, three bedroom, and five bedroom home in the DHWESG and are intended to represent low, medium, and high domestic hot water users. While the profiles are based on assuming a different number of bedrooms in a home, all of these draw profiles are used in a home of the same size. This is to capture the aforementioned variations in occupant behavior, which can lead to large differences in hot water use between two homes with the same number of occupants.

The full annual draw profiles are unfortunately too large to be included here, although summary statistics are provided. Figure 58-Figure 60 provide the annual draw volume broken down by end use for low, medium, and high use homes respectively. Figure 61 provides the volume of water drawn each month for all three draw profiles, Figure 62 provides a histogram of the duration of DHW events, and Figure 63 provides a histogram of the flow rate of DHW events. It should be noted that while the events generally average out to the hourly profile shown in Figure 57, day to day and even month to month the volume of water drawn won't perfectly average out to this draw profile. Due to the nature of the event schedule generator, widely different daily and monthly draw volumes may be specified. As a result, the draw volumes in Figure 61 do not linearly scale as the use increases from low to high.

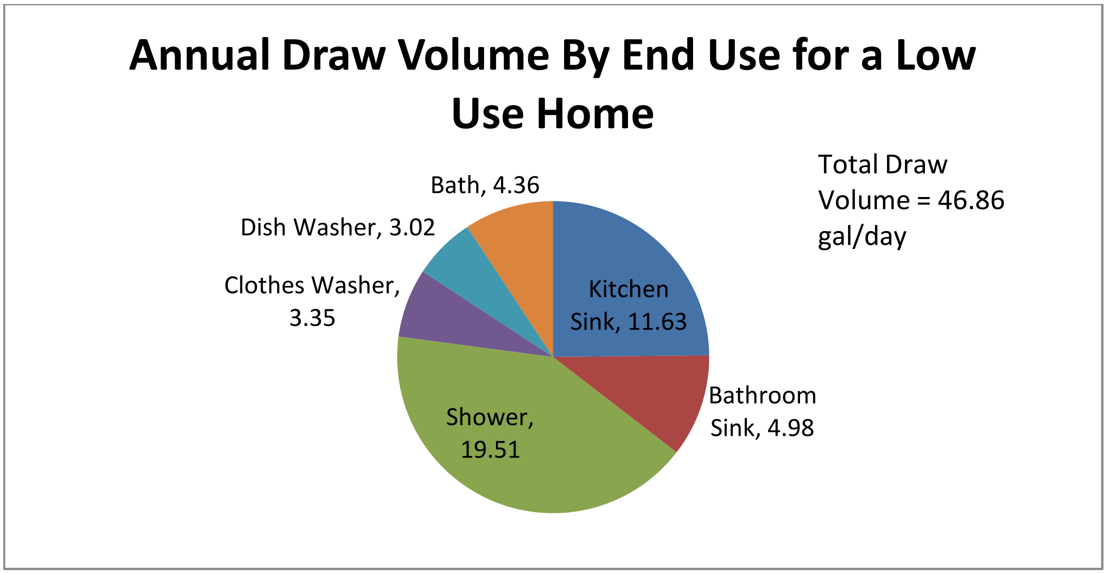


Figure 58: Annual draw volume by end use for a low use home. For clothes washers and dishwashers the volume of hot water drawn is shown, while for all other draws the mixed volume is shown

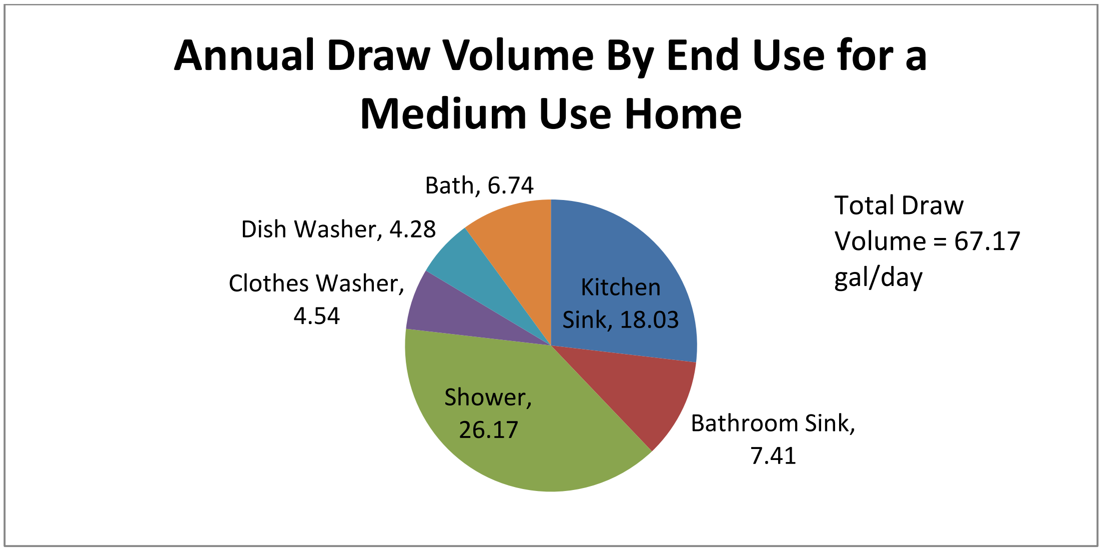


Figure 59: Annual draw volume by end use for a medium use home. For clothes washers and dishwashers the volume of hot water drawn is shown, while for all other draws the mixed volume is shown

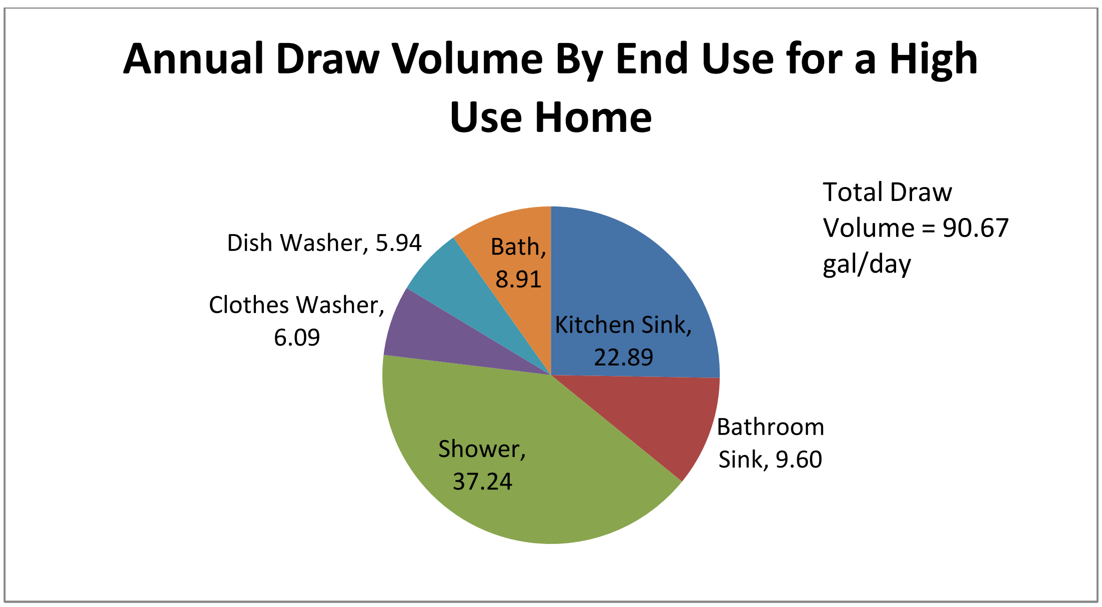


Figure 60: Annual draw volume by end use for a high use home. For clothes washers and dishwashers the volume of hot water drawn is shown, while for all other draws the mixed volume is shown

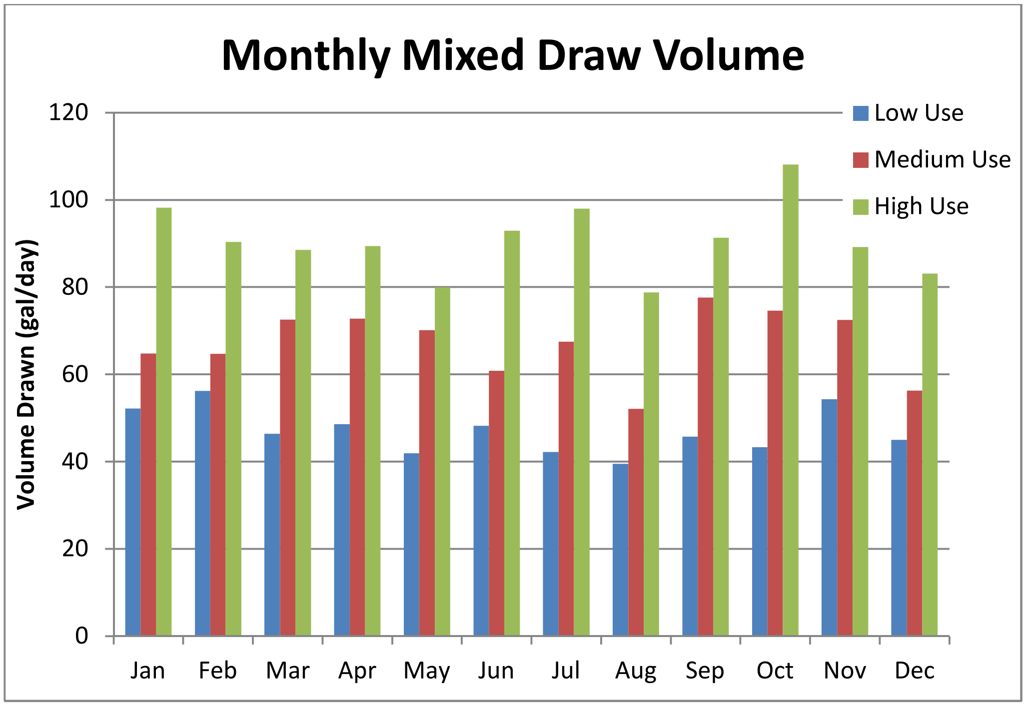


Figure 61: Monthly mixed hot water use for low, medium, and high DHW draw profiles

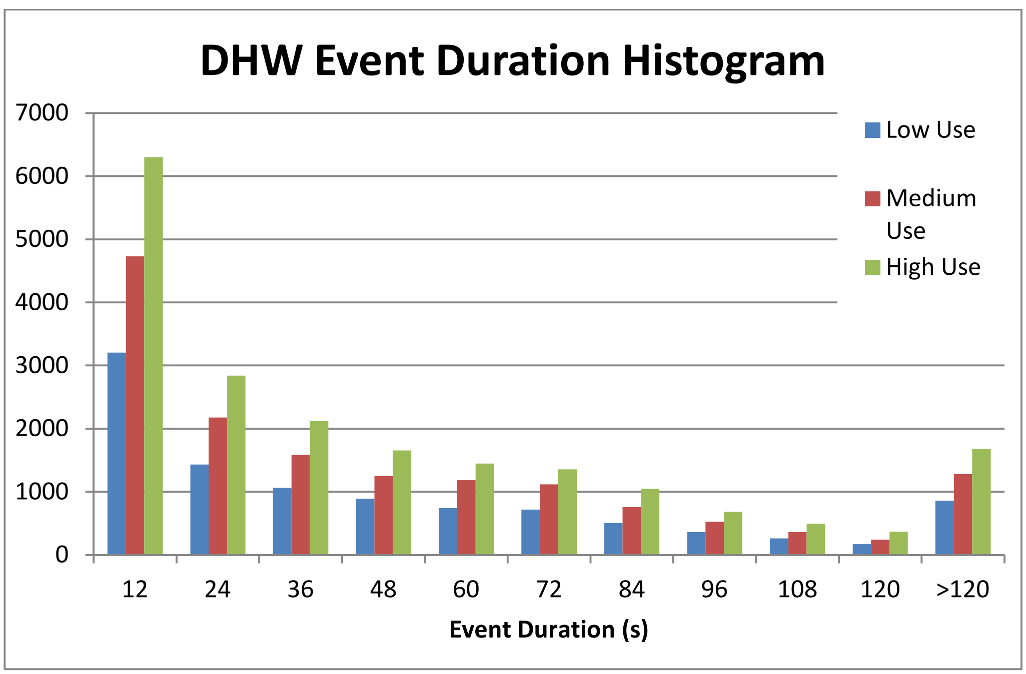


Figure 62: Histogram of DHW event duration

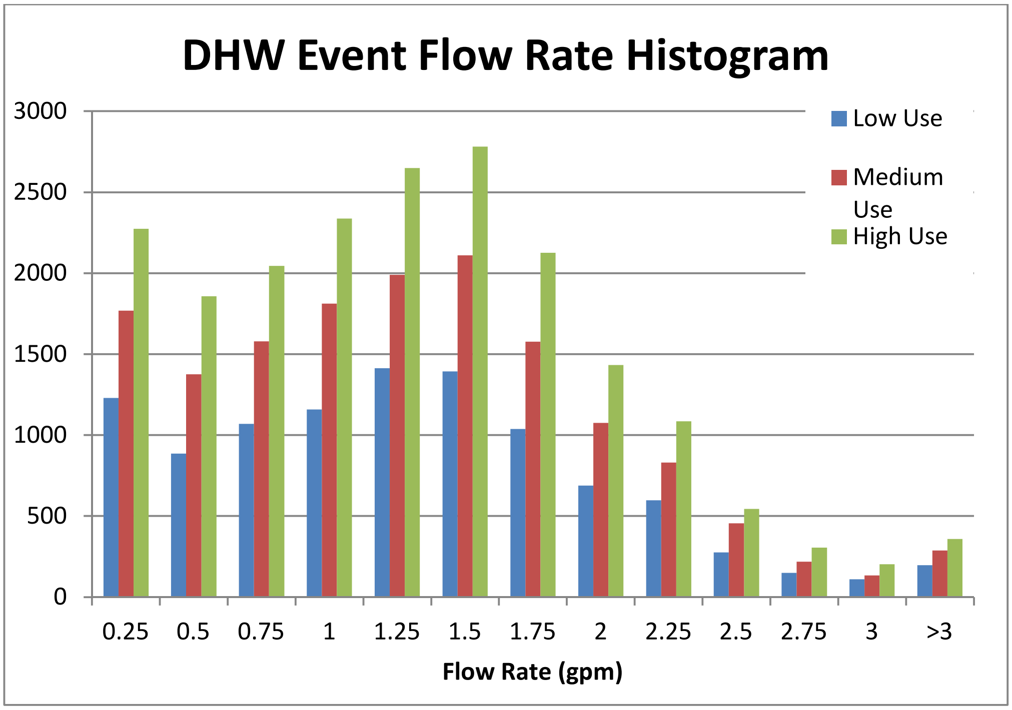


Figure 63: Histogram of DHW event flow rate

Chapter 8: Annual Simulation Results

Annual simulations were performed to answer several key questions about the performance of the various types of water heaters modeled here. For the six representative cities (Chicago, IL, Atlanta, GA, Houston, TX, Los Angeles, CA, Seattle, WA, and Phoenix, AZ), a parametric analysis was performed of all of the types of water heater considered here. They were simulated being subjected to low, medium and high draw profiles while located in either conditioned or unconditioned space. Annual simulations were used to determine what the most energy efficient option as well as the most cost effective option is in each case. Equipment degradation was not considered in these simulations.

For the heat pump water heater, annual simulations were also performed for every site in the continental US and Hawaii for which there is reliable weather data in the TMY3 dataset (63). Simulations were also performed for a gas and electric water heater in the same situation to provide baseline energy consumption for each case. In this study, installation in both conditioned and unconditioned space was considered. Two different sets of heating and cooling equipment were also considered: an air source heat pump and furnace/air conditioner option was considered for each site to represent the likely HVAC equipment for homes with and without gas service. Maps were generated based on the results of this study to illustrate the results and potential energy savings of a HPWH.

Finally, simulations were performed for a HPWH and an electric water heater with a DHW distribution system also modeled to determine what differences there are in the distribution losses between these two technologies.

8.1 Parametric Study Energy Consumption Comparison

When comparing different water heaters in the same location, several factor besides the water heater energy consumption need to be considered. To keep the comparison as even as possible, it is necessary to ensure that all water heaters are meeting the same load. Some technologies, such as heat pump water heaters and tankless water heaters, may have trouble meeting the load due to either sagging outlet temperatures or on time delays. Solar water heaters may provide water at a higher temperature than required because of the higher temperatures allowed in the storage tank. To ensure that all water heaters met the load, normalization energy was included in this analysis. The normalization energy is defined as the thermal energy required to meet the load divided by the instantaneous efficiency of the water heater as shown in the following equation.

$$E_{nrmlz} = \frac{\dot{m}c_p(T_{out} - T_{req})}{\eta} \tag{17}$$

If at any timestep the outlet temperature is lower than the required temperature to meet the load (105 °F for mixed draws and 120 °F for hot draws), the normalization energy is calculated. All of the water heaters required some normalization energy, but the amount varied for different types of water heaters.

In addition, different water heaters have different losses to their surroundings, which impacts the space heating and cooling loads. Changes in heating, cooling, and fan energy consumption are considered in all comparisons between different technologies. The overall energy consumption can therefore be calculated by the following equation:

$$E_{WH,net} = E_{WH} + E_{nrmlz} + \Delta E_{heat} + \Delta E_{cool} + \Delta E_{fan} \tag{18}$$

In the above equation, $E_{WH,net}$ is the overall energy consumption, E_{WH} is the water heater energy consumption, E_{nrmlz} is the normalization energy consumption, E_{heat} is the heating energy consumption, E_{cool} is the heating energy consumption, and E_{fan} is the fan energy consumption. For each comparison, the change in space heating, cooling, and fan energy consumption are calculated relative to the base case (gas storage for gas water heaters and electric storage for electric water heaters). This provides the overall net energy consumption of one water heater relative to another. For electric water heaters, an air source heat pump is utilized for both heating and cooling. For gas water heaters, a furnace provides heating and an air conditioner provides cooling. This is based on the assumption that if a home has natural gas available, they will use it for both space and water heating. Therefore, electric and gas water heater energy consumption cannot be directly compared since the change in space heating and cooling energy consumption is different in these cases, and switches in water heating fuel sources (say from a gas storage to a HPWH) are not considered here.

For gas water heaters, both gas and electricity consumption needs to be considered. Tankless, condensing, and solar water heaters all consume some electricity for controls, venting fans, freeze protection, or pumps, depending on the technology. In addition, the cooling and fan energy consumption differences are in electricity, while the water heater, normalization, and heating energy consumption are gas consumption. To allow these water heaters to be compared, all comparison are done on a source energy basis. National average site to source multipliers of 3.365 and 1.092 are used for electricity and natural gas respectively (51).

For water heaters installed in unconditioned space, unconditioned space is defined as a basement if the home has one or the garage otherwise. Homes in Chicago, Seattle, and Atlanta have basements, while those in Los Angeles, Houston, and Phoenix do not. Basements are much more closely linked to conditioned space and ground temperature than garages, which lead to smaller temperature swings in these spaces when compared to garages. Additionally, the space heating and cooling impact of water heaters installed in a basement is larger than that of a water heater installed in a garage.

In sections 8.1.1-8.1.7, the water heater site energy consumption is examined independent of normalization energy and changes in heating, cooling, and fan energy consumption. This allows for analysis of how climate, draw profile, and water heater installation location affect energy use. In sections 8.1.8 and 8.1.9, electric and gas water heaters are compared on a source energy basis to determine what type of water heater is optimal from an energy perspective in each case. Since in many cases the efficiency is a function of the draw volume, the volume of water drawn by the water heater for every scenario investigated here is give in Appendix H.

8.1.1 Electric Storage Water Heater

For electric storage water heaters, the largest factor impacting the efficiency of these units is the amount of water drawn. As the amount of water drawn increases, the amount of energy delivered increases and the ratio of useful energy (delivered hot water) to wasted energy (tank losses) increases. This leads to the higher efficiency in climates with colder mains water temperatures such as Chicago when compared to locations with warmer mains water temperatures such as Phoenix. This phenomenon can be seen in Figure 64. The energy consumption is also much larger in cases with colder mains temperature due to more energy being required to bring the water up to a useful temperature. When these water heaters are installed in unconditioned space, the tank losses vary depending on the space temperature. This leads to lower tank losses in cooling dominated climates such as Phoenix and higher losses in heating dominated climates such as Chicago. This change in tank losses impacts the efficiency, leading to the roughly uniform efficiency shown in Figure 65. It also increases the energy consumption of the water heater in cold climates and reduces it in warm climates due to the change in tank losses.

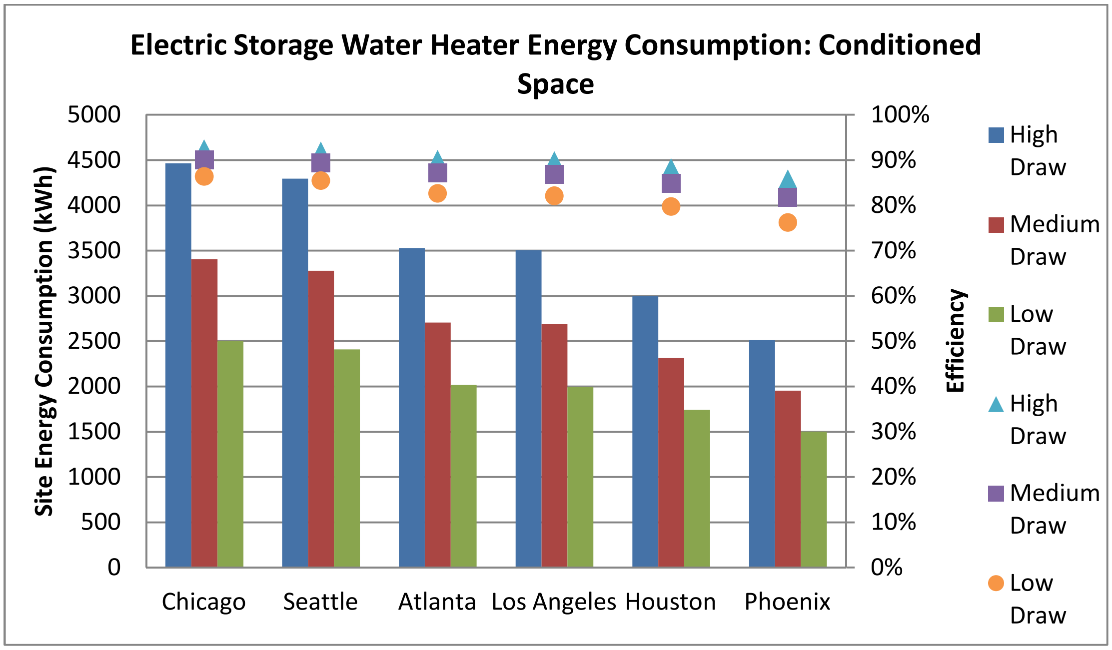


Figure 64: Electric water heater annual energy consumption in conditioned space

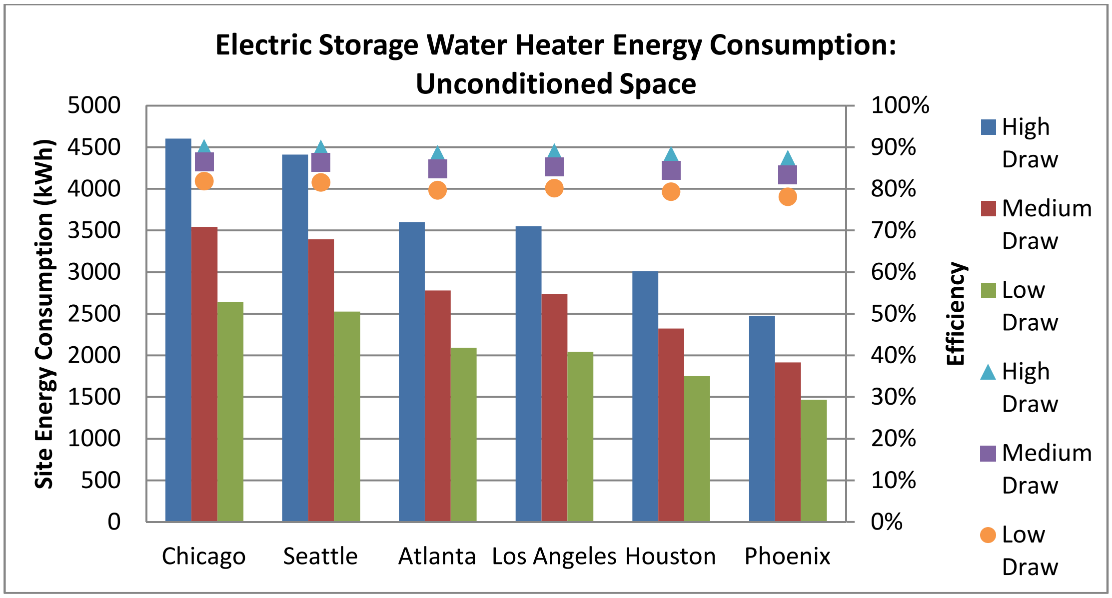


Figure 65: Electric water heater annual energy consumption in unconditioned space

8.1.2 Heat Pump Water Heater

Heat pump water heaters have a large variety of factors that affect their efficiency. The efficiency of the heat pump is impacted by both mains water temperature and the wet bulb temperature of the surrounding air. The overall efficiency of the HPWH as a system is also impacted by how much it can use the heat pump instead of the electric elements. For large draws, the heat pump will not be powerful enough for the tank to recover quickly, leading to the elements turning on. As a result, HPWHs are different from most other water heaters in that their efficiency does not always increase with draw volume. As can be seen in Figure 66 and Figure 67, the most efficient case for HPWHs is the medium draw profile (in the case of Phoenix, medium and high draw cases are roughly equivalent in terms of efficiency).

To help illustrate this phenomenon further, the daily HPWH system COP for homes in Houston with the HPWH in conditioned space for low, medium, and high draw profiles is shown in Figure 68, Figure 69, and Figure 70 respectively. The first thing of note in all of these figures is that there are generally two discrete groups: the upper group is when the heat pump fraction (HPF, defined as the amount of heat added to the tank by the heat pump divided by the total amount of heat added by the heat pump and the elements) is equal to one, while the lower group is when the heat pump fraction is less than one. For this particular HPWH, once the electric elements come on, they stay on until the tank has fully recovered, leading to very few cases where a heat pump fraction just slightly below 1 occurs. There is a much higher chance of the HPF being below one at higher draw volumes as the electric elements are triggered by the tank having had enough water drawn to require the faster recovery rate of the elements as opposed to the heat pump.

In the low use case (Figure 68), there are relatively few days with a HPF less than one. However, the lower use also leads to lower efficiency as the system COP of the HPWHs trends with the log of the daily draw volume. In the medium use case (Figure 69), there are a few more points with a HPF less than one, but the higher draw volume leads to a higher average system COP that makes up for this difference. In the high use case (Figure 70) there are significantly more days with a HPF less than one, leading to the lower annual efficiency. While only the case of a home in Houston with the HPWH in conditioned space is shown here, this same general trend is seen in all cases.

When the HPWH is in conditioned space, the ambient air temperature is kept between 71-76 °F while mains water temperature and humidity vary. This provides more consistent inlet air conditions, leading to the lower variability in efficiency between sites for the conditioned space case than the unconditioned space case. However, changes in the space heating and cooling energy consumption are not taken into account here. These changes increase the energy consumption of space conditioning equipment in heating dominated climates and lower it in cooling dominated climates.

While the efficiency of the heat pump water heater is increased by colder mains temperature, the main factor impacting the efficiency is the inlet air wet bulb temperature. This leads to Houston, a hot and humid location, having the highest efficiency, while Chicago, a cold climate, has the lowest efficiency. In addition to efficiency increasing with wet bulb temperature, there is also a lower and upper limit on the ambient air temperature. Above or below these limits (45-120 °F), the heat pump will not operate. This leads to portions of the year where the system behaves identically to an electric water heater, especially in the case of cold climates such as Chicago.

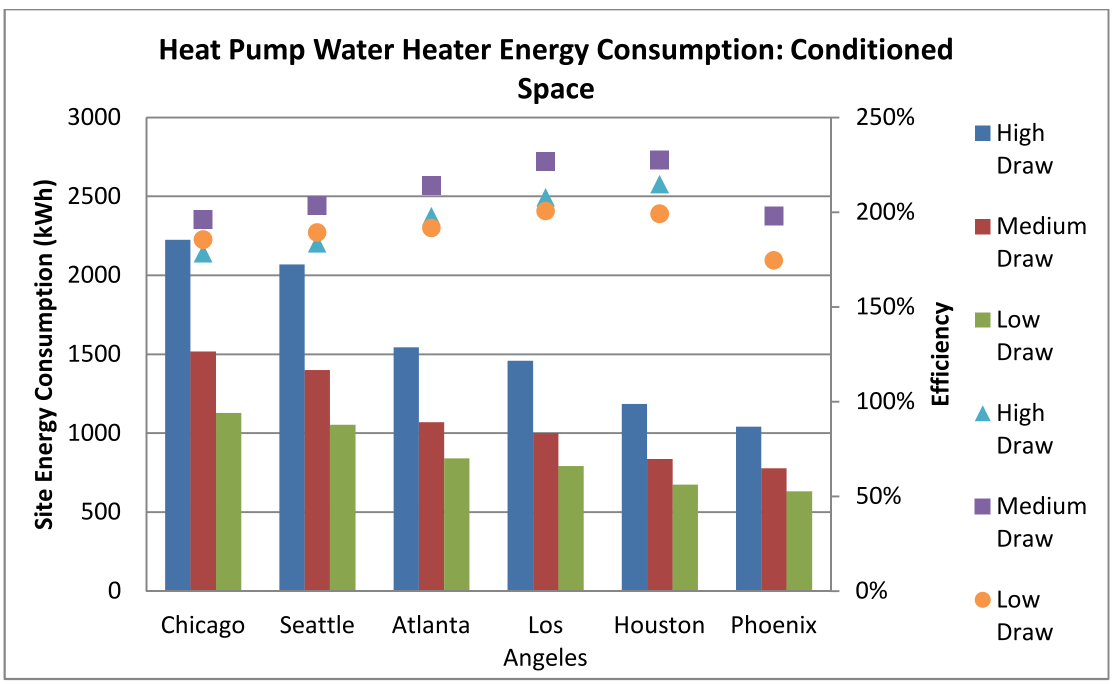


Figure 66: Heat pump water heater annual energy consumption in conditioned space

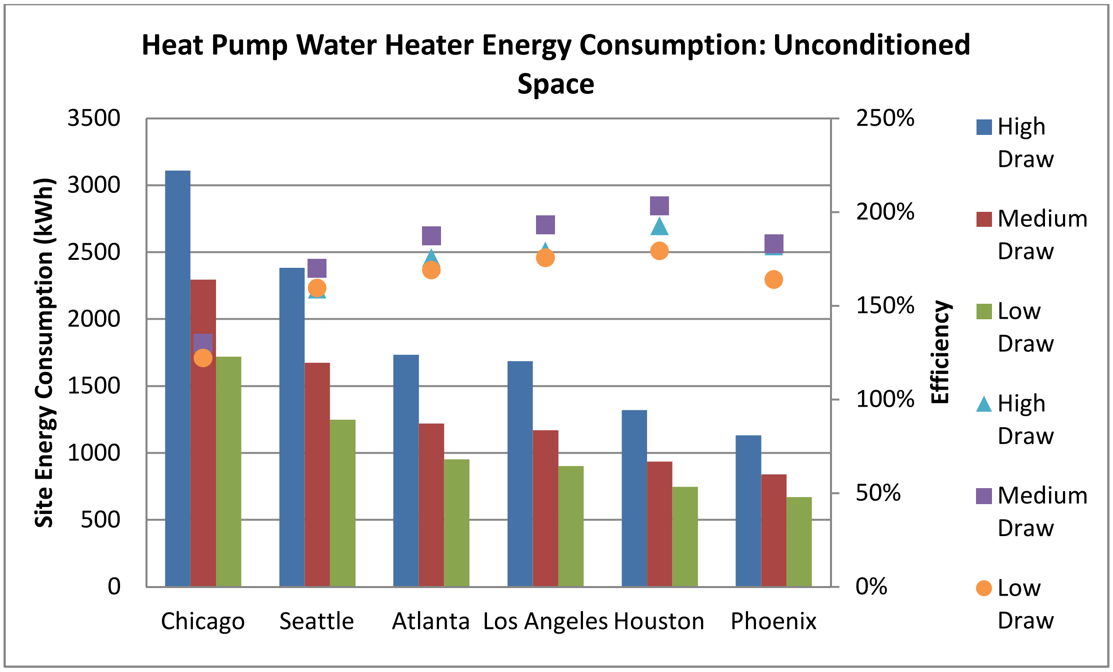


Figure 67: Heat pump water heater energy annual consumption in unconditioned space

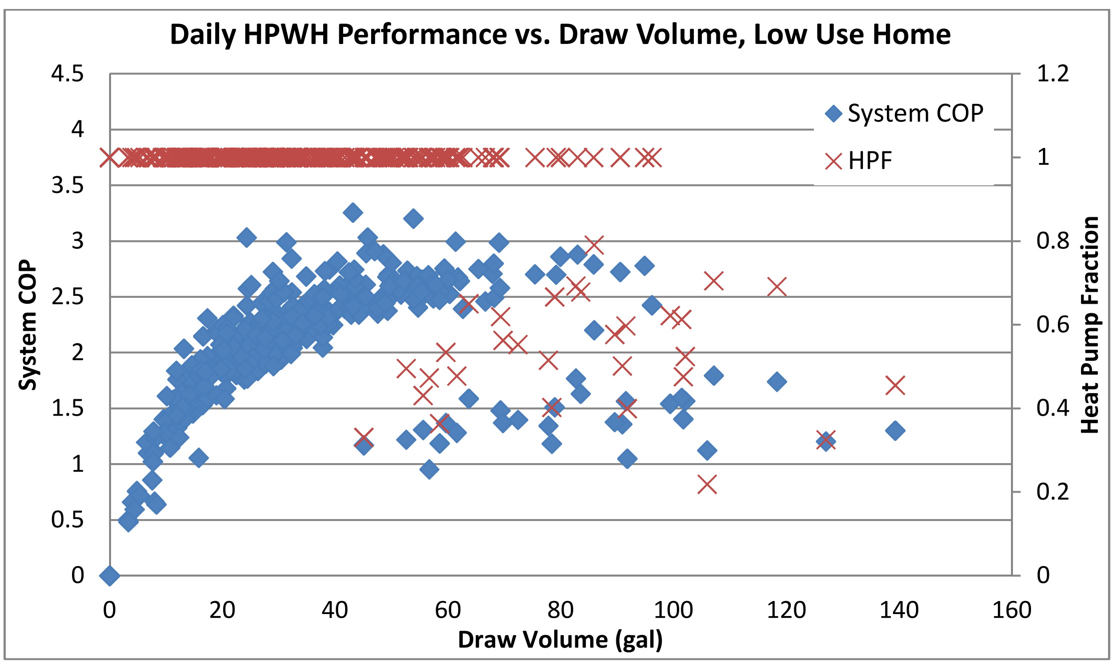


Figure 68: Daily HPWH efficiency for a low use home in Houston

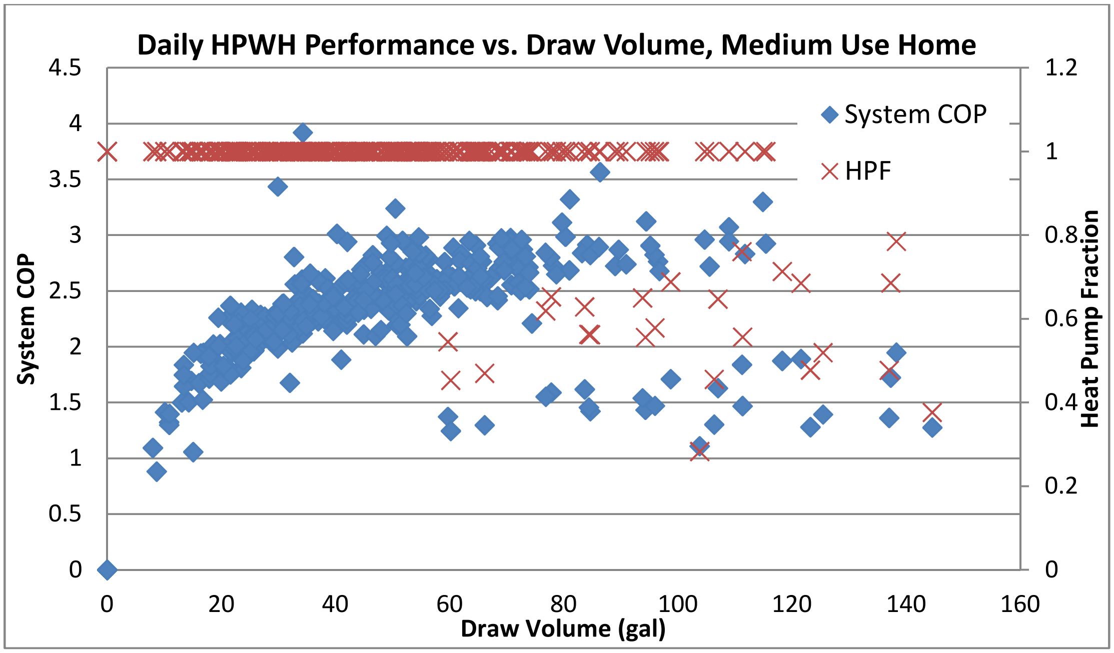


Figure 69: Daily HPWH efficiency for a medium use home in Houston

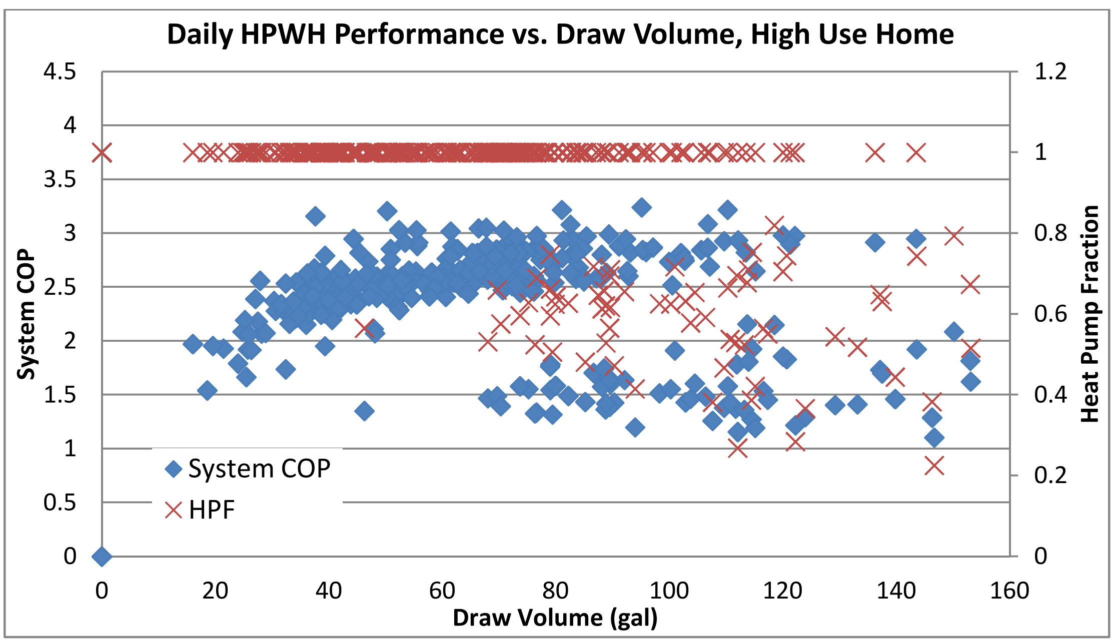


Figure 70: Daily HPWH efficiency for a high use home in Houston

8.1.3 Solar Water Heater with Electric Backup

For solar water heaters, the main driver for efficiency (solar fraction, denoted as SF) is the amount of solar radiation received. Solar fraction is defined as the amount of energy delivered to the storage tank by the solar collector divided by the total energy input into the tank. For this study, solar fraction can be written as:

$$SF = \frac{E_{HX}}{E_{HX} + \left(E_{WH}/\eta \right)} \tag{19}$$

In the above equation, E_{HX} is the energy delivered to the storage tank from the heat exchanger, E_{WH} is the energy consumed by the water heater, and η is the conversion efficiency of the heating device for the water heater. For electric water heaters, η is equal to 1. This definition is slightly different from the definition used by the Solar Ratings and Certification Corporation (SRCC) for rating solar water heaters, which uses the Energy Factor of the backup unit as part of the rating. However, it is in the spirit of the SRCC rating procedure, which defines the solar fraction as "the portion of the total conventional hot water heating load (delivered energy and tank standby losses) provided by solar energy." (64)

Solar water heaters are different from gas and electric storage water heaters, where efficiency is largely driven by the amount of hot water drawn. Instead, the solar fraction is largely driven by the amount of solar radiation at the site. As a result, the trends in efficiency and energy consumption that were previously seen no longer appear as can be seen in Figure 71 and Figure 72. A map of the solar radiation across the US is provided in Figure 73 to help illustrate this trend. In addition to the impact of the change in the amount of solar radiation available at each site the latitude plays an important role. For solar water heaters, the optimal angle to mount the collector at is the same as the latitude. However, all of the solar water heaters considered here are mounted flush with the roof, which has a 6:12 pitch (26.57°) at all locations. This means that the farther north the solar water is, the less optimal this installation angle is. However, having the collector pitched at an angle smaller than the latitude provides more energy from the solar water heater during the winter and less in the summer, which is useful in combined space and water heating applications and when mains water temperatures become lower during the winter months.

For all sites, the solar fraction decreases with increasing draw volume. This is because the solar collector can supply a set amount of energy over the course of a year based on the amount of solar radiation it receives. As the usage increases, more energy is required and the percentage coming from the collector becomes less. This is especially true during winter months, when the demand is larger due to the lower mains water temperature and there is less solar radiation available.

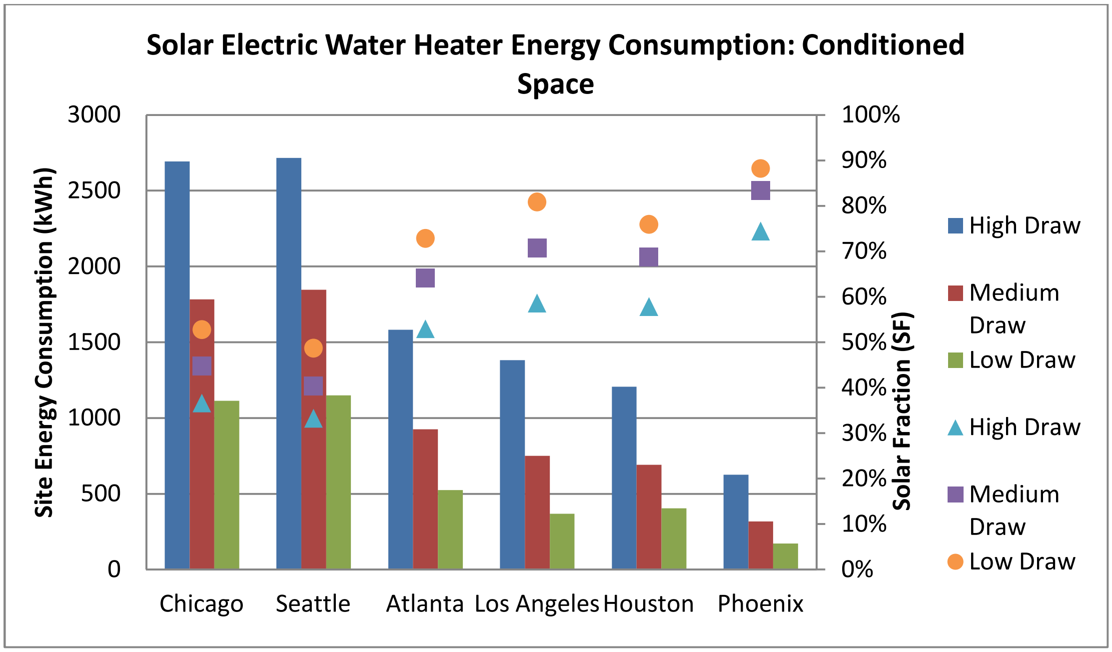


Figure 71: Solar water heater with electric backup annual energy consumption in conditioned space

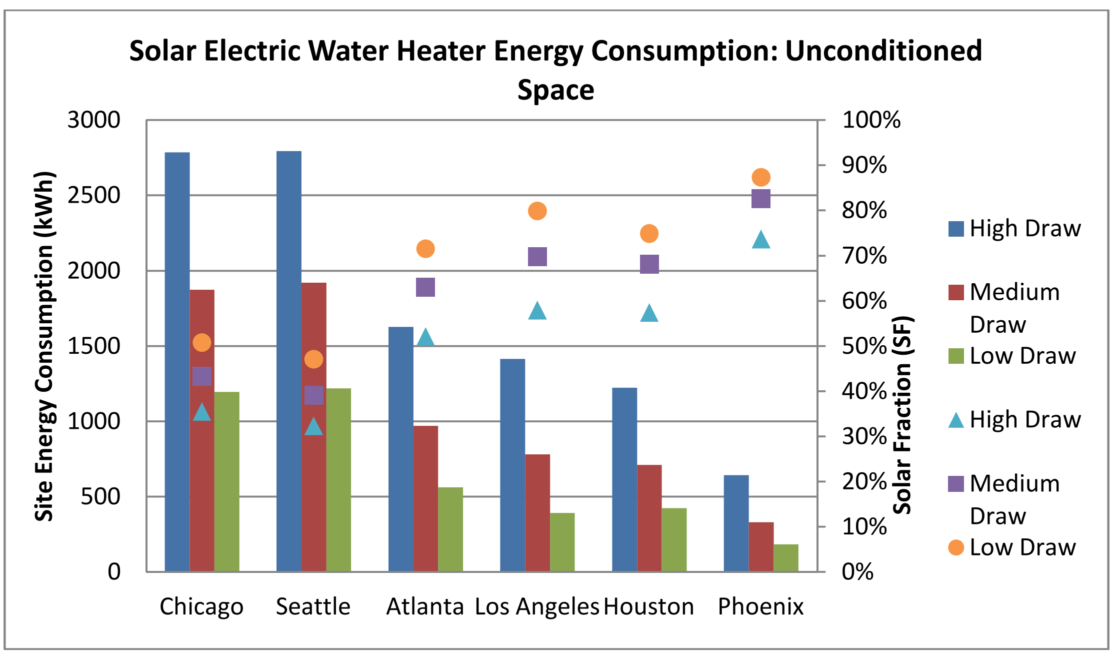


Figure 72: Solar water heater with electric backup annual energy consumption in unconditioned space

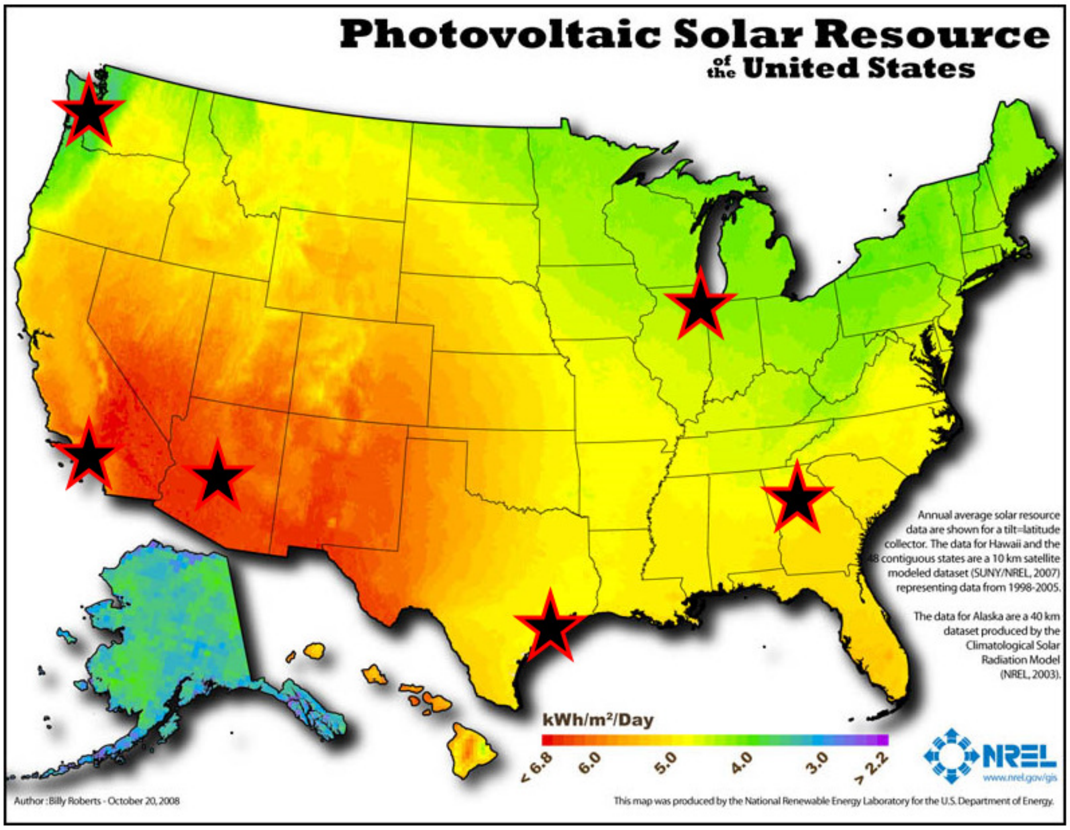


Figure 73: Average solar resource in the US (65). Stars denote the representative cities used in this study

8.1.4 Gas Storage Water Heater

Gas storage water heaters behave very similarly to electric storage water heaters. However, they have a lower efficiency because of higher tank losses (due in part to the central flue) and the combustion efficiency of gas. The same trends of higher efficiency with higher use and an increase or decrease in energy consumption in unconditioned space depending on whether the climate is heating or cooling dominated are seen for gas storage water heaters.

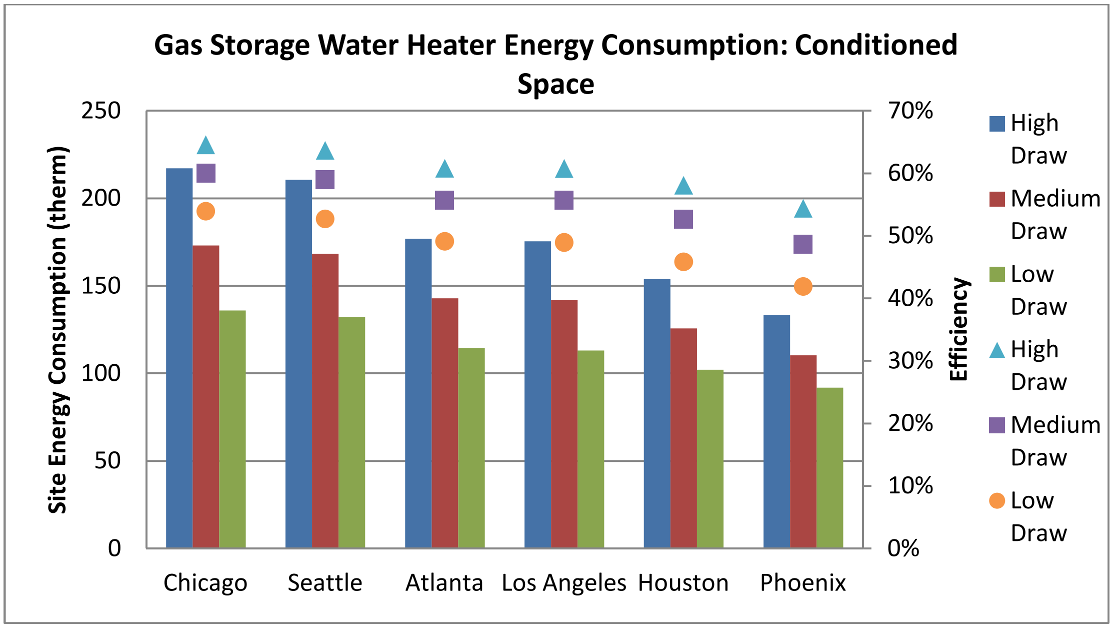


Figure 74: Gas water heater annual energy consumption in conditioned space

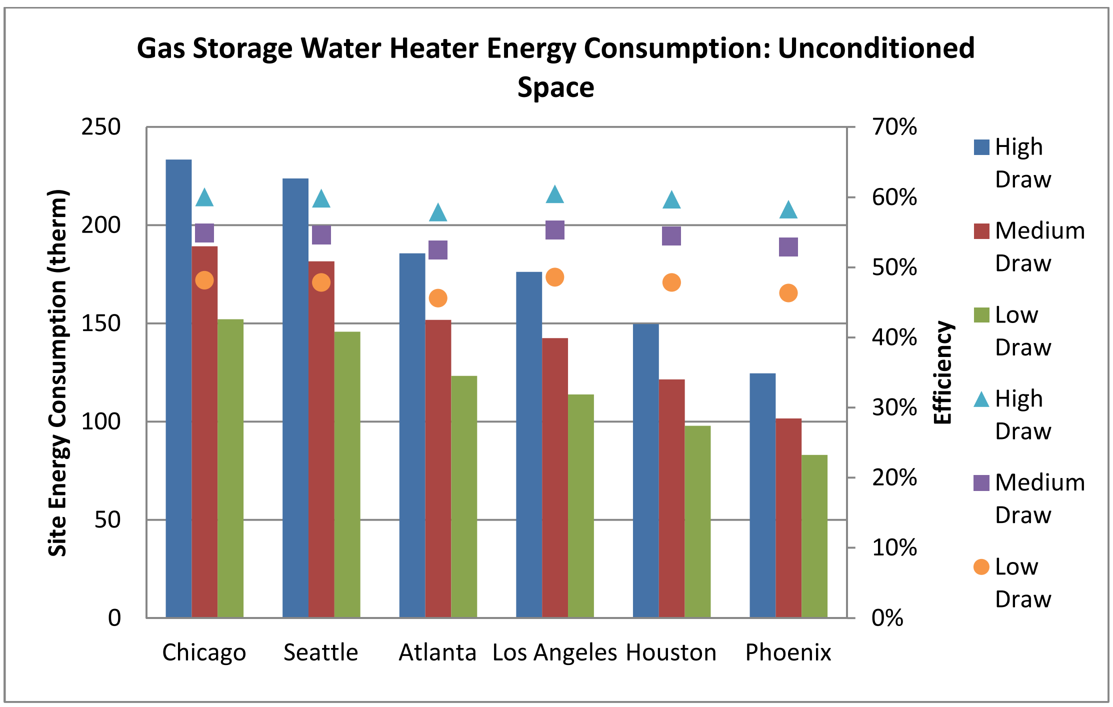


Figure 75: Gas water heater annual energy consumption in unconditioned space

8.1.5 Gas Tankless Water Heater

For tankless water heaters, the efficiency is also a function of the draw volume, with higher efficiencies at higher draw volumes. However, the draw volume is a less significant factor on the overall efficiency than in the case of storage water heaters as can be seen in Figure 76. This is because there are no standby losses, although there are losses from the water heater to ambient air during and after draws. There are also losses from heating the relatively massive heat exchanger. These losses can be significant for short draws where little of the heat from the burner actually goes into the water.

In unconditioned space there are a few other factors in play. For one, the warmer ambient air temperature in unconditioned space can reduce the amount of losses associated with heating and cooling the heat exchanger since the heat exchanger will start at a warmer temperature. Freeze protection energy can also have an impact on the overall energy consumption. Freeze protection was required in Chicago, Houston, and Phoenix, although the amount of energy consumed in Phoenix and Houston was very small (note that Chicago, Seattle, and Antlanta have their tankless water heater in the basement if it is in unconditioned space, while Los Angeles, Houston, and Phoenix locate the water heater in the garage).

There were some issues with implementing the freeze protection algorithm in this model. When freeze protection was added to the model, the freeze protection would rapidly cycle on and off and keep the water heater at 38 °F (the temperature at which freeze protection comes on) instead of heating the unit up to the set point temperature. To capture the freeze protection energy consumption, freeze protection was assumed to be on during any time when the freeze protection was cycling on and off. This will somewhat overestimate the energy consumed by the freeze protection heaters, but this energy was ~1% of the total water heater energy consumption at most and this assumption should have a negligible result on the annual results.

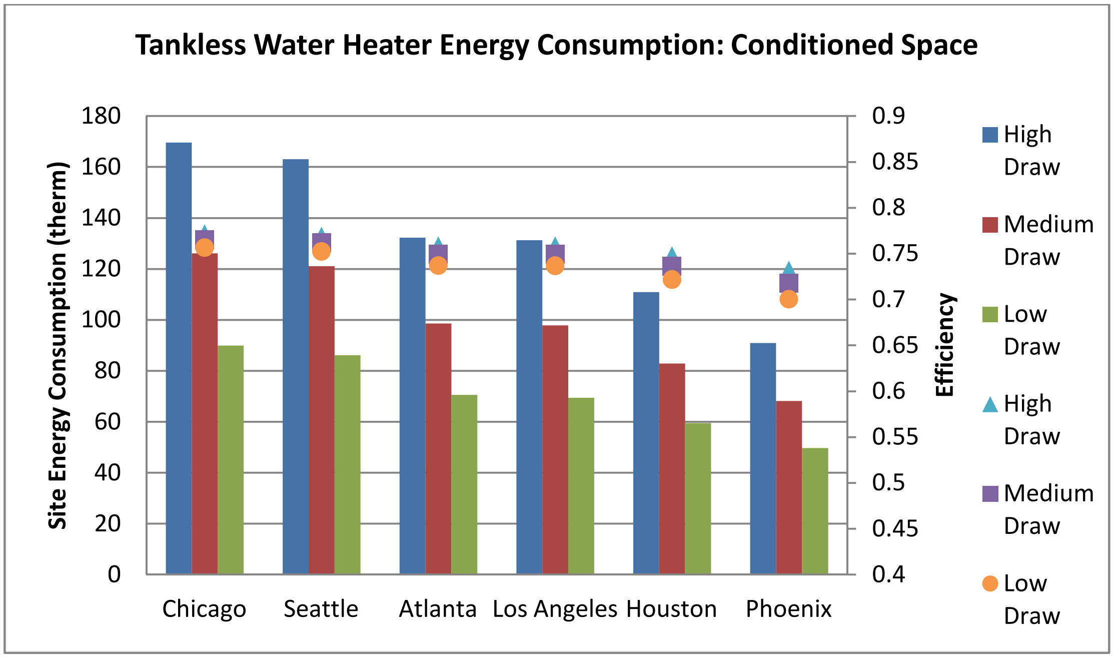


Figure 76: Tankless water heater annual energy consumption in conditioned space

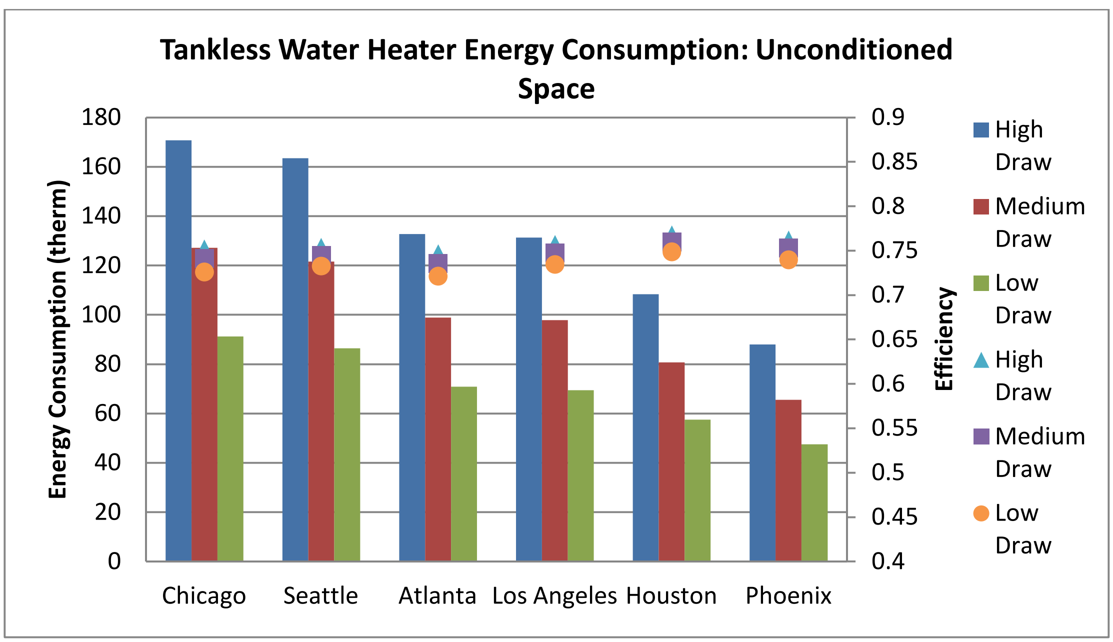


Figure 77: Tankless water heater annual energy consumption in unconditioned space

8.1.6 Condensing Water Heater

The behavior of condensing water heaters is very similar to that of gas storage water heaters. The efficiency is largely a function of the use, leading to higher efficiencies with higher use draw profiles. Efficiency is more strongly a function of mains temperature in this case because the conversion efficiency (the efficiency of turning natural gas into heat stored in the tank) is impacted by the average tank temperature. Condensing water heaters are more efficient than regular gas storage water heaters for two reasons. The conversion efficiency is higher due to the recovery of latent heat from the flue gas (in this case, the efficiency is generally between 92-96% depending on the average tank temperature). Standby losses are also lower due to the flue being replaced by the condensing heat exchanger. This leads to a much higher annual efficiency than can be achieved by a typical gas storage water heater.

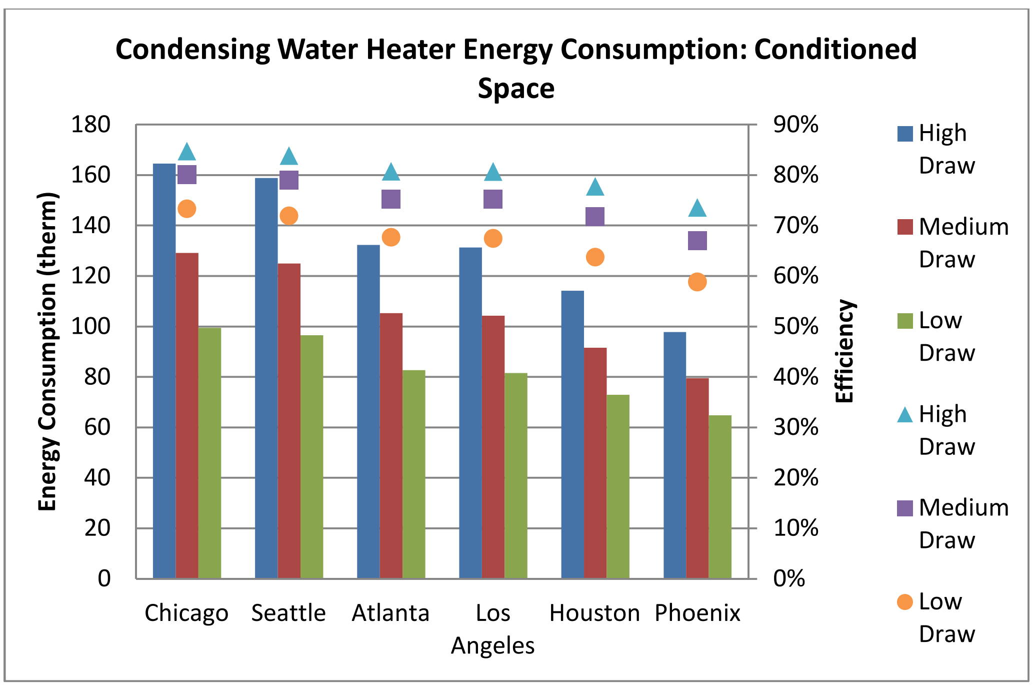


Figure 78: Condensing water heater annual energy consumption in conditioned space

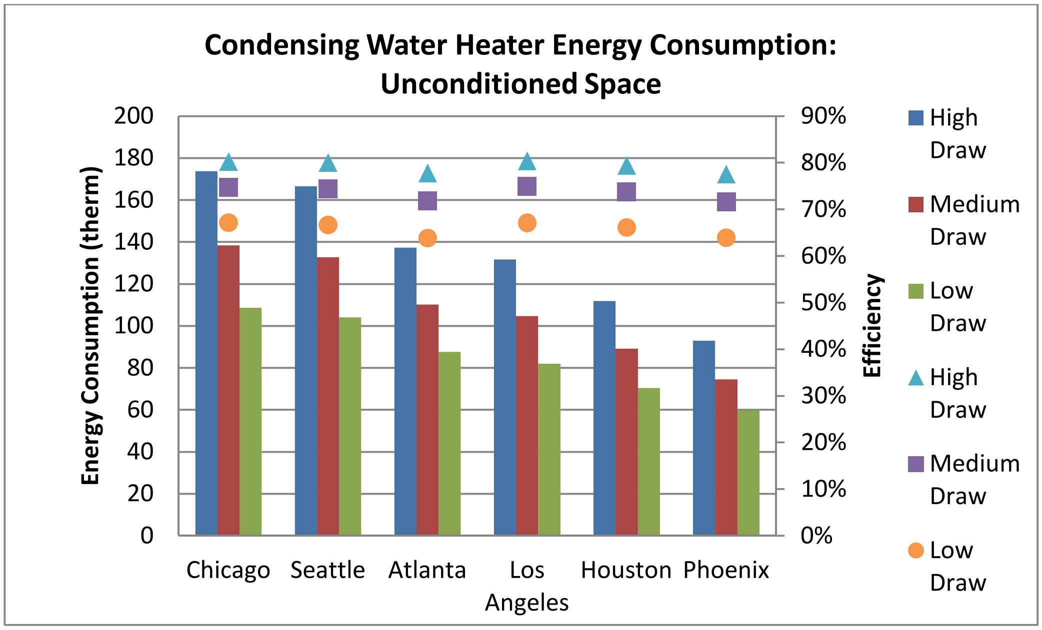


Figure 79: Condensing water heater annual energy consumption in unconditioned space

8.1.7 Solar Water Heater with Gas Backup

In many ways, the solar water heater with gas backup is very similar to the case of a solar water heater with electric backup as shown in Figure 80 and Figure 81. Efficiency (solar fraction) is driven by the amount of solar radiation received (see Figure 73) as opposed to the draw volume. However, the solar water heater with gas backup is a two tank system consisting of a solar preheat tank in series with a standard gas storage water heater. This leads to much higher standby losses and consequently lower annual efficiency than solar water heater with electric backup. While most of the same trends (higher solar fraction with lower use, and higher average annual solar radiation) are the same as for solar water heaters with electric backup, the behavior for Phoenix is different. In Phoenix, the most efficient case is actually the medium draw case in both conditioned and unconditioned space.

To help explain what is happening in Phoenix, the monthly solar fraction for both

Phoenix and Los Angeles are provided in Figure 82 and Figure 83. In a two tank system, the second tank can only be charged by the solar water heater if there is a draw. This means that if there are no draws for a long period, the gas burner must fire to make up the standby losses. In the case of Phoenix, there are several months where the solar fraction is 1 for medium or high draws while it is below 1 for the low draw case. This is because the second tank is not being charged by the solar water heater in the low draw case, while it is being charged in medium and high draw cases. Standby losses cannot be made up by solar in medium or high draw cases as well. With low use case, there is more time for the second tank to stay idling and have its standby loss compensated by gas, thus lowers the annual efficiency. This situation is exacerbated when solar fraction becomes 1 throughout a few months of the year. This only becomes significant in locations where the solar fraction is 1. In the case of Los Angeles, the low draw case always ends up being more efficient than the medium or high draw case as there is never a month for which the solar fraction is 1.

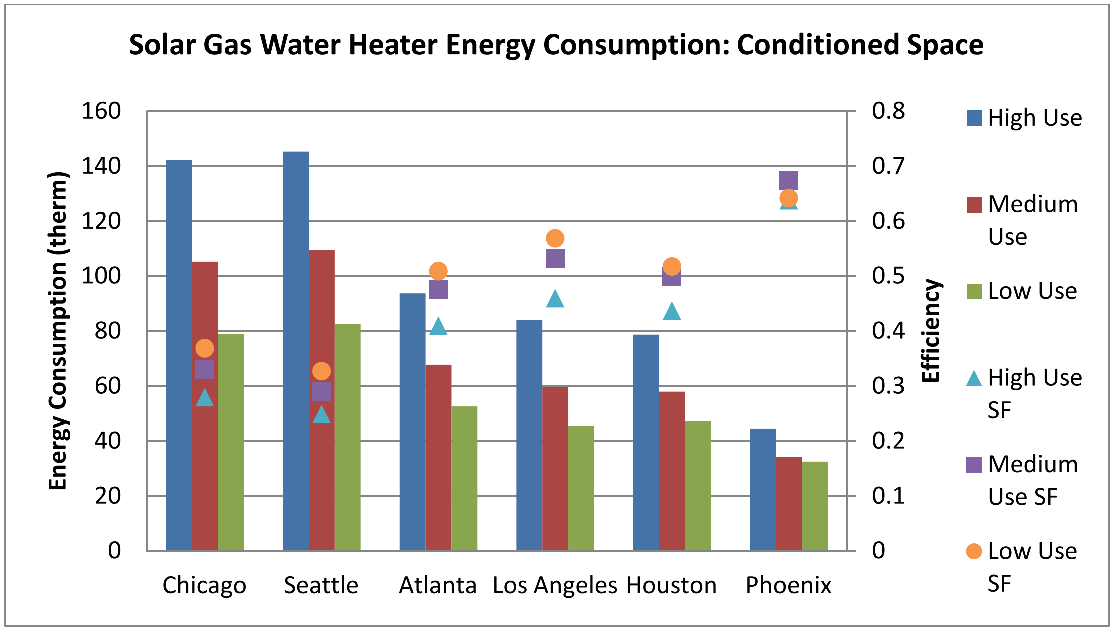


Figure 80: Solar water heater with gas backup annual energy consumption in conditioned space

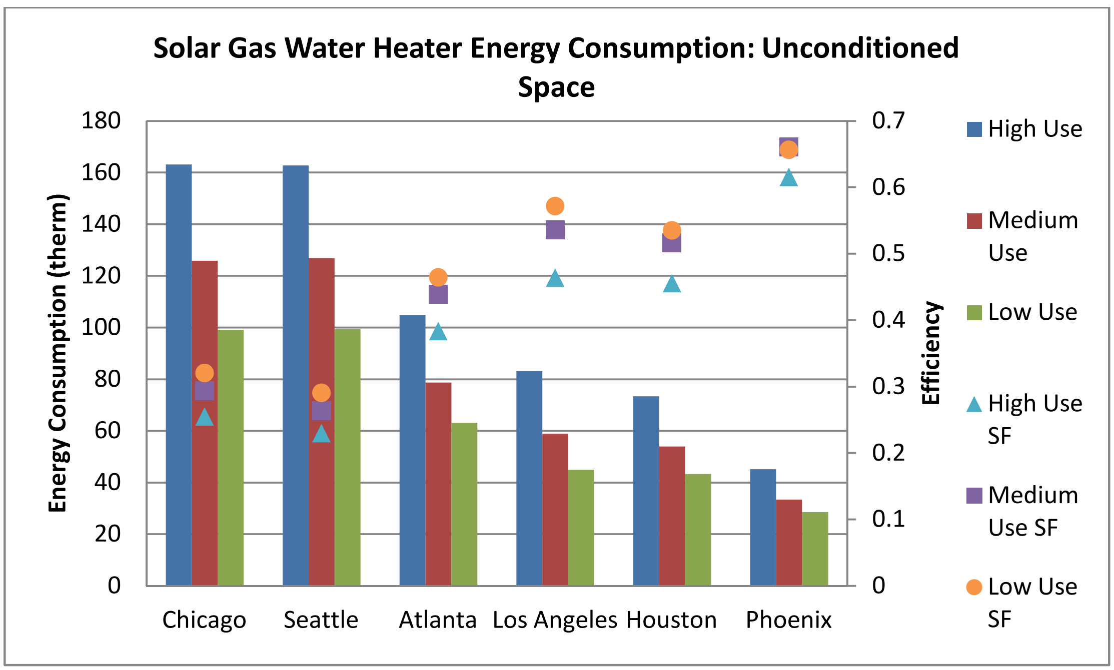


Figure 81: Solar water heater with gas backup annual energy consumption in unconditioned space

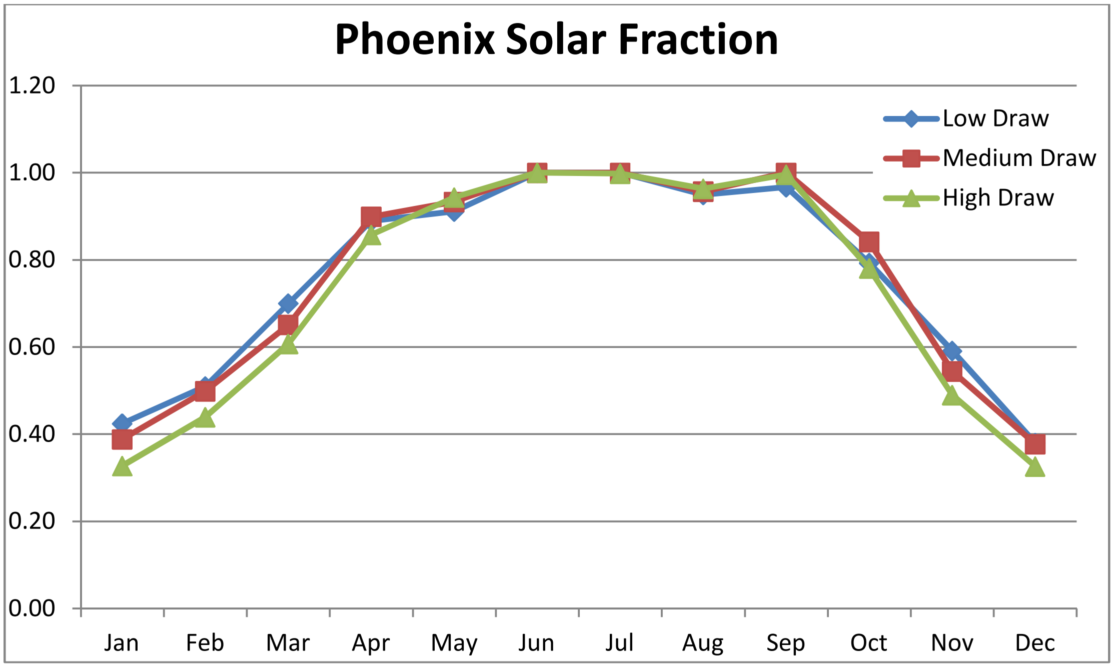


Figure 82: Monthly solar fraction in Phoenix

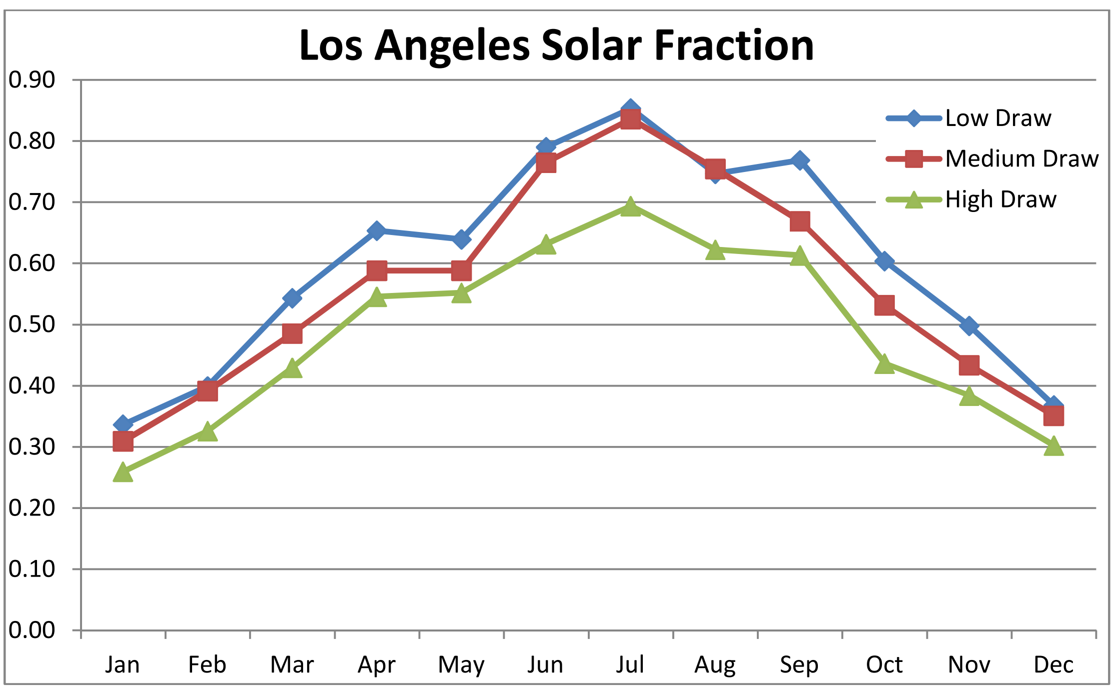


Figure 83: Monthly solar fraction in Los Angeles

8.1.8 Comparison of Electric Water Heaters

When comparing different water heaters, changes in space heating, cooling, and fan energy consumption is taken into account as well as the normalization energy. In addition, any secondary energy consumption (for example, the energy consumed by the pump on the collector loop for solar water heaters) is also taken into account. Since electric water heaters are assumed to be installed in homes without gas service and have an air source heat pump instead of a furnace/AC, gas and electric water heaters are not directly compared here. The most efficient electric water heater in each case is shown in Table 18 and the net energy consumption for each water heater is provided in Table 19-Table 24. In general, solar water heaters are the most energy efficient option for low users of hot water as the solar fraction (efficiency) is larger at low use. Solar water heaters are also a better option in unconditioned space, where HPWHs tend to be less efficient than when they are located in conditioned space. This is true in all locations except Seattle, which has very little sun and therefore a low solar fraction and higher water heater energy consumption.

In conditioned space, for medium and high draw profiles heat pump water heaters often work out to be a more efficient option than solar water heaters. The HPWH provides a net cooling in all climates, which is beneficial in warm climates (which also have the most sun). The solar water heater always provides net heating through losses both in the pipes connecting the collector to the tank and higher tank losses due to the higher storage temperature. This is a detriment in cooling dominated climates, which helps make HPWHs more attractive in warm climates.

Location	Conditioned Space			Unconditioned Space		
	Low	Medium	High	Low	Medium	High
Chicago	Solar	Solar	HPWH	Solar	Solar	Solar
Seattle	Solar	HPWH	HPWH	Solar	HPWH	HPWH
Atlanta	Solar	Solar	HPWH	Solar	Solar	Solar
Los Angeles	Solar	Solar	Solar	Solar	Solar	Solar
Houston	Solar	HPWH	HPWH	Solar	Solar	Solar
Phoenix	Solar	HPWH	HPWH	Solar	Solar	Solar

Table 18: Most efficient water heating option for each case

Installation Location	Draw Volume	Electric Storage	HPWH	Solar Electric	Best
Conditioned	Low	8465	4894	4131	Solar
Conditioned	Medium	11500	6580	6445	Solar
Conditioned	High	15134	9547	9771	HPWH
Unconditioned	Low	8933	5982	4228	Solar
Unconditioned	Medium	11968	8039	6560	Solar
Unconditioned	High	15604	11281	9925	Solar

Table 19: Source energy consumption (in kWh) for electric water heaters in Chicago

Installation Location	Draw Volume	Electric Storage	HPWH	Solar Electric	Best
Conditioned	Low	8136	4922	4340	Solar
Conditioned	Medium	11070	6587	6763	HPWH
Conditioned	High	14554	9569	9937	HPWH
Unconditioned	Low	8525	4771	4350	Solar
Unconditioned	Medium	11459	6393	6799	HPWH
Unconditioned	High	14945	9380	10004	HPWH

Table 20: Source energy consumption (in kWh) for electric water heaters in Seattle

Installation Location	Draw Volume	Electric Storage	HPWH	Solar Electric	Best
Conditioned	Low	6795	2953	2018	Solar
Conditioned	Medium	9123	3783	3391	Solar
Conditioned	High	11901	5689	5709	HPWH
Unconditioned	Low	7046	3274	2120	Solar
Unconditioned	Medium	9376	4250	3518	Solar
Unconditioned	High	12154	6354	5837	Solar

Table 21: Source energy consumption (in kWh) for electric water heaters in Atlanta

Installation Location	Draw Volume	Electric Storage	HPWH	Solar Electric	Best
Conditioned	Low	6721	3360	1636	Solar
Conditioned	Medium	9068	4320	2971	Solar
Conditioned	High	11835	6349	5194	Solar
Unconditioned	Low	6881	3362	1624	Solar
Unconditioned	Medium	9228	4352	2989	Solar
Unconditioned	High	11996	6499	5229	Solar

Table 22: Source energy consumption (in kWh) for electric water heaters in Los Angeles

Installation Location	Draw Volume	Electric Storage	HPWH	Solar Electric	Best
Conditioned	Low	5863	2139	1862	Solar
Conditioned	Medium	7795	2493	2851	HPWH
Conditioned	High	10105	3865	4637	HPWH
Unconditioned	Low	5894	2774	2067	Solar
Unconditioned	Medium	7827	3455	3051	Solar
Unconditioned	High	10137	5015	4832	Solar

Table 23: Source energy consumption (in kWh) for electric water heaters in Houston

Installation Location	Draw Volume	Electric Storage	HPWH	Solar Electric	Best
Conditioned	Low	5054	1976	1017	Solar
Conditioned	Medium	6576	2364	1566	Solar
Conditioned	High	8459	3314	2678	Solar
Unconditioned	Low	4934	2493	1231	Solar
Unconditioned	Medium	6456	3129	1772	Solar
Unconditioned	High	8338	4336	2884	Solar

Table 24: Source energy consumption (in kWh) for electric water heaters in Phoenix

8.1.9 Comparison of Gas Water Heaters

As when comparing different electric water heaters, changes in space heating, cooling, and fan energy consumption is taken into account as well as the normalization energy. For gas water heaters, a solar water heater is almost always the most efficient option. The most efficient gas water heater in each case is shown in Table 25 and the net energy consumption for each water heater is provided in Table 26-Table 31. The only cases where a solar water heater is not the best option is for low use cases in cooling dominated locations (Phoenix and Houston) when the water heater is located in conditioned space. Solar water heaters have high losses due to the losses from both tanks and the pipes connecting the collector to the solar preheat tank. Tankless water heaters have the smallest impact on space heating and cooling loads since they only have losses while they are operating (from smallest to largest, the impact on space heating and cooling for gas water heaters is tankless, condensing, gas storage, then solar). This impact on space conditioning equipment is significant enough to give tankless water heaters an edge in low use cases where the water heating load, and therefore potential savings, is small.

Location	Conditioned Space			Unconditioned Space		
	Low	Medium	High	Low	Medium	High
Chicago	Solar	Solar	Solar	Solar	Solar	Solar
Seattle	Solar	Solar	Solar	Solar	Solar	Solar
Atlanta	Solar	Solar	Solar	Solar	Solar	Solar
Los Angeles	Solar	Solar	Solar	Solar	Solar	Solar
Houston	Tankless	Solar	Solar	Solar	Solar	Solar
Phoenix	Tankless	Solar	Solar	Solar	Solar	Solar

Table 25: Most efficient gas water heating option in each case

Installation Location	Draw Volume	Gas Storage	Tankless	Condensing	Solar Gas	Best
Conditioned	Low	148.93	125.62	123.31	87.45	Solar
Conditioned	Medium	189.80	169.30	156.77	117.10	Solar
Conditioned	High	238.21	220.54	197.02	157.20	Solar
Unconditioned	Low	166.77	125.38	131.37	88.16	Solar
Unconditioned	Medium	207.59	168.12	164.81	117.46	Solar
Unconditioned	High	255.97	219.92	204.71	158.03	Solar

Table 26: Source energy consumption (in therms) of gas water heaters in Chicago

Installation Location	Draw Volume	Gas Storage	Tankless	Condensing	Solar Gas	Best
Conditioned	Low	145.00	133.16	121.15	85.97	Solar
Conditioned	Medium	184.43	175.01	154.21	116.37	Solar
Conditioned	High	230.86	224.44	192.70	155.80	Solar
Unconditioned	Low	159.75	120.49	127.16	74.86	Solar
Unconditioned	Medium	199.14	163.12	159.33	106.19	Solar
Unconditioned	High	245.51	213.19	197.51	146.18	Solar

Table 27: Source energy consumption (in therms) of gas water heaters in Seattle

Installation Location	Draw Volume	Gas Storage	Tankless	Condensing	Solar Gas	Best
Conditioned	Low	125.44	94.84	95.72	64.93	Solar
Conditioned	Medium	156.60	128.32	120.80	80.95	Solar
Conditioned	High	193.59	168.50	151.22	108.37	Solar
Unconditioned	Low	135.09	95.95	108.12	71.32	Solar
Unconditioned	Medium	166.29	130.28	133.39	87.78	Solar
Unconditioned	High	203.24	170.85	163.76	115.21	Solar

Table 28: Source energy consumption (in therms) of gas water heaters in Atlanta

Installation Location	Draw Volume	Gas Storage	Tankless	Condensing	Solar Gas	Best
Conditioned	Low	123.81	87.88	101.12	70.84	Solar
Conditioned	Medium	155.23	122.17	126.34	83.86	Solar
Conditioned	High	191.99	162.05	156.54	109.68	Solar
Unconditioned	Low	124.63	92.17	101.70	56.57	Solar
Unconditioned	Medium	156.10	126.31	127.09	72.27	Solar
Unconditioned	High	192.86	166.04	157.43	99.19	Solar

Table 29: Source energy consumption (in therms) of gas water heaters in Low Angeles

Installation Location	Draw Volume	Gas Storage	Tankless	Condensing	Solar Gas	Best
Conditioned	Low	111.74	67.00	90.55	83.12	Tankless
Conditioned	Medium	137.57	95.78	110.70	92.29	Solar
Conditioned	High	168.25	129.70	135.23	113.50	Solar
Unconditioned	Low	107.10	73.05	88.53	54.44	Solar
Unconditioned	Medium	132.95	100.33	109.45	66.68	Solar
Unconditioned	High	163.67	132.69	134.81	88.13	Solar

Table 30: Source energy consumption (in therms) of gas water heaters in Houston

Installation Location	Draw Volume	Gas Storage	Tankless	Condensing	Solar Gas	Best
Conditioned	Low	100.49	57.19	82.02	72.88	Tankless
Conditioned	Medium	120.79	80.55	97.87	75.79	Solar
Conditioned	High	145.79	108.63	118.04	88.25	Solar
Unconditioned	Low	90.91	61.47	76.90	37.62	Solar
Unconditioned	Medium	111.20	82.68	93.16	43.97	Solar
Unconditioned	High	136.21	109.03	113.62	57.34	Solar

Table 31: Source energy consumption (in therms) of gas water heaters in Phoenix

8.2 Parametric Study Economic Comparison

Along with looking at the source energy consumption and savings of each water heating option in different installation locations, climates, and draw profile, the economic viability of each case was also examined. Two metrics were used to evaluate these technologies: life cycle cost (LCC) and breakeven cost. The annual operating costs of each technology are also provided so that they could be used in any other economic analysis. LCC gives the cost of installing and using each water heater for the entire analysis period (in this case, 13 years, which is the typical lifetime of a gas or electric storage water heater). Breakeven cost gives what the capital cost of any of the more efficient technologies examined here would need to be for that system to be cost neutral with the baseline (gas or electric storage). A simple way of thinking of these two metrics is that LCC evaluates the technologies as they exist today and breakeven cost shows where they need to go in terms of price to be cost effective in the future.

For this analysis, both new construction and retrofit cases are considered, although only capital costs change for these two options. In addition, both cases with and without incentives are considered. When looking at the impact of incentives, there are 4 cases: no incentives, federal only, local only, and federal and local. There is a separate case for federal only and federal and local incentives since the results from the federal incentive case can be generalized over city and state borders, while those using local incentives cannot. The local incentives only case is included because the current federal water heating incentives for all systems except for solar water heaters (although the solar water heater tax credit will change so that it only applies to homes with photovoltaic systems) are set to expire at the end of 2011 and may not be renewed in 2012. This section is broken down into a discussion of capital costs in new construction and retrofit cases, maintenance costs, annual operating costs, and finally life cycle cost and breakeven cost.

8.2.1 Water Heater Capital Costs

The capital cost for each of these units consists of two major components: the actual equipment cost and the installation cost. To determine the capital costs of these units, several sources were examined. One major source is the 2010 federal rule on residential water heater efficiency (6). This ruling set updated minimum efficiency standards for residential water heaters starting in 2015. As part of this ruling, the cost of all water heating technologies excluding solar was evaluated. These cost projections included detailed installation costs, going over individual costs (for example, adding an electric outlet or a drain pan) for both new and retrofit cases. This was the main source of installation cost information.

Equipment costs for each technology were determined from a mix of the 2010 federal rulemaking and looking at online retailers for the price each water heater is typically sold for. In some cases, the federal rulemaking had a significantly lower equipment cost than what may be typical of the equipment available today as shown in Table 32. For these cases, an equipment cost was determined by looking at the retail price of these units (generally at “big box” retailers such as Loews or Home Depot). The one exception to this is the condensing water heater. The model developed here is based on a high efficiency, very expensive unit. To more accurately represent a typical condensing water heater cost, the equipment cost was based on a more typical unit, the AO Smith Vertex. Since this unit is not carried in "big box" stores currently, the price from more niche suppliers (such as PexSupply.com) was used. Installation costs were derived from the federal rulemaking and itemized lists of cost included here are provided in Table 33 and Table 34.

Water Heater Type	Equipment Cost: 2010 Rulemaking ($)	MSRP (or price online) ($)	Cost Used Here ($)	Installation Cost: New ($)	Installation Cost: Retrofit ($)	Retrofit: Conditioned Space Adder ($)
Gas	450.00	385.00	450.00	934.19	458.04	0.00
Gas Tankless	1109.00	849.00	1109.00	773.16	1463.00	0.00
Gas Condensing	894.65	4580.3	1632.98	635.88	1032.89	0.00
Electric	282.60	282.00	282.60	218.47	364.11	0.00
HPWH	1169.35	1400.00	1400.00	287.65	433.29	383.52
Solar Gas	0.00	0.00	0.00	8074.19	7547.42	0.00
Solar Electric	0.00	0.00	0.00	6690.00	6690.00	0.00

Table 32: Water heater equipment and installation costs

Water Heat Type	Gas	Gas Tankless	Gas Condensing	Electric	HPWH
Cost	Basic Installation	Basic Installation	Basic Installation	Basic Installation	Basic Installation
Amount	$ 487.16	$ 1294.45	$ 487.16	$ 364.11	$ 364.11
Cost			Space Constraints		Additional Labor
Amount			$ 257.78		$ 63.50
Cost					Louvered Door
Amount					$ 383.52
Cost			Venting Costs		Drain Pan Increase
Amount			$ 287.95		$ 5.68

Table 33: Itemized installation costs for retrofit homes

Water Heat Type	Gas	Gas Tankless	Gas Condensing	Electric	HPWH
Cost	Basic Installation	Basic Installation	Basic Installation	Basic Installation	Basic Installation
Amount	$ 428.48	$ 773.16	$ 428.48	$ 218.47	$ 218.47
Cost					Additional Labor
Amount					$ 63.50
Cost			Drain Pan Increase		Drain Pan Increase
Amount			$ 0.75		$ 5.68
Cost	Venting (Steel, shared with furnace)		Venting Costs (PVC)		
Amount	$ 416.11		$ 206.65		
Cost	Venting Connector				
Amount	$ 89.60				

Table 34: Itemized installation costs for new construction homes

There are several interesting things to note in these tables. For gas water heaters, there are several potential venting options as shown in Figure 84. For this study, it is assumed that homes with a gas water heater also have a furnace and configuration a is likely to be the predominant venting configuration. Configurations c and d would only work with a condensing furnace, while b would work for a power vent or condensing water heater and a regular furnace. In configuration a, the venting is shared between both the furnace and the water heater. In this case, the venting cost is assumed to be half of the total cost of installing the vent as it is necessary for both appliances. In addition, there is a cost associated with connecting the water heater to the common vent.

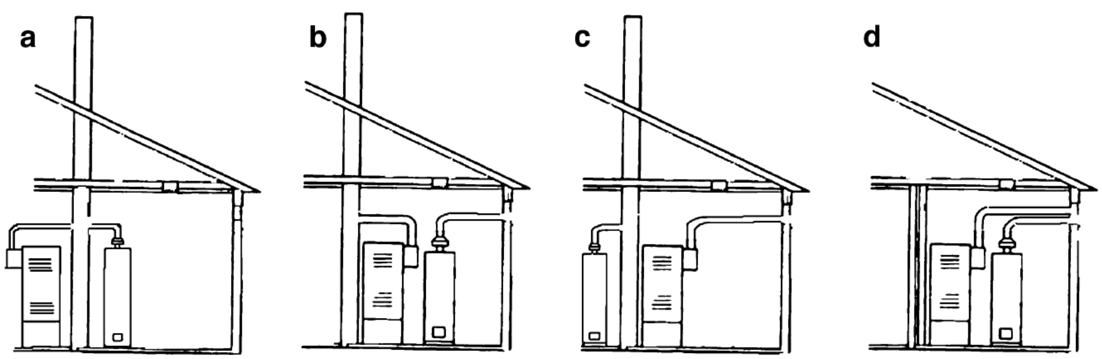


Figure 84: Potential venting configurations for homes with a gas water heater and a furnace (15)

In the case of retrofit homes, the installed cost of both tankless and condensing water heaters is significantly higher than a traditional gas storage water heater. In the case of tankless water heater, this is due to the need to modify the existing vents, which is rolled into the basic installation cost. Additionally, for some cases a larger gas line may need to be installed to accommodate the larger burner size of the tankless water heater. For the condensing water there are space constraints due to the larger size of the unit as well as additional venting costs.

However, in new construction cases both of these units have lower installation costs than the gas storage water heater. For the condensing water heater, this is due to having PVC vents instead of steel, which is significantly cheaper. Additional, the water heater may vent out of the side of the house as shown in configurations b and d in Figure 84. For the tankless water heater in new construction homes, the installation costs and venting costs combined are less than for a gas storage unit.

For HPWHs, there is an additional labor cost associated with the added complexity of this unit. It is assumed that in general it will take more time to set up these units than an electric storage water heater. In addition, in conditioned space there is an additional cost for installing a louvered door. It is assumed in this case that the HPWH will be installed in a utility closet and require the louvered door to provide sufficient airflow to the unit.

While these costs are assumed to be correct for the majority of installations, they will not be correct in every single application. Some installations may cost more (for example, a condensate pump may be required for condensing water heaters or HPWHs) and others may be cheaper (for example, using the existing venting or gas line for a tankless water heater). This makes breakeven cost an important metric, as the calculated breakeven cost presented here is the installed cost for the more efficient water heater and includes no assumptions about the installation cost of the upgrade.

The federal rulemaking does not cover solar water heaters. To determine the cost for these units, the costs from a survey of several water heater installers was used. This survey was performed at NREL by the residential buildings group and consisted of several people getting in touch with solar water heater installers to gather data on the total installed cost of solar water heaters. This survey gives the installed system cost for glazed flat plate solar water heaters (as well as several other types of solar water heaters) and the installed system size (in ft^2). The installed cost used here for both new and retrofit cases is based on the average installed cost of 46 different systems with sizes ranging from 60-66 ft^2 of collector area. Since the majority of solar water heaters are single tank systems with electric backup, the costs for a solar water heater with gas backup are the sum of the solar water heater installed cost and the gas water heater installed cost. This takes into account the extra cost associated with installing the second tank. Since solar water heater installation costs were found to be highly variable, the LCC cost analysis is also performed for cases where the installed cost is equal to the minimum cost ($3000).

Another key factor in capital costs are incentives. As was previously mentioned, cases with no incentives, federal incentives, local incentives, and federal plus local incentives are considered here. Current federal incentives for 2011 are provided in Table 35, while local incentives are provided in Table 36. For local incentives, the provider is also listed to show where the incentive can be applied. Local incentives were taken from the DSIRE database (66) which gives all local incentives, even for programs that may currently be closed due to lack of funding. Note that Atlanta has the largest incentives, but most of them are only available for homes in the city proper (not the suburbs).

Technology	Federal Incentive
Gas	$ 0
Tankless	$ 300
Condensing	$ 300
Solar Gas	30% of cost
Electric	$ 0
HPWH	$ 300
Solar Electric	30% of cost

Table 35: Federal incentives for different water heating technologies

Location	Water Heater	Local Incentive Provider	Value
Atlanta	HPWH	City of Atlanta	$ 1,000.00
	Tankless	City of Atlanta	$ 500
	Condensing	City of Atlanta	$ 200
	Solar	City of Atlanta	$ 1500
	Solar	State	35% up to 2500
	HPWH	Georgia Power	$ 250
	Solar	Georgia Power	$ 250
Chicago	Solar	DCEO	30%
Los Angeles	Solar Gas	State	$ 1875
	Solar Electric	State	$ 1089
Phoenix	Solar	State	25% up to $ 1000
	Solar	APS	$0.50/kWh of first year savings up to 50%
Seattle	HPWH	Seattle Power & Light	$ 250 (unconditioned space only)

Table 36: Local incentives for different water heating technologies

8.2.2 Water Heater Maintenance Costs

For several of these technologies, annual maintenance is required to keep them performing optimally. It is especially important to consider maintenance here as no annual performance degradation is considered in this analysis, meaning that maintenance is assumed to keep the equipment operating like new. The 2010 federal rulemaking also includes maintenance costs as well as the likelihood that a homeowner will actually perform the necessary maintenance. For this study maintenance costs were only applied to a technology that had a likelihood of higher than 10% that the maintenance would be performed. In the case of solar water heaters, which are not covered by the rulemaking, the system was assumed to need annual maintenance with a cost equal to 1% of the retrofit installed cost. These maintenance costs assume that the maintenance is done by a professional and not the homeowner. In the case of homeowner maintenance, the costs may be significantly less. However, for many of these technologies such as HPWHs and solar water heaters, the homeowner will likely be unable to perform maintenance on these units themselves. Maintenance costs are provided in Table 37.

Water Heat Type	Maintenance	Cost	Period	% likelihood
Gas	Annual Flush	$ 124.57	1	*10%*
Gas Tankless	De-liming	$ 71.71	1	56%
Condensing	Annual Flush	$ 124.57	1	*10%*
Electric	Annual Flush	$ 123.05	1	*10%*
HPWH	Total Maintenance	$ 94.50	5	*27%*
Solar Gas	Total Maintenance	$ 75.47	1	-
Solar Electric	Total Maintenance	$ 66.90	1	-

Table 37: Maintenance costs for each type of water heater. Highlighted rows denote maintenance that is not considered here.

8.2.3 Annual Operating Costs

Annual operating costs were determined using the monthly energy consumption to take into account seasonal variations in energy consumption. All water heaters have some seasonal variation as use changes with mains water temperature, but several technologies, including solar and HPWHs, are even more sensitive to seasonal changes in rates, with higher savings during the summer (when both gas and electricity rates peak). Energy consumption was calculated using Equation 18 and takes into account changes in space conditioning energy consumption and normalization energy as well as the water heater energy consumption. Monthly rates for both gas and electricity at each location are based on the rates for the largest utility at each location and are provided in Table 38 and Table 39. Annual operating costs, broken into cost for water heater energy consumption, normalization, and change in space conditioning equipment energy consumption is given in Appendix I.

	Atlanta	Chicago	Houston	Los Angeles	Phoenix	Seattle
January	$ 1.26	$ 0.84	$ 1.02	$ 1.11	$ 1.61	$ 1.20
February	$ 1.35	$ 0.90	$ 1.18	$ 1.15	$ 1.63	$ 1.20
March	$ 1.47	$ 1.01	$ 1.19	$ 1.16	$ 1.72	$ 1.22
April	$ 1.67	$ 1.12	$ 1.62	$ 1.31	$ 1.85	$ 1.20
May	$ 2.06	$ 1.35	$ 1.70	$ 1.45	$ 2.07	$ 1.27
June	$ 2.41	$ 1.60	$ 2.09	$ 1.52	$ 2.16	$ 1.34
July	$ 2.47	$ 1.92	$ 2.23	$ 1.66	$ 2.40	$ 1.47
August	$ 2.54	$ 1.90	$ 1.85	$ 1.48	$ 2.48	$ 1.53
September	$ 2.43	$ 1.62	$ 1.87	$ 1.28	$ 2.41	$ 1.44
October	$ 1.75	$ 1.26	$ 1.51	$ 1.17	$ 2.21	$ 1.32
November	$ 1.39	$ 1.11	$ 1.15	$ 0.95	$ 1.94	$ 1.40
December	$ 1.39	$ 0.92	$ 1.09	$ 0.93	$ 1.70	$ 1.39
Average	$ 1.85	$1.30	$1.54	$1.26	$ 2.02	$ 1.33

Table 38: Monthly gas rates in $/therm

	Atlanta	Chicago	Houston	Los Angeles	Phoenix	Seattle
January	8.85 ¢	9.99 ¢	12.00 ¢	14.95 ¢	9.38 ¢	9.32 ¢
February	9.06 ¢	10.33 ¢	11.77 ¢	14.53 ¢	9.56 ¢	9.42 ¢
March	9.52 ¢	10.91 ¢	12.37 ¢	14.30 ¢	9.95 ¢	9.37 ¢
April	9.67 ¢	11.51 ¢	12.82 ¢	14.04 ¢	10.62 ¢	9.36 ¢
May	9.95 ¢	12.12 ¢	13.24 ¢	14.70 ¢	11.69 ¢	8.92 ¢
June	11.22 ¢	12.10 ¢	13.77 ¢	15.68 ¢	11.73 ¢	9.60 ¢
July	11.41 ¢	11.93 ¢	14.45 ¢	15.60 ¢	11.61 ¢	9.68 ¢
August	11.37 ¢	11.75 ¢	14.21 ¢	15.75 ¢	11.52 ¢	9.70 ¢
September	10.89 ¢	12.16 ¢	13.87 ¢	15.19 ¢	11.27 ¢	9.76 ¢
October	10.53 ¢	12.58 ¢	13.91 ¢	14.20 ¢	11.09 ¢	9.70 ¢
November	9.82 ¢	12.79 ¢	13.68 ¢	15.37 ¢	10.12 ¢	9.67 ¢
December	9.47 ¢	11.30 ¢	13.52 ¢	14.93 ¢	10.18 ¢	9.41 ¢
Average	10.15 ¢	11.62 ¢	13.30 ¢	14.94 ¢	10.73 ¢	9.49 ¢

Table 39: Monthly electricity rates in ¢/kWh

8.2.4 Life Cycle Costs

The life cycle cost (LCC) analysis performed here was done in accordance with Federal Energy Management Program (FEMP) guidelines for LCC calculations (67). The simplified LCC cost equation, "for computing the LCC of energy and water conservation projects in buildings" (67) is given below.

$$LCC = I + Repl - Res + E + W + OM\&R \tag{20}$$

In this equation, *I* is the investment costs, *Repl* is the replacement costs, *Res* is the residual value (scrap, resale, or salvage), *E* is the energy costs, *W* is the water costs and *OM&R* is the present value non fuel operating, maintenance, and repair costs. All costs are present value and therefore include discount rates for costs in the future to take into account the time value of money. A 3% discount rate, in line with current federal guidelines (68), was applied. Fuel escalation costs were assumed to equal inflation in this case. Water costs were assumed to be the same in all cases (when comparing at the same draw profile) and excluded from the LCC calculations. Even though the volume of hot water drawn may change, the mixed draw volume is constant, leading to constant water use for each draw profile.

To determine the present value of annually recurring costs (fuel consumption and maintenance for most water heaters), the uniform present value (UPV) factor is applied to these costs. The uniform present value can be defined as:

$$UPV = \frac{(1+d)^y + 1}{d(1+d)^y} \tag{21}$$

In the above equation, d is the discount rate (3%) and y is the length of the study period. For nonrecurring costs (maintenance costs for a HPWH and residual value of water heaters with a lifetime over 30 years) the present value is calculated using the single present value (SPV), which is defined as:

$$SPV = \frac{1}{(1+d)^t} \tag{22}$$

For most cases, the water heater life is 13 years (hence why it was chosen as the analysis period), so there is no residual value. However, tankless water heaters and solar waters are assumed to last for longer than 13 years (20 and 30 years respectively) and therefore do have a residual value due to their remaining lifetime. In these cases, the residual value was calculated as:

$$Res = \frac{(N-y)}{N} I \cdot SPV \tag{23}$$

In the above equation, N is the expected lifetime of the system. This essentially linearly devalues the equipment based on its remaining lifetime. Note for equipment with a 13 year lifetime this gives a residual value of 0. When calculating the residual value, the investment cost is always the cost after incentives.

While only the lowest LCC option is presented in the following results, graphs of the

LCC of each option for every case are provided in Appendix J. For specific cases of interest, it is best to consult this appendix to see how close other options are to the lowest cost option, as in many cases there are several options with very similar life cycle costs.

For new construction homes with no incentives and gas water heating, a typical gas storage water heater was the most cost effective option in all cases as shown in Table 40. However, there are some cases in which a more efficient option was very close to being cost effective as shown in Appendix J. Condensing water heaters were closest to being effective in cool locations for high draw in unconditioned space due to reduced tank losses as well as the higher efficiency of the unit. Tankless offers higher savings at low use in unconditioned space due to very small tank losses, although as usage increases tank losses are a smaller portion of total energy use and condensing becomes more efficient. For Houston and Phoenix, the reduced space load of a tankless water heater allows these options to be very close to having a LCC savings. Due to the very high installed cost of solar, it is never close to being the most cost effective option without incentives.

Location	Conditioned Space			Unconditioned Space		
	Low	Medium	High	Low	Medium	High
Chicago	Gas	Gas	Gas	Gas	Gas	Gas
Seattle	Gas	Gas	Gas	Gas	Gas	Gas
Atlanta	Gas	Gas	Gas	Gas	Gas	Gas
Los Angeles	Gas	Gas	Gas	Gas	Gas	Gas
Houston	Gas	Gas	Gas	Gas	Gas	Gas
Phoenix	Gas	Gas	Gas	Gas	Gas	Gas

Table 40: Lowest LCC gas water heating option for new construction homes with no incentives

For new construction homes with no incentives and electric water heating, HPWHs often beat electric storage water heaters in terms of LCC. They generally do better with higher use due to higher potential savings, but in some cases work out even at low use depending on local electricity rates. In Chicago, there is a significant performance boost going from unconditioned space to conditioned space due to a lack of icing which makes them cost effective in low use cases when in conditioned space only. Once again, the high cost of solar water heaters prevents them from being the most cost effective option.

Location	Conditioned Space			Unconditioned Space		
	Low	Medium	High	Low	Medium	High
Chicago	HPWH	HPWH	HPWH	Electric	HPWH	HPWH
Seattle	Electric	HPWH	HPWH	Electric	HPWH	HPWH
Atlanta	Electric	HPWH	HPWH	Electric	HPWH	HPWH
Los Angeles	HPWH	HPWH	HPWH	HPWH	HPWH	HPWH
Houston	HPWH	HPWH	HPWH	HPWH	HPWH	HPWH
Phoenix	Electric	HPWH	HPWH	Electric	Electric	HPWH

Table 41: Lowest LCC electric water heating option for new construction homes with no incentives

When federal incentives are taken into account, more efficient options become much more attractive for homes using gas as shown in Table 42. Condensing water heaters become attractive at high use in unconditioned space in cold climates and at medium and high use in conditioned space in Atlanta, which has very high gas rates. Tankless water heaters also become cost effective in several locations, in particular in conditioned space in hot locations due to their lower impact on the building's cooling load. In addition, although solar never becomes the lowest cost option, it comes very close to regular gas storage water heaters in some high use cases, particularly in Atlanta and Phoenix.

Location	Conditioned Space			Unconditioned Space		
	Low	Medium	High	Low	Medium	High
Chicago	Gas	Gas	Gas	Gas	Gas	Condensing
Seattle	Gas	Gas	Gas	Gas	Gas	Condensing
Atlanta	Gas	Condensing	Condensing	Tankless	Tankless	Condensing
Los Angeles	Gas	Gas	Gas	Gas	Gas	Gas
Houston	Tankless	Tankless	Gas	Gas	Gas	Gas
Phoenix	Tankless	Tankless	Tankless	Gas	Gas	Gas

Table 42: Lowest LCC gas water heating option for new construction homes with federal incentives

For electric water heaters with federal incentives, HPWHs become attractive in almost every situation as shown in Table 43. There are a few low use situations where the HPWH does not manage to be cost effective, although in the most extreme case the HPWH has a LCC less than 10% higher than an electric storage water heater.

Location	Conditioned Space			Unconditioned Space		
	Low	Medium	High	Low	Medium	High
Chicago	HPWH	HPWH	HPWH	HPWH	HPWH	HPWH
Seattle	Electric	HPWH	HPWH	HPWH	HPWH	HPWH
Atlanta	HPWH	HPWH	HPWH	HPWH	HPWH	HPWH
Los Angeles	HPWH	HPWH	HPWH	HPWH	HPWH	HPWH
Houston	HPWH	HPWH	HPWH	HPWH	HPWH	HPWH
Phoenix	HPWH	HPWH	HPWH	Electric	HPWH	HPWH

Table 43: Lowest LCC electric water heating option for new construction homes with federal incentives

When just local incentives are considered, solar water heaters become much more attractive in locations with significant solar water heating incentives although it does not become the lowest cost option. However, incentives for tankless and condensing water heaters in Atlanta lead to these options becoming the most cost effective option as shown in Table 44. Solar water heating incentives are also large in Atlanta, leading to solar water heaters with gas backup having a lower LCC than regular gas storage water heaters in some cases as shown in Appendix J. Phoenix, Chicago and Los Angeles also offer incentives, but the value of these incentives is not large enough to make a solar water heater the most cost effective option. For electric water heaters, HPWHs continue to be the most cost effective option in the majority of cases as shown in Table 45.

Location	Conditioned Space			Unconditioned Space		
	Low	Medium	High	Low	Medium	High
Chicago	Gas	Gas	Gas	Gas	Gas	Gas
Seattle	Gas	Gas	Gas	Gas	Gas	Gas
Atlanta	Tankless	Tankless	Condensing	Tankless	Tankless	Tankless
Los Angeles	Gas	Gas	Gas	Gas	Gas	Gas
Houston	Gas	Gas	Gas	Gas	Gas	Gas
Phoenix	Gas	Gas	Gas	Gas	Gas	Gas

Table 44: Lowest LCC gas water heating option for new construction homes with local incentives

Location	Conditioned Space			Unconditioned Space		
	Low	Medium	High	Low	Medium	High
Chicago	HPWH	HPWH	HPWH	Electric	HPWH	HPWH
Seattle	Electric	HPWH	HPWH	HPWH	HPWH	HPWH
Atlanta	HPWH	HPWH	HPWH	HPWH	HPWH	HPWH
Los Angeles	HPWH	HPWH	HPWH	HPWH	HPWH	HPWH
Houston	HPWH	HPWH	HPWH	HPWH	HPWH	HPWH
Phoenix	Electric	HPWH	HPWH	Electric	Electric	HPWH

Table 45: Lowest LCC electric water heating option for new construction homes with local incentives

When both federal and local incentives are taken into account, solar becomes an attractive option in Atlanta and Phoenix for homes with gas water heaters as shown in Table 46. However, a gas storage water heater continues to be the most cost effective option in many cases. For electric water heaters, the results are almost identical to the case of just federal incentives as the local incentives are not large enough to change what the lowest LCC option is as shown in Table 47. The only exception is that a HPWH becomes cost effective for low use in Phoenix when the water heater is installed in unconditioned space.

	Conditioned Space			Unconditioned Space		
Location	**Low**	**Medium**	**High**	**Low**	**Medium**	**High**
Chicago	Gas	Gas	Gas	Gas	Gas	Condensing
Seattle	Gas	Gas	Gas	Gas	Gas	Condensing
Atlanta	Solar	Solar	Solar	Solar	Solar	Solar
Los Angeles	Gas	Gas	Gas	Gas	Gas	Gas
Houston	Tankless	Tankless	Gas	Gas	Gas	Gas
Phoenix	Tankless	Solar	Solar	Solar	Solar	Solar

Table 46: Lowest LCC gas water heating option for new construction homes with federal and local incentives

	Conditioned Space			Unconditioned Space		
Location	**Low**	**Medium**	**High**	**Low**	**Medium**	**High**
Chicago	HPWH	HPWH	HPWH	HPWH	HPWH	HPWH
Seattle	Electric	HPWH	HPWH	HPWH	HPWH	HPWH
Atlanta	HPWH	HPWH	HPWH	HPWH	HPWH	HPWH
Los Angeles	HPWH	HPWH	HPWH	HPWH	HPWH	HPWH
Houston	HPWH	HPWH	HPWH	HPWH	HPWH	HPWH
Phoenix	HPWH	HPWH	HPWH	HPWH	HPWH	HPWH

Table 47: Lowest LCC electric water heating option for new construction homes with federal and local incentives

When looking at retrofit situations, there are some differences from new construction cases. For gas water heaters, gas storage is again always the most cost effective option as shown in Table 48. There are a few reasons for this. For one, gas water heaters are cheaper to install as a retrofit option than a new construction option as much of the new construction cost of gas storage water heaters comes from installing the venting for the unit. Condensing water heaters become less attractive as much of their savings came from using PVC venting instead of metal (PVC can be used for condensing water heaters since they vent at a lower temperature). Tankless water heaters become more expensive in retrofit situations due to their higher burn rate, which can require larger venting.

	Conditioned Space			Unconditioned Space		
Location	**Low**	**Medium**	**High**	**Low**	**Medium**	**High**
Chicago	Gas	Gas	Gas	Gas	Gas	Gas
Seattle	Gas	Gas	Gas	Gas	Gas	Gas
Atlanta	Gas	Gas	Gas	Gas	Gas	Gas
Los Angeles	Gas	Gas	Gas	Gas	Gas	Gas
Houston	Gas	Gas	Gas	Gas	Gas	Gas
Phoenix	Gas	Gas	Gas	Gas	Gas	Gas

Table 48: Lowest LCC gas water heating option for retrofit homes with no incentives

HPWHs also become less attractive options in retrofit scenarios as shown in Table 50. This is especially true in conditioned space, which has an additional installation cost associated with it. This extra cost is for a louvered door, which is required to ensure that there is sufficient airflow to the HPWH. This makes HPWHs less attractive in some cases, although they still remain a more cost effective option in the majority of cases considered here.

	Conditioned Space			Unconditioned Space		
Location	**Low**	**Medium**	**High**	**Low**	**Medium**	**High**
Chicago	Electric	HPWH	HPWH	Electric	HPWH	HPWH
Seattle	Electric	Electric	Electric	Electric	HPWH	HPWH
Atlanta	Electric	HPWH	HPWH	Electric	HPWH	HPWH
Los Angeles	Electric	HPWH	HPWH	HPWH	HPWH	HPWH
Houston	HPWH	HPWH	HPWH	HPWH	HPWH	HPWH
Phoenix	Electric	Electric	HPWH	Electric	Electric	HPWH

Table 49: Lowest LCC electric water heating option for retrofit homes with no incentives

The addition of federal incentives does little to change the cost effectiveness of different gas technologies. Traditional gas storage units remain the most cost effective option as shown in Table 50. The picture does change somewhat for electric water heaters as shown in Table 51. HPWHs become more cost effective in several situations, including low use in unconditioned space in several locations.

	Conditioned Space			Unconditioned Space		
Location	**Low**	**Medium**	**High**	**Low**	**Medium**	**High**
Chicago	Gas	Gas	Gas	Gas	Gas	Gas
Seattle	Gas	Gas	Gas	Gas	Gas	Gas
Atlanta	Gas	Gas	Gas	Gas	Gas	Gas
Los Angeles	Gas	Gas	Gas	Gas	Gas	Gas
Houston	Gas	Gas	Gas	Gas	Gas	Gas
Phoenix	Gas	Gas	Gas	Gas	Gas	Gas

Table 50: Lowest LCC gas water heating option for retrofit homes with federal incentives

	Conditioned Space			Unconditioned Space		
Location	**Low**	**Medium**	**High**	**Low**	**Medium**	**High**
Chicago	Electric	HPWH	HPWH	HPWH	HPWH	HPWH
Seattle	Electric	Electric	HPWH	HPWH	HPWH	HPWH
Atlanta	Electric	HPWH	HPWH	HPWH	HPWH	HPWH
Los Angeles	HPWH	HPWH	HPWH	HPWH	HPWH	HPWH
Houston	HPWH	HPWH	HPWH	HPWH	HPWH	HPWH
Phoenix	Electric	HPWH	HPWH	Electric	HPWH	HPWH

Table 51: Lowest LCC electric water heating option for retrofit homes with federal incentives

In the case where only local incentives are considered, the results are similar to the new construction case. For gas water heaters, solar once again becomes cost effective in Atlanta (for all cases) and Phoenix (for high use cases) due to their high incentives as shown in Table 52. For electric water heaters, solar again does not becomes cost effective as HPWHs remain cost effective even though solar has large incentives due to the low installed cost of HPWHs relative to solar water heaters as shown in Table 53. HPWHs become cost effective at low use in unconditioned space only in Seattle due to the incentive there.

Location	Conditioned Space			Unconditioned Space		
	Low	Medium	High	Low	Medium	High
Chicago	Gas	Gas	Gas	Gas	Gas	Gas
Seattle	Gas	Gas	Gas	Gas	Gas	Gas
Atlanta	Solar	Solar	Solar	Solar	Solar	Solar
Los Angeles	Gas	Gas	Gas	Gas	Gas	Gas
Houston	Gas	Gas	Gas	Gas	Gas	Gas
Phoenix	Gas	Gas	Solar	Gas	Solar	Solar

Table 52: Lowest LCC gas water heating option for retrofit homes with local incentives

Location	Conditioned Space			Unconditioned Space		
	Low	Medium	High	Low	Medium	High
Chicago	Electric	HPWH	HPWH	Electric	HPWH	HPWH
Seattle	Electric	Electric	Electric	HPWH	HPWH	HPWH
Atlanta	HPWH	HPWH	HPWH	HPWH	HPWH	HPWH
Los Angeles	Electric	HPWH	HPWH	HPWH	HPWH	HPWH
Houston	HPWH	HPWH	HPWH	HPWH	HPWH	HPWH
Phoenix	Electric	Electric	HPWH	Electric	Electric	HPWH

Table 53: Lowest LCC electric water heating option for retrofit homes with local incentives

When both federal and local incentives are considered energy saving technologies begin to look more attractive. For gas water heaters, solar becomes cost effective in a few more cases in Phoenix than in the case with just local incentives as shown in Table 54. For electric water heaters, solar becomes cost effective in some situations in Atlanta, Los Angeles, and Phoenix as shown in Table 55 However, HPWHs remain the most cost effective option in the majority of cases.

Location	Conditioned Space			Unconditioned Space		
	Low	Medium	High	Low	Medium	High
Chicago	Gas	Gas	Gas	Gas	Gas	Gas
Seattle	Gas	Gas	Gas	Gas	Gas	Gas
Atlanta	Solar	Solar	Solar	Solar	Solar	Solar
Los Angeles	Gas	Gas	Gas	Gas	Gas	Gas
Houston	Gas	Gas	Gas	Gas	Gas	Gas
Phoenix	Gas	Solar	Solar	Solar	Solar	Solar

Table 54: Lowest LCC gas water heating option for retrofit homes with federal and local incentives

Location	Conditioned Space			Unconditioned Space		
	Low	Medium	High	Low	Medium	High
Chicago	Electric	HPWH	HPWH	HPWH	HPWH	HPWH
Seattle	Electric	Electric	HPWH	HPWH	HPWH	HPWH
Atlanta	Solar	HPWH	HPWH	HPWH	HPWH	HPWH
Los Angeles	Solar	HPWH	HPWH	HPWH	HPWH	HPWH
Houston	HPWH	HPWH	HPWH	HPWH	HPWH	HPWH
Phoenix	Electric	Solar	Solar	Electric	Solar	Solar

Table 55: Lowest LCC electric water heating option for retrofit homes with federal and local incentives

8.2.5 Sensitivity to Solar Costs

Given the wide variability in solar water heater installation costs, the LCC analysis was also performed assuming that the solar water heater installed cost is equal to the minimum installed cost of water heaters in this size range instead of the average cost. This leads to an installed cost of $3000 instead of $6690. For all cases, this lowers the LCC by $3690 for solar water heaters. New table showing the most cost effective option with this solar water heating price are given in Table 56-Table 71. Situations where solar water heaters are now the most cost effective option are highlighted.

For gas water heaters, there are relatively few cases where this low cost assumption leads to solar water heaters being cost effective without incentives. However, when combined with the federal incentive, the number of cases where solar water heaters become cost effective jumps drastically. For electric water heaters there are more cases where solar water heaters work without incentives, although the number of cases also jumps significantly when solar incentives are applied. Given that many of the cases considered here had significant local incentives that had already made solar water heaters cost effective, solar becomes an even cheaper option but there is no change in what is most effective.

Location	Conditioned Space			Unconditioned Space		
	Low	Medium	High	Low	Medium	High
Chicago	Gas	Gas	Gas	Gas	Gas	Gas
Seattle	Gas	Gas	Gas	Gas	Gas	Gas
Atlanta	Gas	Gas	Solar	Gas	Solar	Solar
Los Angeles	Gas	Gas	Gas	Gas	Gas	Gas
Houston	Gas	Gas	Gas	Gas	Gas	Gas
Phoenix	Gas	Gas	Gas	Gas	Gas	Solar

Table 56: Lowest LCC gas water heating option for new construction homes with no incentives for the low cost solar case

Location	Conditioned Space			Unconditioned Space		
	Low	Medium	High	Low	Medium	High
Chicago	HPWH	HPWH	HPWH	Solar	Solar	Solar
Seattle	Electric	HPWH	HPWH	Electric	HPWH	HPWH
Atlanta	Electric	HPWH	HPWH	Electric	HPWH	HPWH
Los Angeles	Solar	Solar	Solar	Solar	Solar	Solar
Houston	HPWH	HPWH	HPWH	HPWH	HPWH	HPWH
Phoenix	Electric	HPWH	HPWH	Electric	Electric	Solar

Table 57: Lowest LCC electric water heating option for new construction homes with no incentives for the low cost solar case

Location	Conditioned Space			Unconditioned Space		
	Low	Medium	High	Low	Medium	High
Chicago	Gas	Solar	Solar	Solar	Solar	Solar
Seattle	Gas	Solar	Solar	Solar	Solar	Solar
Atlanta	Solar	Solar	Solar	Solar	Solar	Solar
Los Angeles	Gas	Gas	Solar	Solar	Solar	Solar
Houston	Tankless	Tankless	Solar	Solar	Solar	Solar
Phoenix	Tankless	Solar	Solar	Solar	Solar	Solar

Table 58: Lowest LCC gas water heating option for new construction homes with federal incentives for the low cost solar case

Location	Conditioned Space			Unconditioned Space		
	Low	Medium	High	Low	Medium	High
Chicago	Solar	HPWH	HPWH	Solar	Solar	Solar
Seattle	Solar	HPWH	HPWH	Solar	HPWH	HPWH
Atlanta	Solar	Solar	HPWH	Solar	Solar	Solar
Los Angeles	Solar	Solar	Solar	Solar	Solar	Solar
Houston	HPWH	HPWH	HPWH	Solar	HPWH	HPWH
Phoenix	Solar	Solar	HPWH	Solar	Solar	Solar

Table 59: Lowest LCC electric water heating option for new construction homes with federal incentives for the low cost solar case

Location	Conditioned Space			Unconditioned Space		
	Low	Medium	High	Low	Medium	High
Chicago	Gas	Solar	Solar	Solar	Solar	Solar
Seattle	Gas	Gas	Gas	Gas	Gas	Gas
Atlanta	Solar	Solar	Solar	Solar	Solar	Solar
Los Angeles	Solar	Solar	Solar	Solar	Solar	Solar
Houston	Gas	Gas	Gas	Gas	Gas	Gas
Phoenix	Solar	Solar	Solar	Solar	Solar	Solar

Table 60: Lowest LCC gas water heating option for new construction homes with local incentives for the low cost solar case

Location	Conditioned Space			Unconditioned Space		
	Low	Medium	High	Low	Medium	High
Chicago	Solar	Solar	Solar	Solar	Solar	Solar
Seattle	Electric	HPWH	HPWH	HPWH	HPWH	HPWH
Atlanta	Solar	Solar	Solar	Solar	Solar	Solar
Los Angeles	Solar	Solar	Solar	Solar	Solar	Solar
Houston	HPWH	HPWH	HPWH	HPWH	HPWH	HPWH
Phoenix	Solar	Solar	Solar	Solar	Solar	Solar

Table 61: Lowest LCC electric water heating option for new construction homes with local incentives for the low cost solar case

Location	Conditioned Space			Unconditioned Space		
	Low	Medium	High	Low	Medium	High
Chicago	Solar	Solar	Solar	Solar	Solar	Solar
Seattle	Gas	Solar	Solar	Solar	Solar	Solar
Atlanta	Solar	Solar	Solar	Solar	Solar	Solar
Los Angeles	Solar	Solar	Solar	Solar	Solar	Solar
Houston	Tankless	Solar	Solar	Solar	Solar	Solar
Phoenix	Solar	Solar	Solar	Solar	Solar	Solar

Table 62: Lowest LCC gas water heating option for new construction homes with local and federal incentives for the low cost solar case

Location	Conditioned Space			Unconditioned Space		
	Low	Medium	High	Low	Medium	High
Chicago	Solar	Solar	Solar	Solar	Solar	Solar
Seattle	Solar	HPWH	HPWH	Solar	HPWH	HPWH
Atlanta	Solar	Solar	Solar	Solar	Solar	Solar
Los Angeles	Solar	Solar	Solar	Solar	Solar	Solar
Houston	Solar	Solar	HPWH	Solar	Solar	Solar
Phoenix	Solar	Solar	Solar	Solar	Solar	Solar

Table 63: Lowest LCC electric water heating option for new construction homes with local and federal incentives for the low cost solar case

Location	Conditioned Space			Unconditioned Space		
	Low	Medium	High	Low	Medium	High
Chicago	Gas	Gas	Gas	Gas	Gas	Gas
Seattle	Gas	Gas	Gas	Gas	Gas	Gas
Atlanta	Gas	Gas	Gas	Gas	Gas	Solar
Los Angeles	Gas	Gas	Gas	Gas	Gas	Gas
Houston	Gas	Gas	Gas	Gas	Gas	Gas
Phoenix	Gas	Gas	Gas	Gas	Gas	Gas

Table 64: Lowest LCC gas water heating option for retrofit homes with no incentives for the low cost solar case

Location	Conditioned Space			Unconditioned Space		
	Low	Medium	High	Low	Medium	High
Chicago	Solar	Solar	Solar	Solar	Solar	Solar
Seattle	Electric	Electric	Electric	Electric	HPWH	HPWH
Atlanta	Solar	Solar	Solar	Solar	Solar	HPWH
Los Angeles	Solar	Solar	Solar	Solar	Solar	Solar
Houston	Solar	HPWH	HPWH	HPWH	HPWH	HPWH
Phoenix	Electric	Solar	Solar	Electric	Solar	Solar

Table 65: Lowest LCC electric water heating option for retrofit homes with no incentives for the low cost solar case

Location	Conditioned Space			Unconditioned Space		
	Low	Medium	High	Low	Medium	High
Chicago	Gas	Gas	Solar	Solar	Solar	Solar
Seattle	Gas	Gas	Gas	Solar	Solar	Solar
Atlanta	Solar	Solar	Solar	Solar	Solar	Solar
Los Angeles	Gas	Gas	Gas	Gas	Solar	Solar
Houston	Gas	Gas	Gas	Gas	Solar	Solar
Phoenix	Solar	Solar	Solar	Solar	Solar	Solar

Table 66: Lowest LCC gas water heating option for retrofit homes with federal incentives for the low cost solar case

Location	Conditioned Space			Unconditioned Space		
	Low	Medium	High	Low	Medium	High
Chicago	Solar	Solar	Solar	Solar	Solar	Solar
Seattle	Solar	Solar	Solar	Solar	HPWH	HPWH
Atlanta	Solar	Solar	Solar	Solar	Solar	Solar
Los Angeles	Solar	Solar	Solar	Solar	Solar	Solar
Houston	Solar	Solar	Solar	Solar	Solar	Solar
Phoenix	Solar	Solar	Solar	Solar	Solar	Solar

Table 67: Lowest LCC electric water heating option for retrofit homes with federal incentives for the low cost solar case

Location	Conditioned Space			Unconditioned Space		
	Low	Medium	High	Low	Medium	High
Chicago	Gas	Gas	Gas	Gas	Solar	Solar
Seattle	Gas	Gas	Gas	Gas	Gas	Gas
Atlanta	Solar	Solar	Solar	Solar	Solar	Solar
Los Angeles	Gas	Solar	Solar	Solar	Solar	Solar
Houston	Gas	Gas	Gas	Gas	Gas	Gas
Phoenix	Solar	Solar	Solar	Solar	Solar	Solar

Table 68: Lowest LCC gas water heating option for retrofit homes with local incentives for the low cost solar case

Location	Conditioned Space			Unconditioned Space		
	Low	Medium	High	Low	Medium	High
Chicago	Solar	Solar	Solar	Solar	Solar	Solar
Seattle	Electric	Electric	Electric	HPWH	HPWH	HPWH
Atlanta	Solar	Solar	Solar	Solar	Solar	Solar
Los Angeles	Solar	Solar	Solar	Solar	Solar	Solar
Houston	Solar	HPWH	HPWH	HPWH	HPWH	HPWH
Phoenix	Solar	Solar	Solar	Solar	Solar	Solar

Table 69: Lowest LCC electric water heating option for retrofit homes with local incentives for the low cost solar case

Location	Conditioned Space			Unconditioned Space		
	Low	Medium	High	Low	Medium	High
Chicago	Solar	Solar	Solar	Solar	Solar	Solar
Seattle	Gas	Gas	Gas	Solar	Solar	Solar
Atlanta	Solar	Solar	Solar	Solar	Solar	Solar
Los Angeles	Solar	Solar	Solar	Solar	Solar	Solar
Houston	Gas	Gas	Gas	Gas	Solar	Solar
Phoenix	Solar	Solar	Solar	Solar	Solar	Solar

Table 70: Lowest LCC gas water heating option for retrofit homes with federal and local incentives for the low cost solar case

Location	Conditioned Space			Unconditioned Space		
	Low	Medium	High	Low	Medium	High
Chicago	Solar	Solar	Solar	Solar	Solar	Solar
Seattle	Solar	Solar	Solar	HPWH	HPWH	HPWH
Atlanta	Solar	Solar	Solar	Solar	Solar	HPWH
Los Angeles	Solar	Solar	Solar	Solar	Solar	Solar
Houston	Solar	Solar	Solar	Solar	Solar	Solar
Phoenix	Solar	Solar	Solar	Solar	Solar	Solar

Table 71: Lowest LCC electric water heating option for retrofit homes with federal and local incentives for the low cost solar case

8.2.6 Sensitivity of Life Cycle Costs to Utility Rates

A sensitivity study was performed to determine the sensitivity of the water heater life cycle cost to the assumed utility rates. While the utility rates used here are reasonable for these locations at this time, utility rates are volatile and may change with time. To determine how changes in utility rates would change which water heating option would be most cost effective plots were created of the water heater LCC at different utility rates. In the sensitivity study, only medium use for water heaters in conditioned space in the new construction scenario with no incentives was considered.

For gas water heaters, the life cycle cost depends on both the gas rate and the electricity rate. Tankless, condensing and solar water heaters all consume both electricity and gas. The change in space conditioning energy consumption also includes some electricity use. In this case, gas rates between 0 and 10 $/therm and electricity rates between 0 and 30 ¢/kWh were considered. The lowest LCC option for each climate is shown in Figure 85-Figure 90.

For all locations, a gas water heater is the most cost effective option in the case of free energy and a solar water heater is the best option at the highest utility rates. In between these extremes, either a condensing water heater or a tankless water heater is the most cost effective option depending on the location. In cold locations where the tank losses help to reduce the space heating load condensing water heaters are more cost effective, while in hot locations a tankless water heater is more cost effective. Depending primarily on the water heater's interaction with the space conditioning energy consumption, the cost effectiveness of different water heaters at different electricity rates can change. This is especially apparent in the case of Phoenix, where the solar water heater becomes cost effective at a rate of about 6 $/therm if electricity is free and 8 $/therm if electricity costs 30 ¢/kWh due to both the solar water heater's electricity consumption and the additional space cooling energy consumption required due to the higher losses from the solar water heating system to the space.

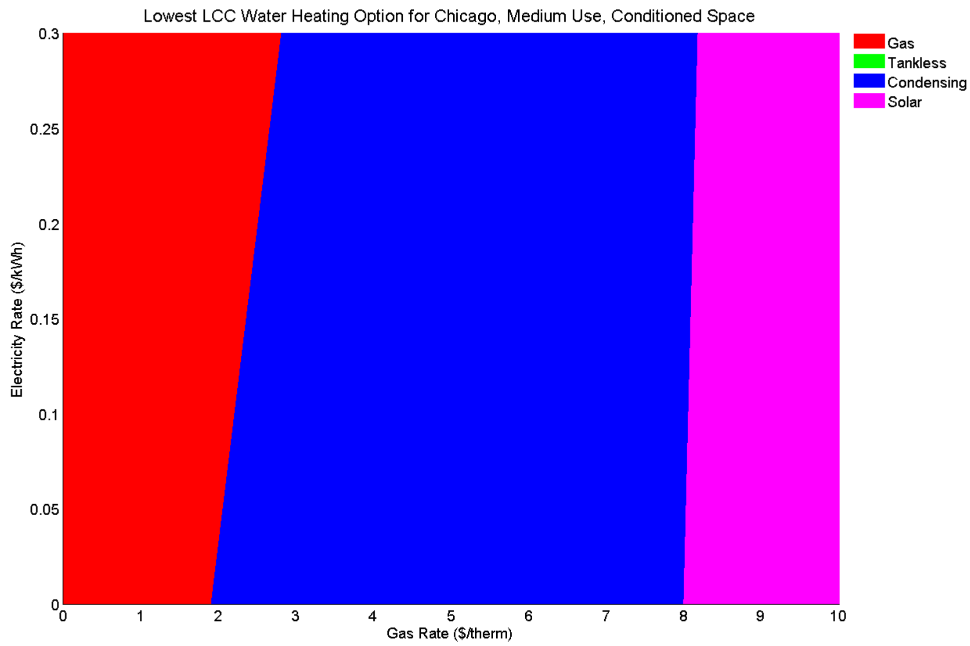


Figure 85: Lowest LCC gas water heating option for Chicago at different rates

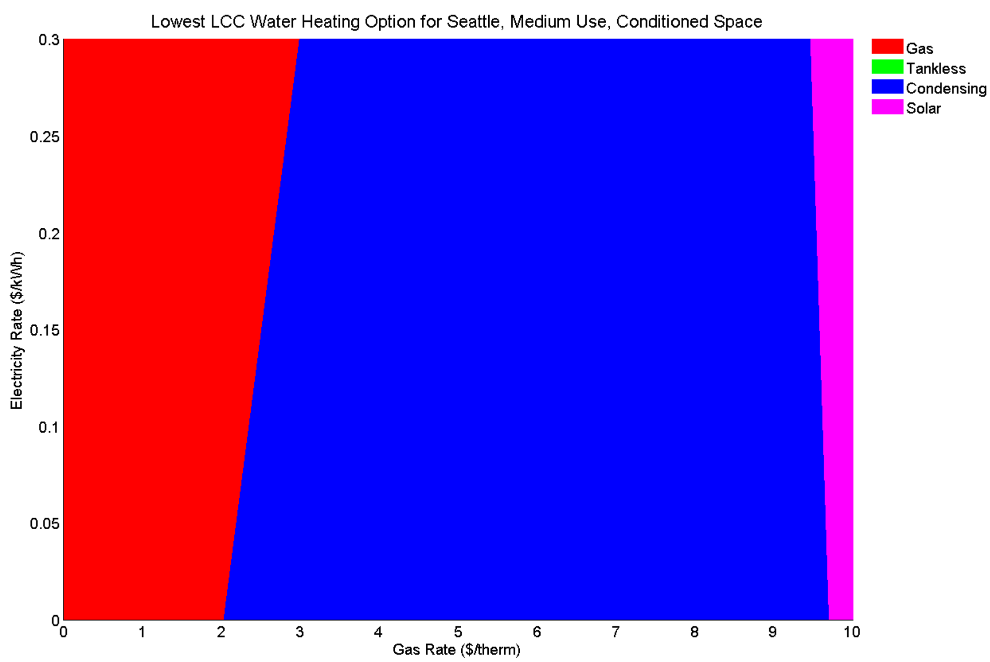


Figure 86: Lowest LCC gas water heating option for Seattle at different rates

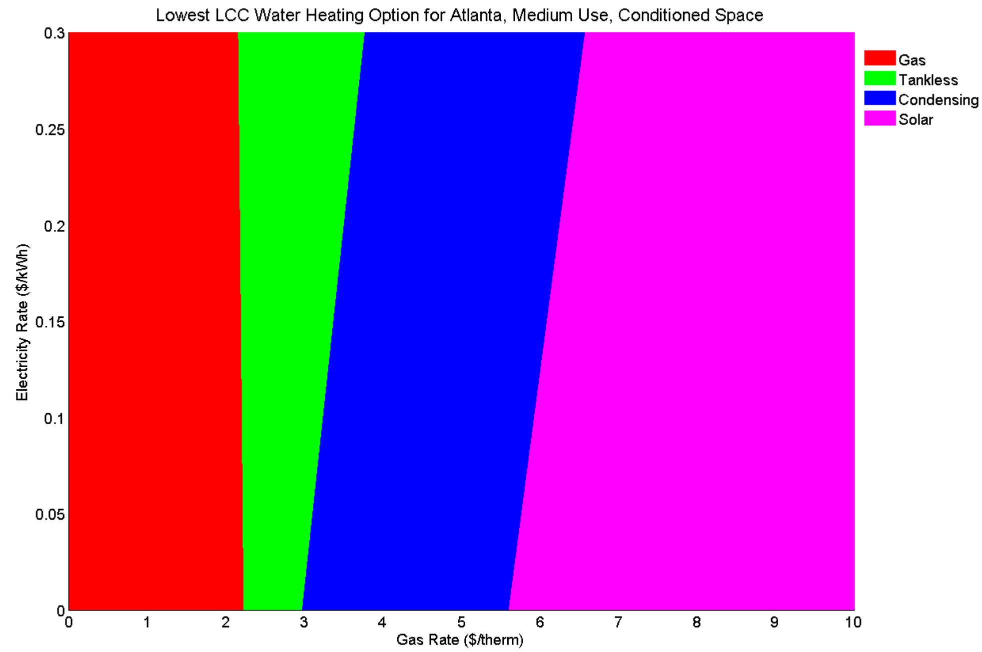


Figure 87: Lowest LCC gas water heating option for Atlanta at different rates

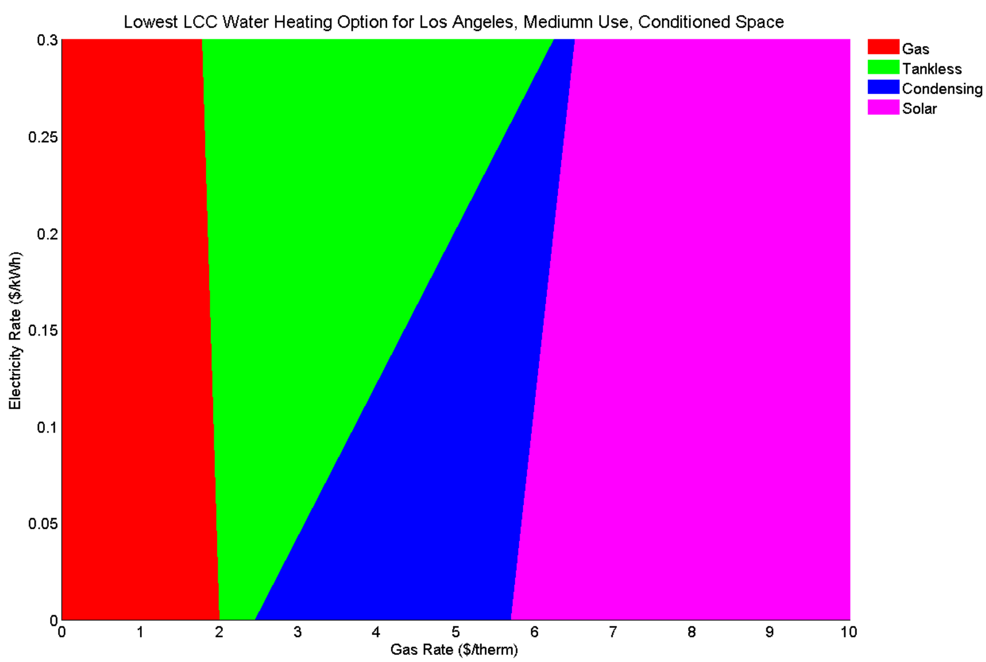


Figure 88: Lowest LCC gas water heating option for Los Angeles at different rates

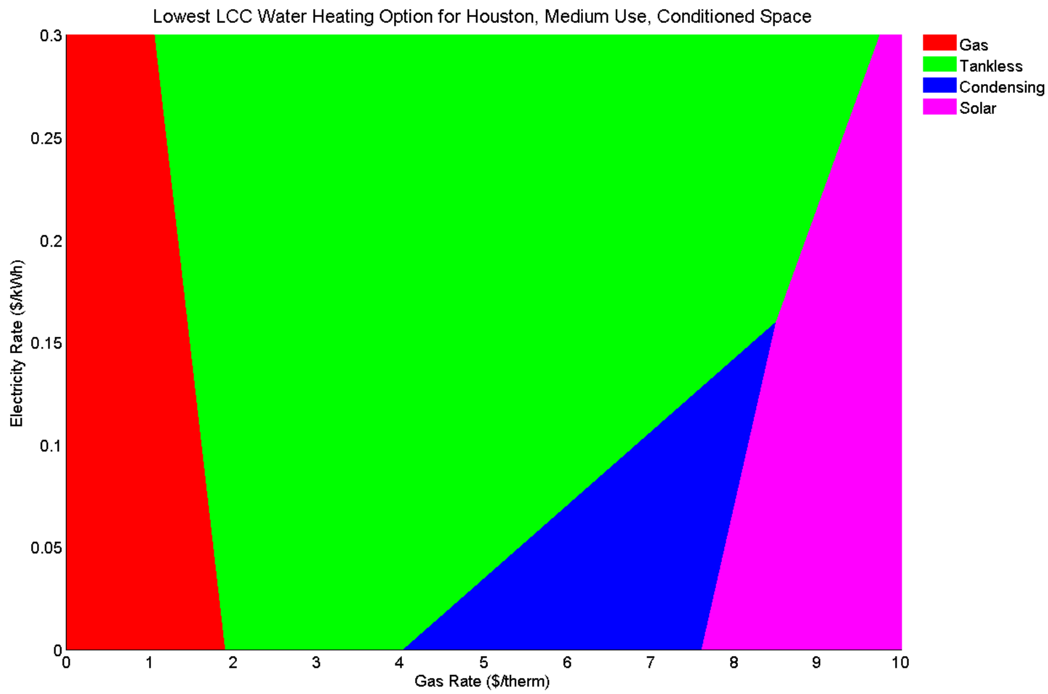


Figure 89: Lowest LCC gas water heating option for Houston at different rates

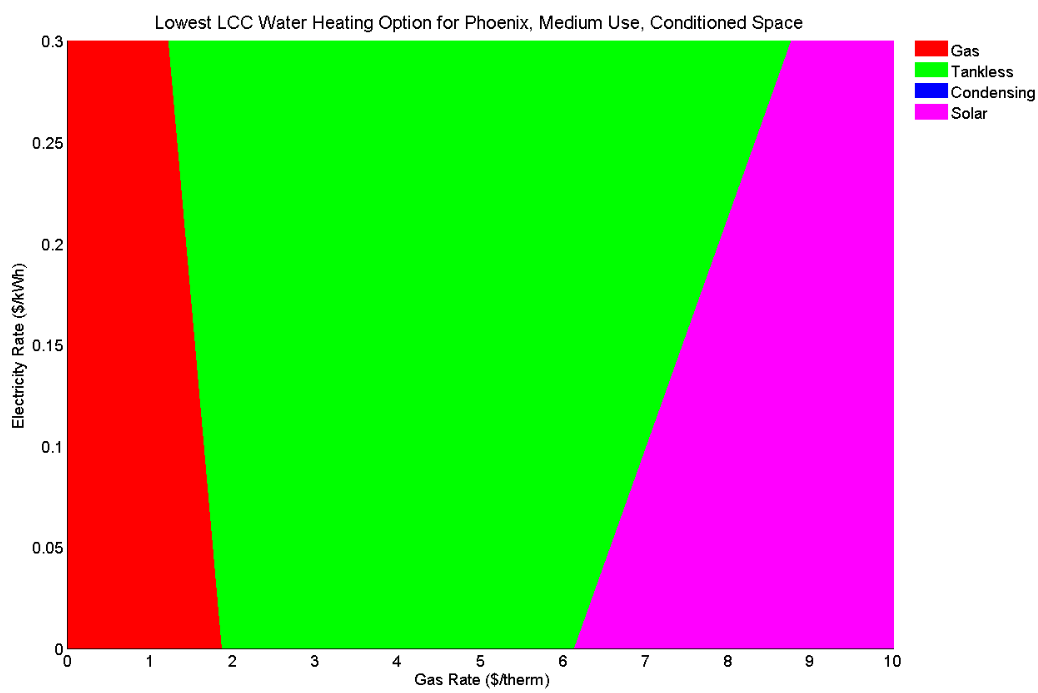


Figure 90: Lowest LCC gas water heating option for Phoenix at different rates

For electric water heaters, the LCC depends only on the electricity rate since it is assumed these water heaters will only be installed in homes which do not have gas. The LCC of each electric water heating option in all climates is shown in Figure 91-Figure 96. In this case, a regular electric storage water heater is the lowest cost option for the case of free electricity and a HPWH is the most cost effective option at the high cost case of 30 ¢/kWh. Since the HPWH and the solar water heater save comparable amounts of energy and the solar water heater has a higher installed cost, the solar water heater is never the most cost effective option for the range of rates considered here. However, in some cases a solar water heater would become the most cost effective option if even higher electricity rates are considered. This is especially apparent in the case of Los Angeles where the solar water heater and HPWH LCCs are coming closer together as the electricity rates increase.

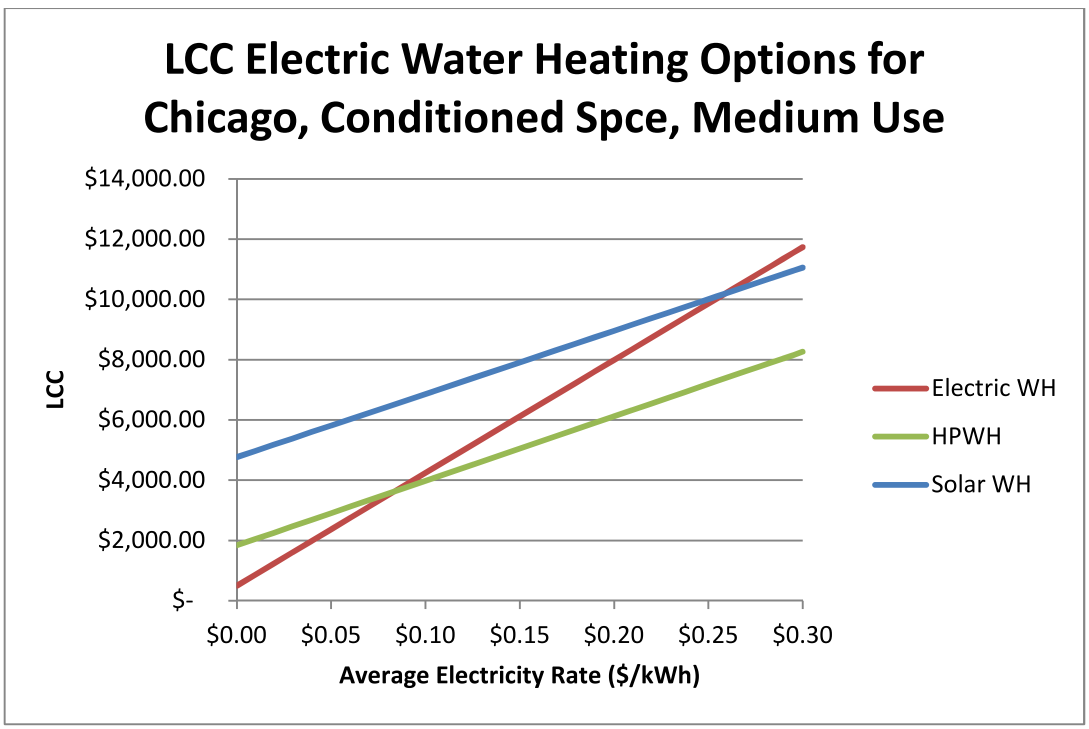


Figure 91: LCC of electric water heating options for Chicago at different rates

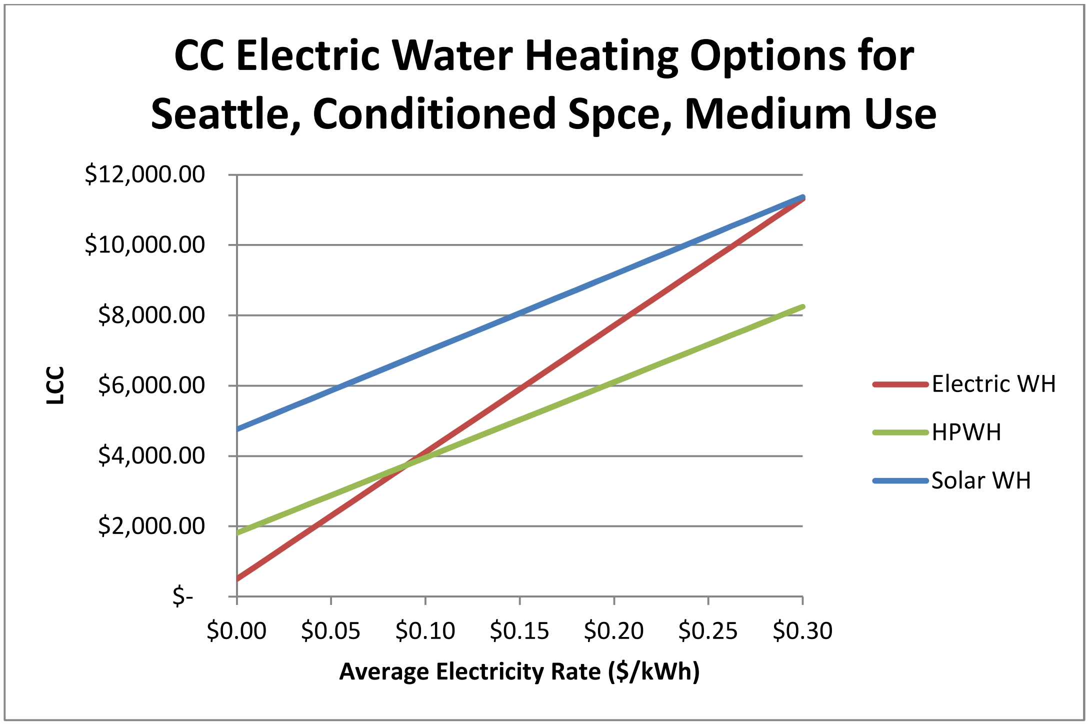


Figure 92: LCC of electric water heating options for Seattle at different rates

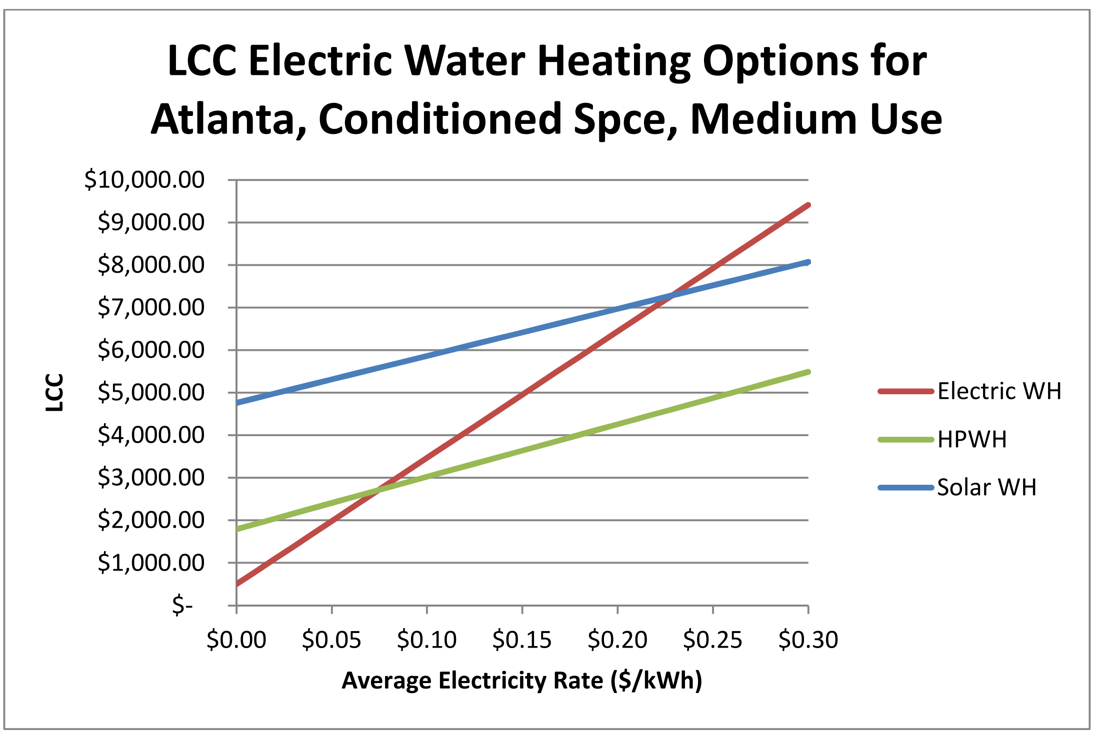


Figure 93: LCC of electric water heating options for Atlanta at different rates

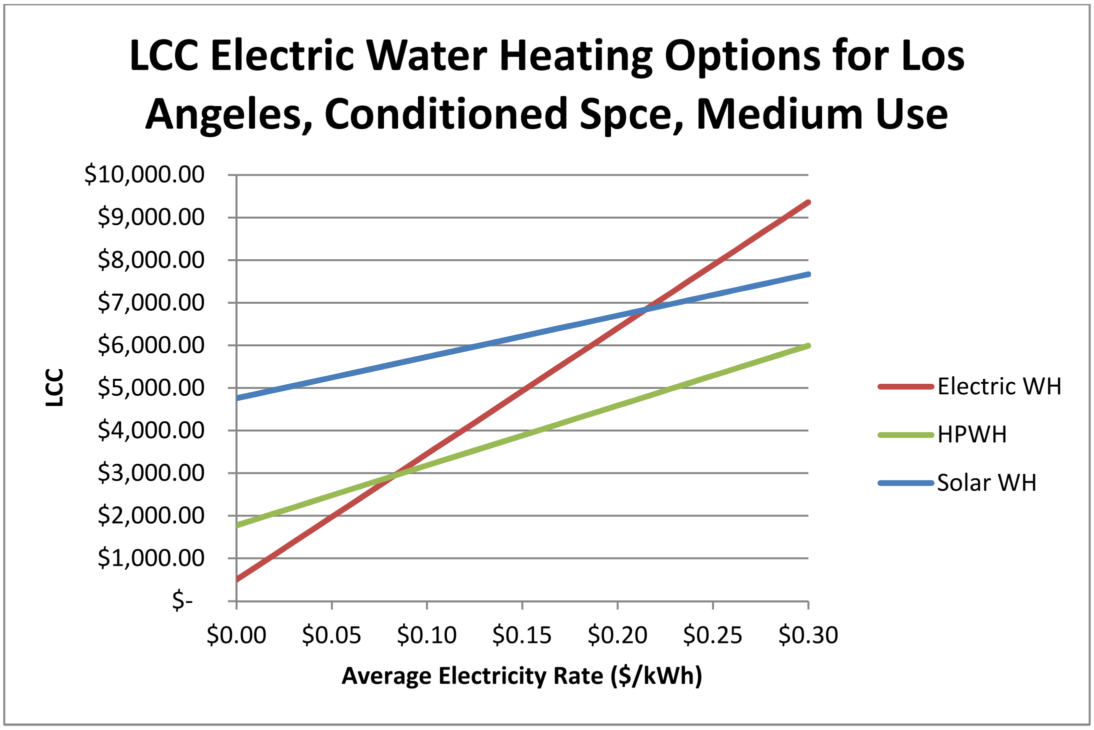


Figure 94: LCC of electric water heating options for Los Angeles at different rates

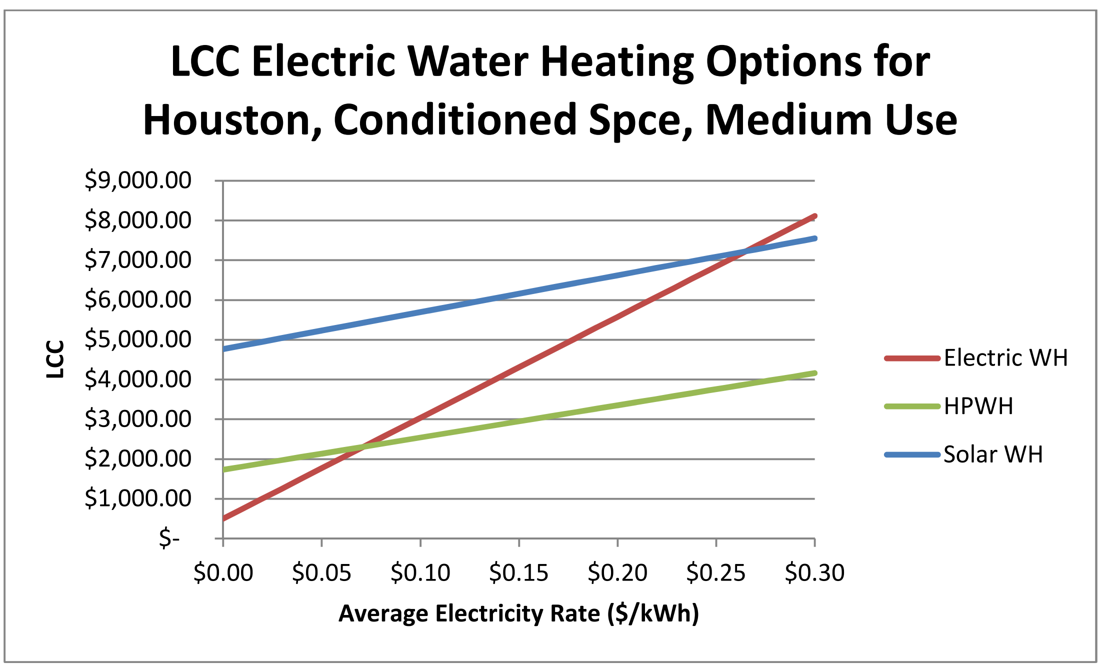


Figure 95: LCC of electric water heating options for Houston at different rates

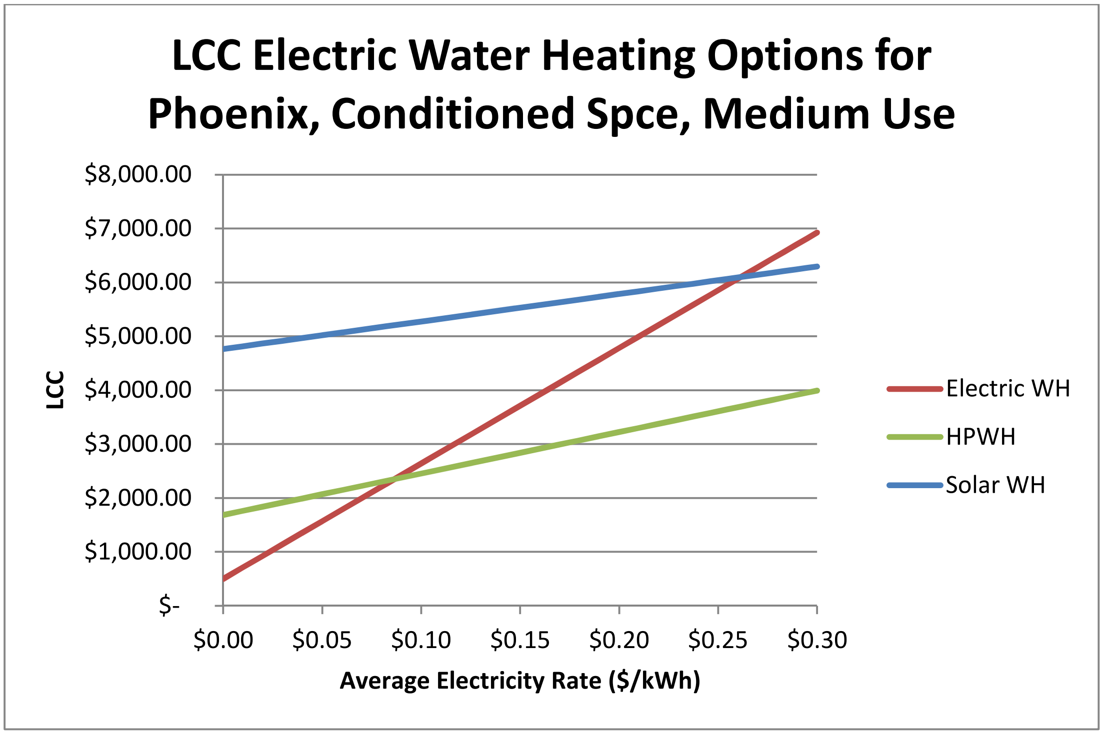


Figure 96: LCC of electric water heating options for Phoenix at different rates

8.2.7 Breakeven Costs

Breakeven cost is essentially how much each water heating system would need to cost (capital cost and installation costs) in order for it to be a cost neutral alternative to the base case of either gas or electric storage water heaters. The breakeven cost is calculated by setting the two life cycle costs equal to each other and then calculating the necessary capital cost. The breakeven cost can be expressed as:

$$Cost_{breakeven} = \frac{Cost_{base} + MC_{base} + OC_{base} - OC_{WH} - MC_{WH}}{1 - \left(\frac{N-y}{N}\right) SPV_y} \qquad (24)$$

Breakeven cost is useful for several reasons. For one, installation costs can vary by location (for example, in some situations a tankless water heater may need a new gas line or a HPWH may not need a louvered door). The breakeven cost gives what the total installed cost needs to be for the efficiency upgrade to be cost neutral and therefore none of those assumptions need to be made for the technology being considered. For homeowners, it's useful as guidance for whether or not to invest in a more efficient water heating option. If a homeowner can purchase the more efficient option for less than the breakeven cost, they are making a cost saving investment as well as an energy saving investment. Note that the breakeven costs calculated here do not include any incentives.

The breakeven costs for each case are given in Table 72-Table 77. Highlighted values indicate that the option has a breakeven cost larger than the installed cost assumed in the LCC section and is likely to be a cost saving measure. Out of all the technologies considered here, the HPWH has by far the most savings potential, especially when looking at medium and high use homes. Another factor of note is that none of the gas water heaters have a breakeven cost less than their previously assumed cost for the retrofit case. This doesn't necessarily mean they won't be cost effective (as previous results including incentives have shown), but does indicate that there is no technology that is a clear winner when looking at gas water heaters.

Climate	Installation Location	Tankless	Condensing	Solar Gas	HPWH	Solar Electric
Chicago	Conditioned	$1,119	$1,636	$2,148	$1,713	$2,360
Seattle	Conditioned	$927	$1,662	$2,142	$1,340	$1,563
Atlanta	Conditioned	$1,458	$1,849	$2,849	$1,641	$2,232
Los Angeles	Conditioned	$1,304	$1,554	$1,760	$1,988	$3,718
Houston	Conditioned	$1,592	$1,645	$1,613	$1,972	$2,483
Phoenix	Conditioned	$1,744	$1,786	$2,311	$1,419	$1,904
Chicago	Unconditioned	$1,376	$1,737	$2,565	$1,495	$2,580
Seattle	Unconditioned	$1,400	$1,776	$2,811	$1,505	$1,756
Atlanta	Unconditioned	$1,684	$1,881	$3,141	$1,593	$2,299
Los Angeles	Unconditioned	$1,178	$1,545	$2,384	$2,056	$3,851
Houston	Unconditioned	$1,202	$1,578	$2,333	$1,692	$2,341
Phoenix	Unconditioned	$1,350	$1,689	$2,722	$1,180	$1,686

Table 72: Breakeven cost for new construction, low use homes

Climate	Installation Location	Tankless	Condensing	Solar Gas	HPWH	Solar Electric
Chicago	**Conditioned**	$1,050	$1,718	$2,396	$2,235	$2,821
Seattle	**Conditioned**	$866	$1,733	$2,317	$1,733	$1,825
Atlanta	**Conditioned**	$1,381	$1,942	$3,305	$2,145	$2,771
Los Angeles	**Conditioned**	$1,239	$1,623	$2,159	$2,662	$4,535
Houston	**Conditioned**	$1,517	$1,719	$2,045	$2,662	$3,172
Phoenix	**Conditioned**	$1,658	$1,858	$2,917	$1,816	$2,475
Chicago	**Unconditioned**	$1,319	$1,819	$2,835	$1,881	$3,032
Seattle	**Unconditioned**	$1,324	$1,858	$2,979	$1,912	$2,005
Atlanta	**Unconditioned**	$1,590	$1,975	$3,597	$2,044	$2,824
Los Angeles	**Unconditioned**	$1,118	$1,612	$2,743	$2,714	$4,644
Houston	**Unconditioned**	$1,144	$1,644	$2,727	$2,236	$3,035
Phoenix	**Unconditioned**	$1,286	$1,757	$3,226	$1,488	$2,263

Table 73: Breakeven cost for new construction, medium use homes

Climate	Installation Location	Tankless	Condensing	Solar Gas	HPWH	Solar Electric
Chicago	**Conditioned**	$982	$1,802	$2,571	$2,492	$3,018
Seattle	**Conditioned**	$792	$1,820	$2,445	$1,888	$1,985
Atlanta	**Conditioned**	$1,286	$2,038	$3,572	$2,445	$3,036
Los Angeles	**Conditioned**	$1,167	$1,692	$2,386	$3,023	$4,978
Houston	**Conditioned**	$1,438	$1,797	$2,280	$3,075	$3,559
Phoenix	**Conditioned**	$1,571	$1,934	$3,344	$2,146	$2,928
Chicago	**Unconditioned**	$1,240	$1,907	$3,011	$2,039	$3,206
Seattle	**Unconditioned**	$1,239	$1,949	$3,099	$2,066	$2,150
Atlanta	**Unconditioned**	$1,486	$2,072	$3,874	$2,275	$3,089
Los Angeles	**Unconditioned**	$1,050	$1,678	$2,953	$3,017	$5,074
Houston	**Unconditioned**	$1,076	$1,709	$2,992	$2,563	$3,425
Phoenix	**Unconditioned**	$1,214	$1,830	$3,631	$1,725	$2,716

Table 74: Breakeven cost for new construction, high use homes

Climate	Installation Location	Tankless	Condensing	Solar Gas	HPWH	Solar Electric
Chicago	Conditioned	$591	$1,238	$1,486	$1,859	$2,601
Seattle	Conditioned	$399	$1,264	$1,480	$1,485	$1,805
Atlanta	Conditioned	$930	$1,450	$2,187	$1,786	$2,474
Los Angeles	Conditioned	$776	$1,155	$1,098	$2,133	$3,959
Houston	Conditioned	$1,063	$1,246	$952	$2,117	$2,725
Phoenix	Conditioned	$1,216	$1,388	$1,649	$1,564	$2,145
Chicago	Unconditioned	$847	$1,339	$1,903	$1,641	$2,822
Seattle	Unconditioned	$872	$1,378	$2,149	$1,651	$1,998
Atlanta	Unconditioned	$1,156	$1,482	$2,479	$1,739	$2,540
Los Angeles	Unconditioned	$650	$1,147	$1,723	$2,201	$4,092
Houston	Unconditioned	$674	$1,180	$1,672	$1,838	$2,582
Phoenix	Unconditioned	$822	$1,290	$2,060	$1,326	$1,928

Table 75: Breakeven cost for retrofit, low use homes

Climate	Installation Location	Tankless	Condensing	Solar Gas	HPWH	Solar Electric
Chicago	Conditioned	$521	$1,319	$1,734	$2,380	$3,063
Seattle	Conditioned	$337	$1,334	$1,656	$1,878	$2,066
Atlanta	Conditioned	$852	$1,544	$2,643	$2,291	$3,013
Los Angeles	Conditioned	$711	$1,224	$1,497	$2,807	$4,776
Houston	Conditioned	$989	$1,321	$1,384	$2,808	$3,414
Phoenix	Conditioned	$1,130	$1,459	$2,256	$1,962	$2,717
Chicago	Unconditioned	$791	$1,421	$2,173	$2,026	$3,273
Seattle	Unconditioned	$796	$1,459	$2,318	$2,058	$2,246
Atlanta	Unconditioned	$1,062	$1,576	$2,936	$2,189	$3,066
Los Angeles	Unconditioned	$590	$1,213	$2,081	$2,859	$4,886
Houston	Unconditioned	$616	$1,245	$2,066	$2,381	$3,276
Phoenix	Unconditioned	$758	$1,358	$2,565	$1,634	$2,505

Table 76: Breakeven cost for retrofit, medium use homes

Climate	Installation Location	Tankless	Condensing	Solar Gas	HPWH	Solar Electric
Chicago	**Conditioned**	$454	$1,404	$1,910	$2,637	$3,260
Seattle	**Conditioned**	$264	$1,422	$1,784	$2,034	$2,227
Atlanta	**Conditioned**	$758	$1,639	$2,910	$2,591	$3,278
Los Angeles	**Conditioned**	$639	$1,294	$1,724	$3,169	$5,220
Houston	**Conditioned**	$910	$1,398	$1,618	$3,220	$3,801
Phoenix	**Conditioned**	$1,043	$1,536	$2,683	$2,291	$3,170
Chicago	**Unconditioned**	$712	$1,509	$2,350	$2,184	$3,447
Seattle	**Unconditioned**	$711	$1,551	$2,438	$2,212	$2,392
Atlanta	**Unconditioned**	$958	$1,673	$3,213	$2,420	$3,330
Los Angeles	**Unconditioned**	$522	$1,280	$2,292	$3,162	$5,316
Houston	**Unconditioned**	$547	$1,311	$2,330	$2,708	$3,667
Phoenix	**Unconditioned**	$686	$1,432	$2,970	$1,870	$2,958

Table 77: Breakeven cost for retrofit, high use homes

While the breakeven cost doesn't include any assumption about the installed cost of the new water heater installed, an assumption of the cost of the baseline water heater (either gas or electric storage) is required. However, the installed cost of a gas or electric water heater can change drastically from home to home, particularly for gas water heaters. To determine what effect the assumed installed cost had on the breakeven cost, the breakeven cost was recalculated for a range of different baseline water heater costs for the case of medium use homes with the water heater installed in conditioned space as shown in Figure 97-Figure 102.

A reasonable range of installed costs for gas and electric water heaters was determined based on the most recent DOE water heater rulemaking *(69)*. The technical support documents provide the range of installed costs for each water heater as well as the average installed cost. For this sensitivity study, the 5th and 95th percentile installed costs for gas and electric water heaters were used to define the range of installed costs for these water heaters. For gas water heaters, the 5th percentile cost is $736 and the 95th percentile cost is $1883. For gas water heaters, these costs are $445 and respectively.

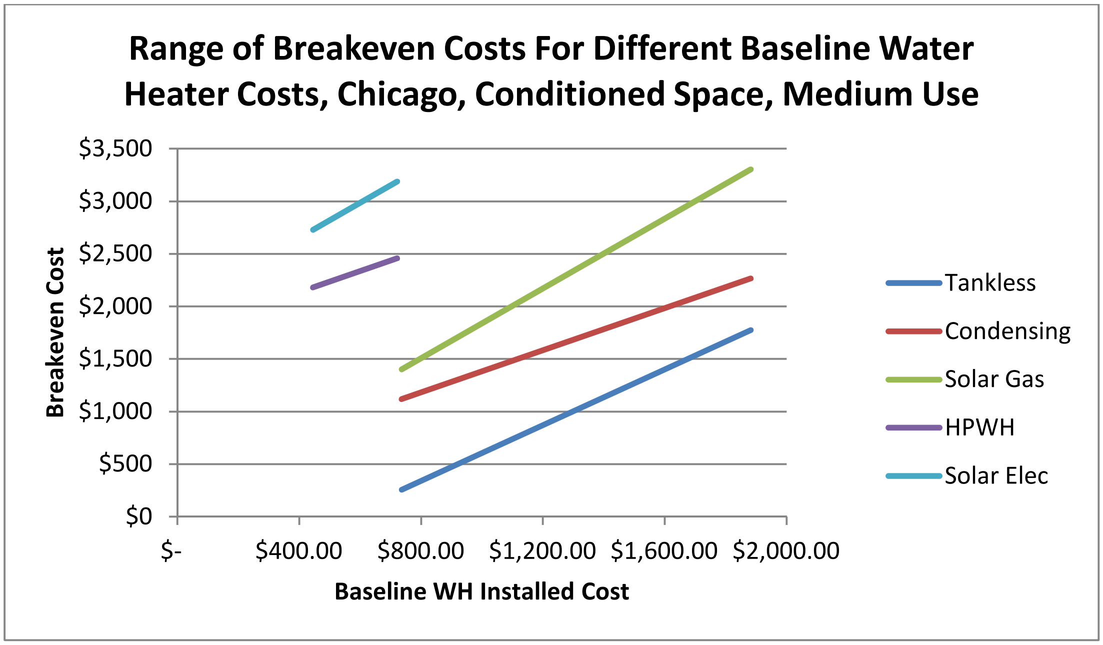


Figure 97: Breakeven cost at different baseline water heater costs in Chicago for medium use homes with the water heater in conditioned space

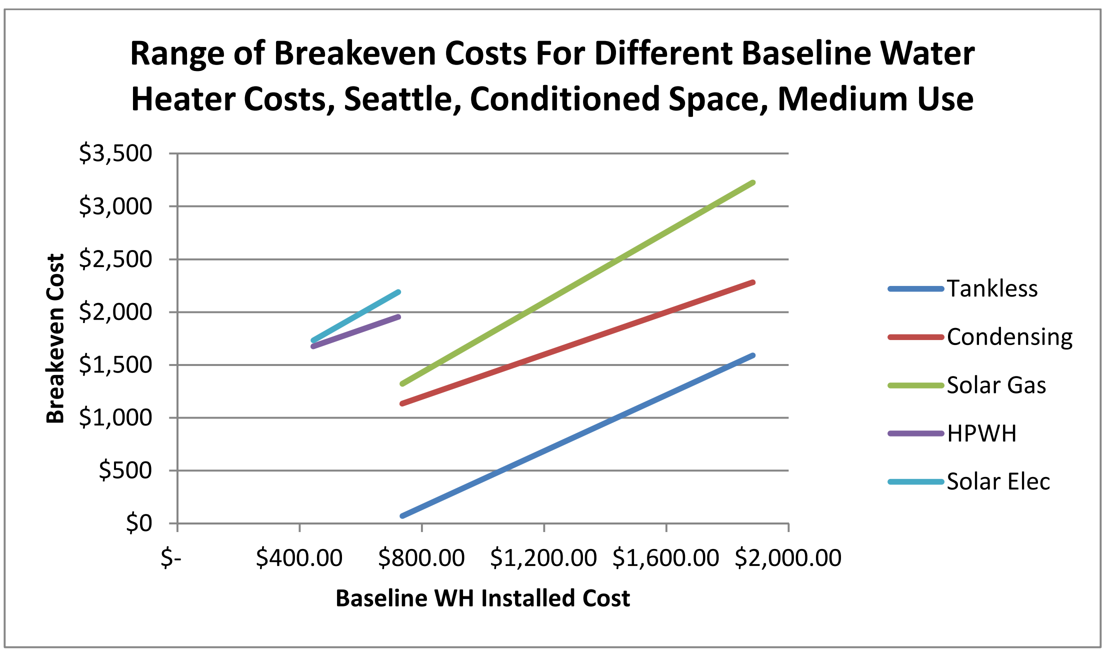


Figure 98: Breakeven cost at different baseline water heater costs in Seattle for medium use homes with the water heater in conditioned space

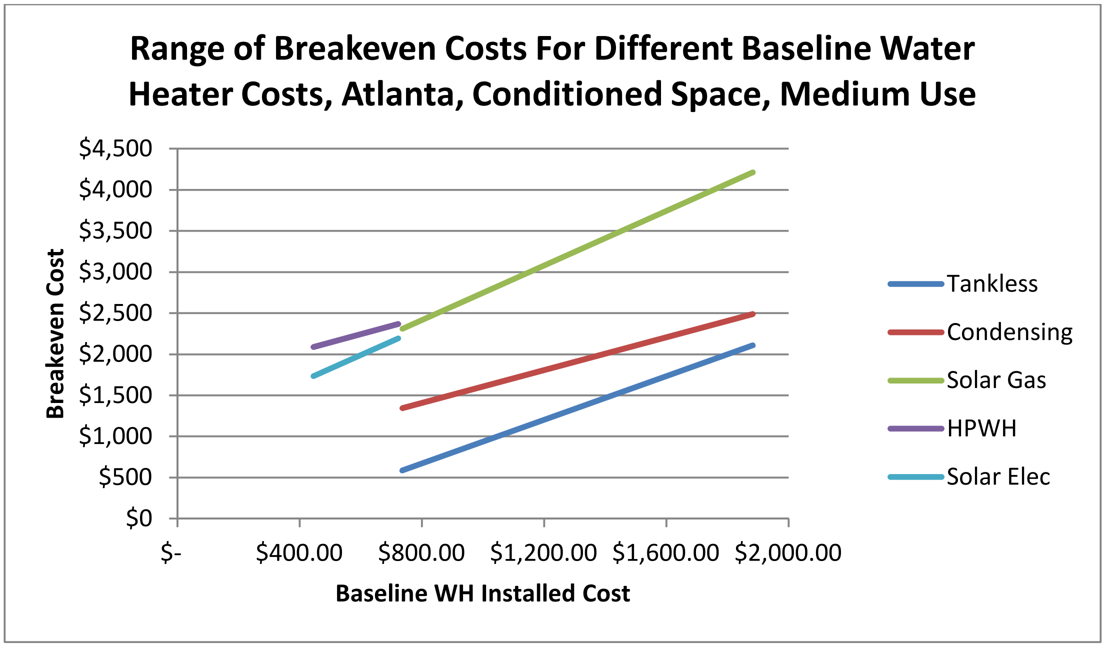


Figure 99: Breakeven cost at different baseline water heater costs in Atlanta for medium use homes with the water heater in conditioned space

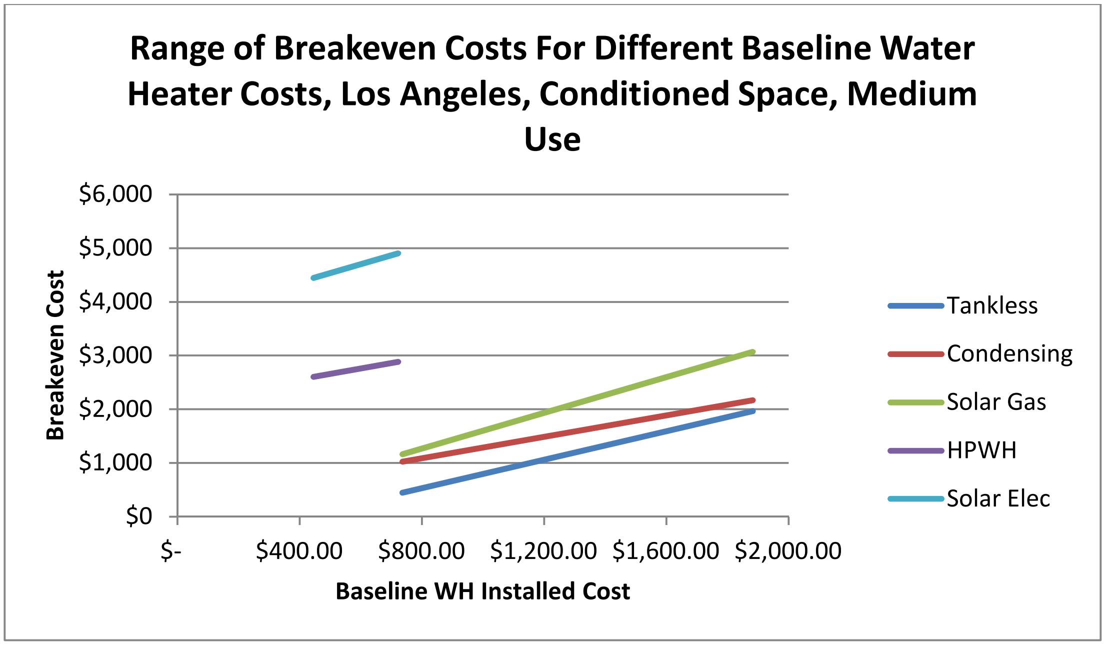


Figure 100: Breakeven cost at different baseline water heater costs in Los Angeles for medium use homes with the water heater in conditioned space

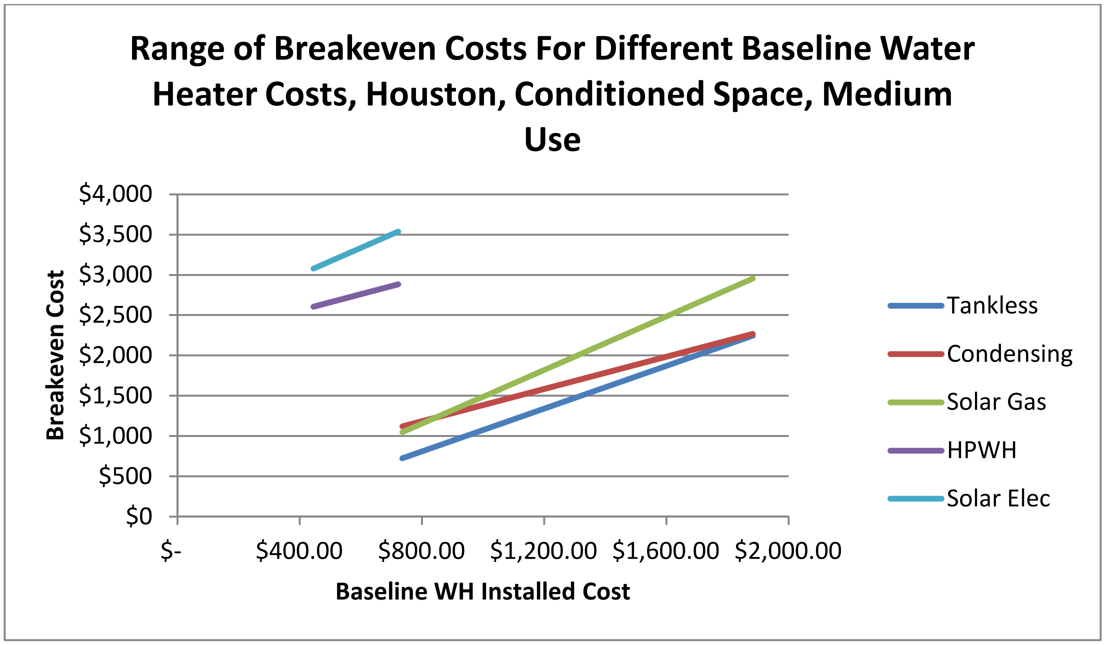


Figure 101: Breakeven cost at different baseline water heater costs in Houston for medium use homes with the water heater in conditioned space

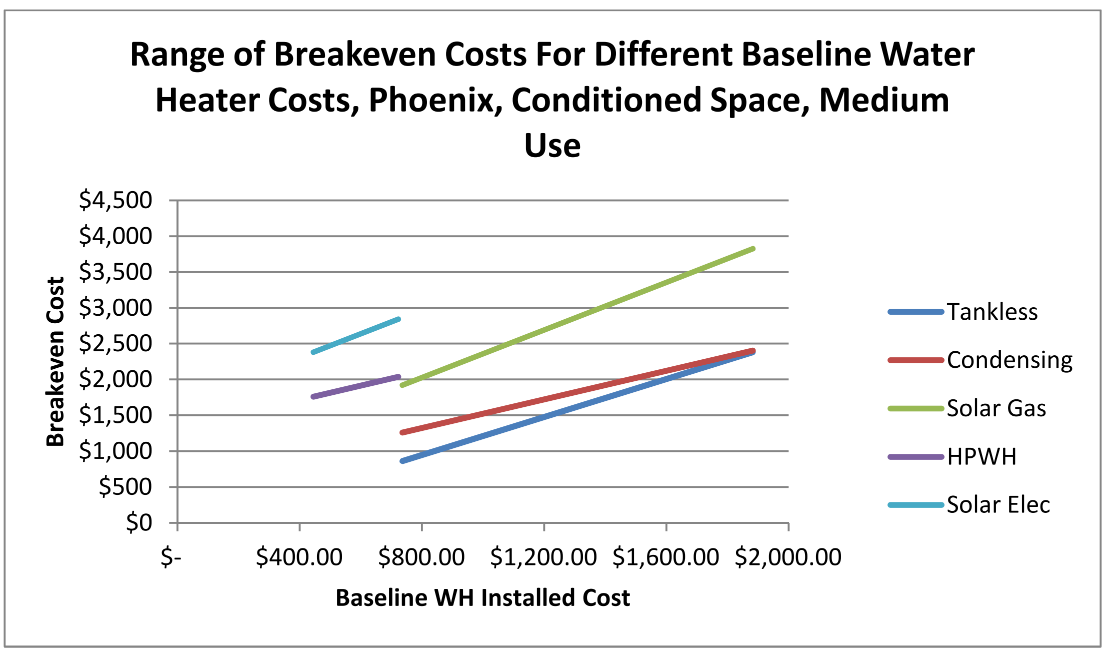


Figure 102: Breakeven cost at different baseline water heater costs in Phoenix for medium use homes with the water heater in conditioned space

8.3 Heat Pump Water Heater Mapping Study Results

For the heat pump water heater, annual simulations were also performed for every site in the continental US and Hawaii for which there is reliable weather data in the TMY3 dataset (63). In this study, a one minute draw profile was used instead of the six second draw profile that was used in past studies due to the large number of simulations that needed to be run and only medium use (corresponding to a 3 bedroom home) was considered. Simulations were also done for homes with a gas and electric water heater to allow a direct comparison to be made. In addition to simulating all three types of water heaters, installation in both conditioned and unconditioned space and two types of HVAC equipment (either a furnace/AC or an air source heat pump) were considered, leading to 12 different cases in total.

This study included every site in the continental US and Hawaii, but excluded Alaska due to issues modeling locations above the Arctic Circle in TRNSYS and the fact that HPWHs are generally better in warm, humid climates, making them unlikely to be effective in Alaska. Out of all of the TMY3 sites, only one was excluded from the final results. This site was Mount Washington in New Hampshire, which is famous for its high wind speeds and is not representative of where someone would actually build a home in that region. Results for Hawaii are not shown in the maps presented here but are consistent with those seen in southern Florida.

When comparing different types of water heating technologies, interactions with the space heating and cooling energy consumption are also considered, as well as the normalization energy previously discussed in Section 8.1. The governing equation for determining the overall water heater energy consumption is:

$$E_{WH,net} = E_{WH} + E_{nrmlz} + \Delta E_{heat} + \Delta E_{cool} \tag{25}$$

This equation is almost identical to Equation 18, which was used in the parametric study to compare cost and energy savings of different technologies. However, fan energy consumption is no longer considered in this equation. This is because all of the heating and cooling equipment for this case was oversized since there is no convenient way to auto size heating and cooling equipment in TRNSYS. Since fan speed and power was tied to the air conditioner size, this led to oversized fans which drew significantly more power than a properly sized fan would in that situation.

Since this study looked at homes in significantly more locations than the past study, additional building models were required. All of the internal gains, thermal mass, and infiltration and ventilation assumptions are identical to those described in Chapter 7: Home Models. The only difference is that a wider variety of homes needed to be considered, leading to a larger number of insulation levels for the homes in this study and the inclusion of homes with crawlspace foundations. The same foundation assumption (that all foundations in that state are whatever was most typical) is included in this study. Refer to Figure 50 for a map showing the prevalence of each type of foundation by state. The inclusion of the entire continental US led to several cases where crawlspaces were the most common foundation type. Insulation levels were taken from the Building America House Simulation Protocols (51) specifications for new construction homes and are consistent with IECC 2009 guidelines. Insulation levels for all homes are given in Table 78. All building foundations were modeled using the Winkelmann (60) methodology as previously discussed in Section 7.4.

2009 IECC Climate Zone	Ceiling	Frame Wall (Detached)	Frame Wall (Attached)	Floor	Basement Wall	Crawl Space Wall	Slab R-Value and Depth
1	30	13	13	13	0	0	0
2	30	13	13	13	0	0	0
3	30	13	13	19	0	5	0
4 except Marine	38	13	13	19	10	10	10, 2 ft
5 and Marine 4	38	13+5	19	30	10	10	10, 2 ft
6	49	13+5	19	30	15	10	10, 4 ft
7 and 8	49	21	19	38	15	10	10, 4 ft

Table 78: R-values for the building envelopes of homes used in the HPWH mapping study (51). Note that 13+5 indicates R-13 cavity insulation with R-5 sheathing.

8.3.1 Unconditioned Space Results

Unconditioned space for this study was defined as a basement if a home had one or the garage otherwise, as this is the most likely place for a water heater to be installed. In unconditioned space, the efficiency of the HPWH is driven by the space temperature and humidity as the interactions with the buildings heating and cooling systems is fairly minimal. In the case of garages, the unconditioned space temperature is most strongly impacted by the outside ambient air conditions. In the case of basements, ground temperature and the space temperature have a more significant impact on the space temperature, although there is some infiltration into the basement from outside.

The annual efficiency (system COP) of the HPWH located in unconditioned space is shown in Figure 103. Since the HPWH works best in locations with a high wet bulb temperature, performance is highest in hot and humid locations such as southern Florida and Texas. This also means that very hot but dry locations such as parts of southern California and Arizona can also have relatively high efficiencies. Even in cold climates, the heat pump will provide some of the energy to heat the tank over the course of a year as shown in Figure 104. This leads to the vast majority of locations having a system COP over 1.

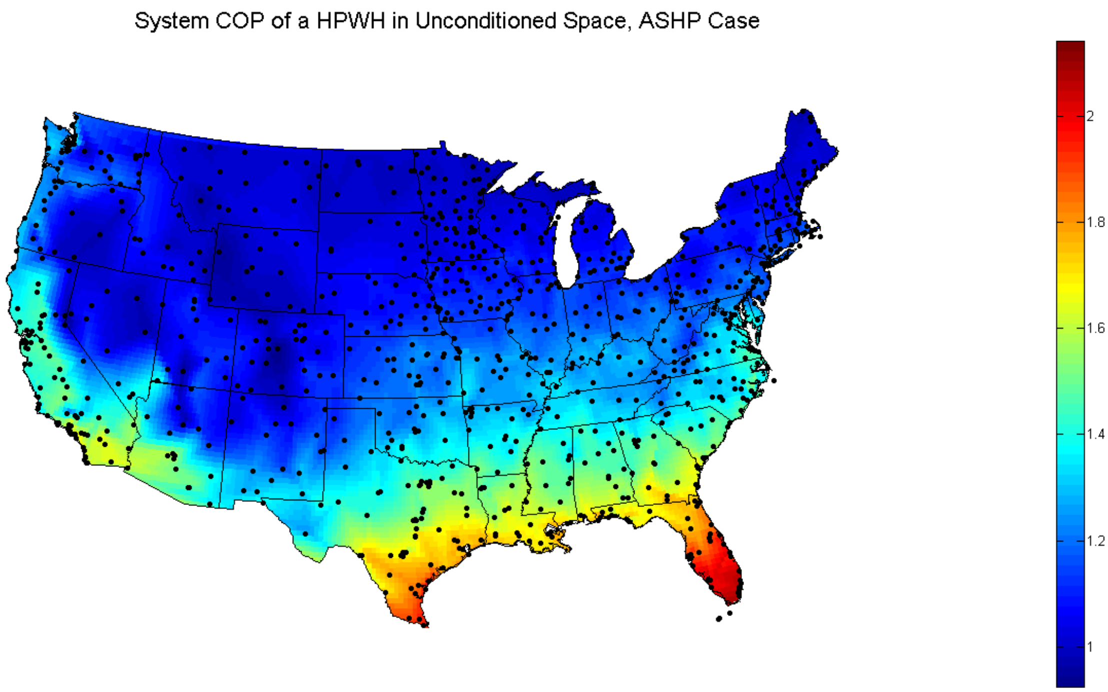


Figure 103: Annual efficiency of the HPWH in unconditioned space

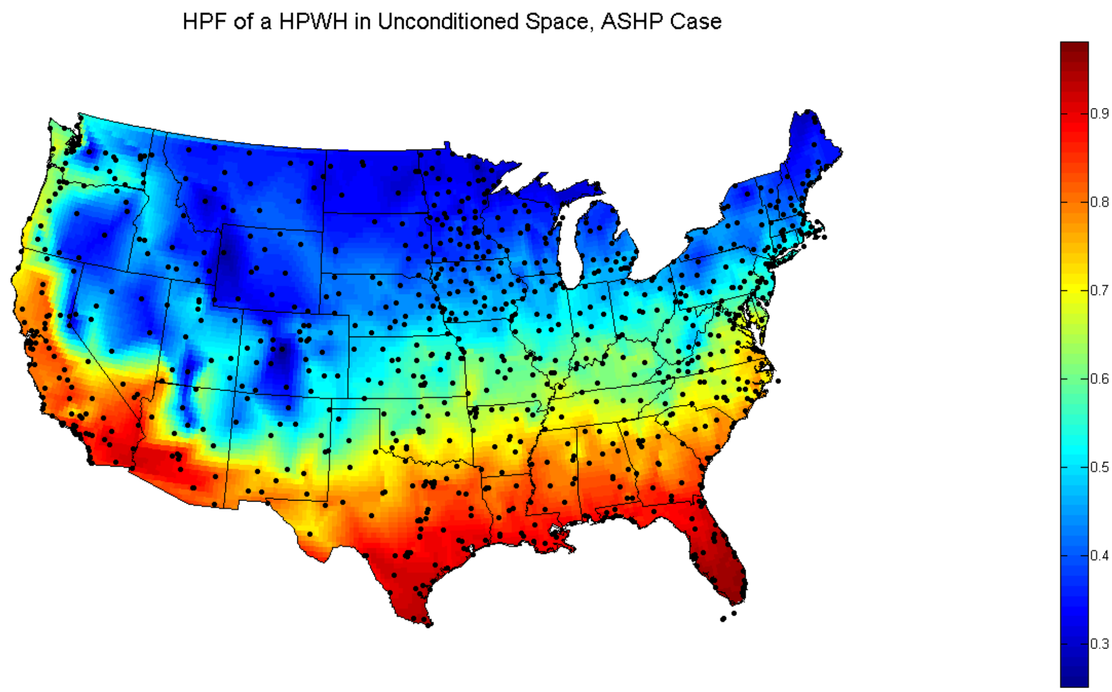


Figure 104: Heat pump fraction of the HPWH in unconditioned space

Even in cold climates there is some savings when comparing a HPWH to an electric water heater as shown in Figure 105. These savings are much larger in the southeast and along the west coast where there is relatively warm and humid air. When comparing a HPWH to a gas water heater, there is not always positive source energy savings as shown in Figure 106. The site to source ratio for electricity is much higher than for gas (3.365 for electricity, 1.092 for gas), so there must be very significant site energy savings from a HPWH for it to be an effective replacement. This only happens in the locations that are best suited for a HPWH, which is along the Gulf Coast and in areas of southern California and Arizona. To help differentiate which sites have positive source energy savings from those with negative savings, a "go/no go" map is provided in Figure 107. Sites that have positive source energy savings are green (go) and those with negative energy savings are red (no go).

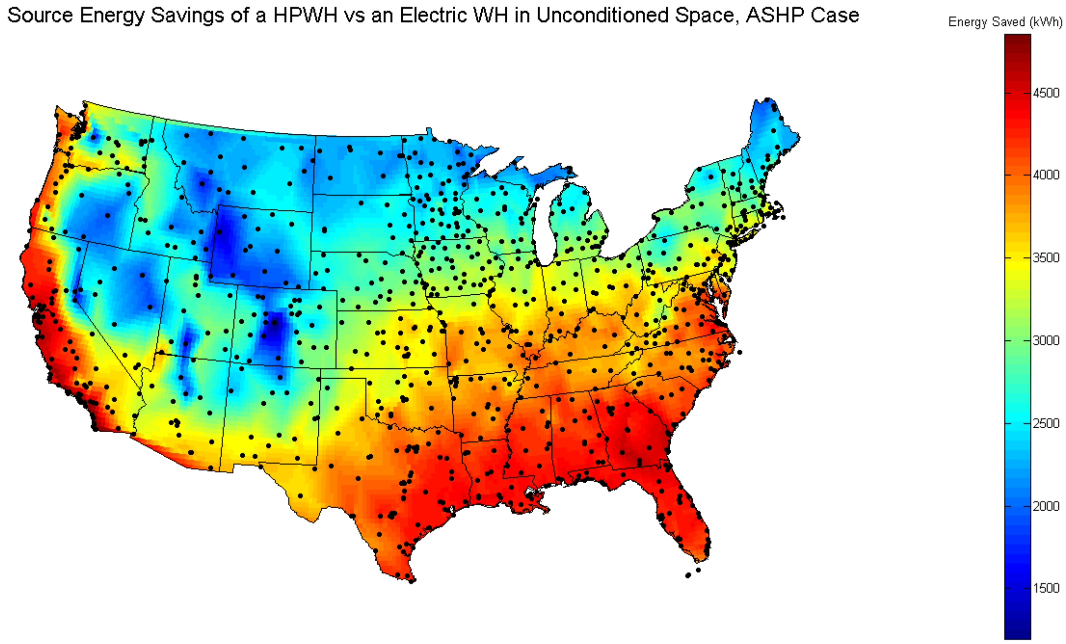


Figure 105: Source energy savings of a HPWH vs. an electric water heater in unconditioned space

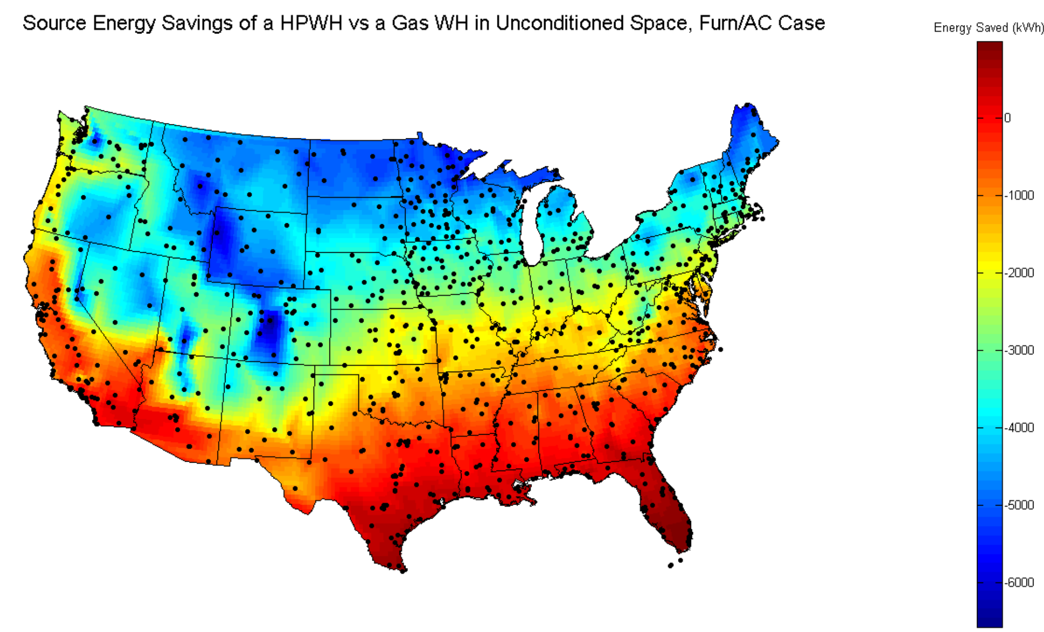


Figure 106: Source energy savings of a HPWH vs. a gas water heater in unconditioned space

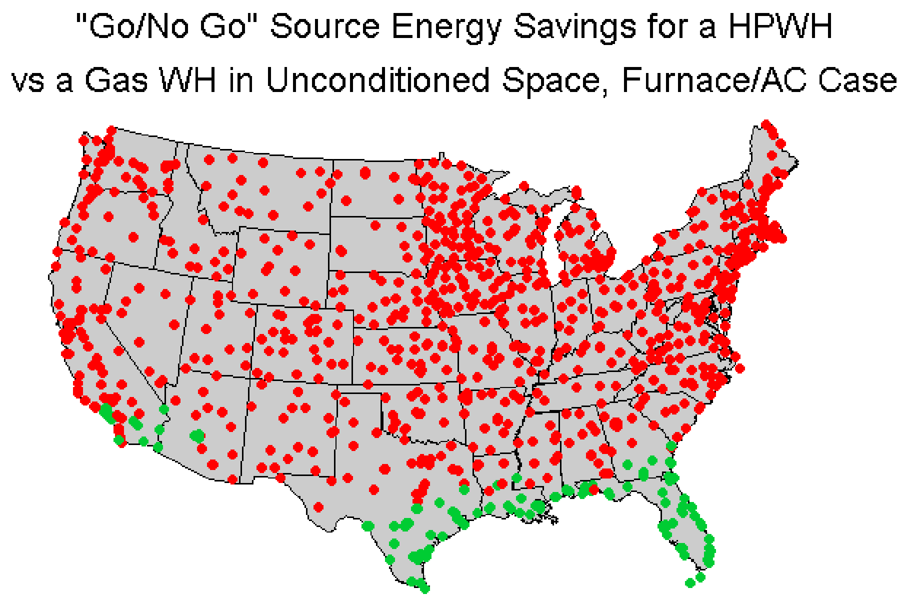


Figure 107: "Go/No Go" map of HPWH source energy savings vs. a gas water heater in unconditioned space

8.3.2 Conditioned Space Results

In conditioned space, the efficiency of the heat pump water heater varies less from location to location since the indoor air temperature is kept between 71-76°F year round. However, mains water temperature still varies by location, as does the indoor air humidity, which is uncontrolled. This keeps the annual efficiency of the HPWH much higher in colder climates as can be seen in Figure 108. However, in conditioned space the HPWH can have a significant impact on the heating and cooling energy consumption of the home. In colder climates, this means an increase in the heating energy consumption, while in warmer climates there is a beneficial reduction in cooling energy consumption. Figure 109 shows the net change in heating and cooling energy consumption for each location.

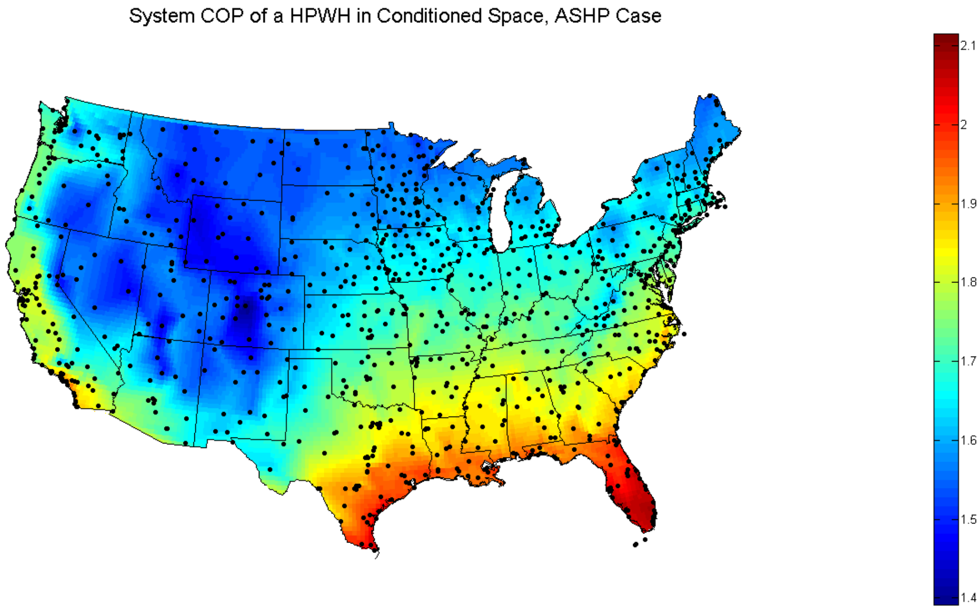


Figure 108: Annual efficiency of a HPWH in conditioned space

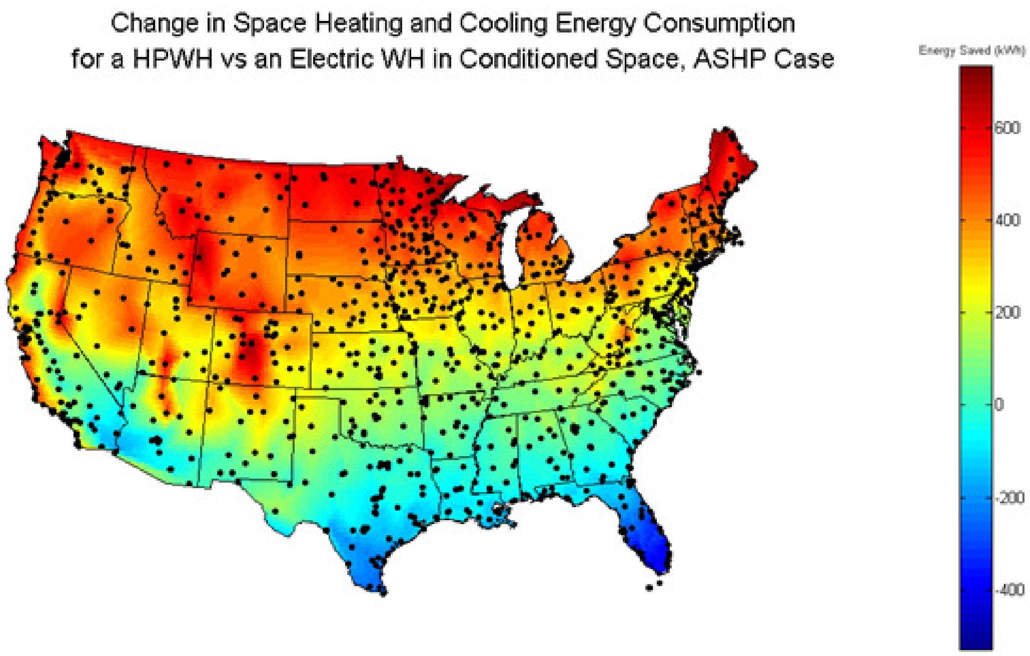


Figure 109: Net change in space heating and cooling energy consumption for a HPWH vs. an electric water heater in conditioned space

In conditioned space the source energy savings is generally larger for HPWHs vs. electric water heaters than in the unconditioned space as shown in Figure 110. This is because the HPWH benefits from the relatively constant space temperature and high humidity while the home cooling systems can benefit from the space cooling provided by the HPWH. This is especially true in areas with a negligible or zero space heating load such as southern Florida. The impact on the space heating system in cold climates is less of a detriment on the HPWH performance than having it located in a cold area where the heat pump may not be able to run for a significant portion of the year, leading to higher source energy savings from locating the HPWH in conditioned space vs. the unconditioned space case.

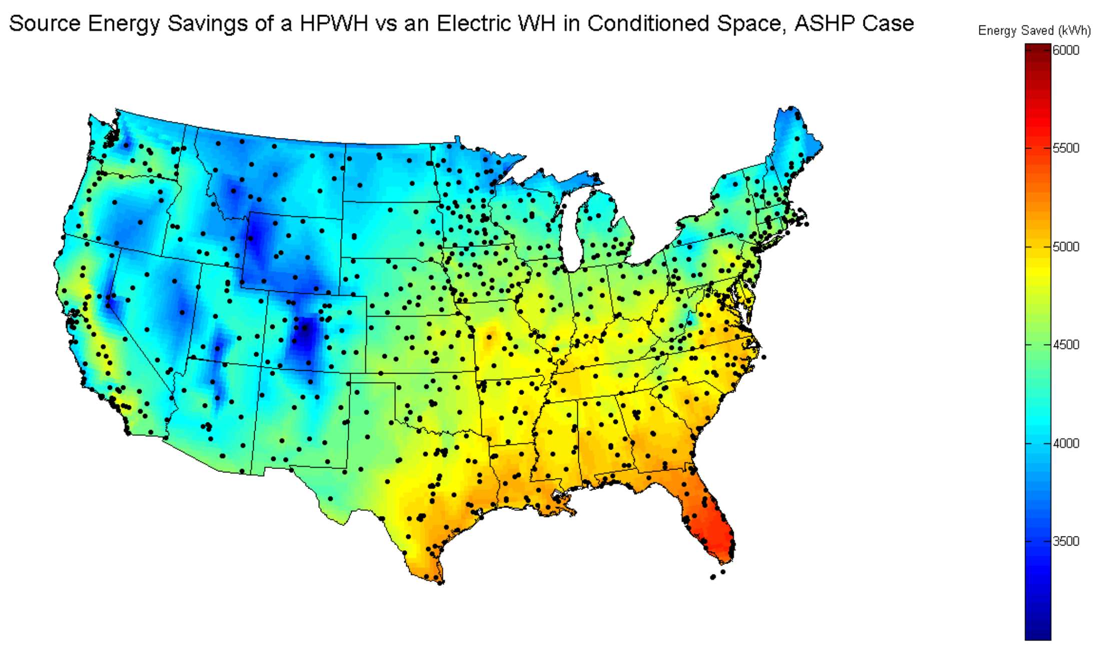


Figure 110: Source energy savings of a HPWH vs. an electric water heater in conditioned space

In the case of a HPWH vs. a gas water heater in conditioned space, the area in which a HPWH provides net source energy savings is larger than for the case of a HPWH in unconditioned space as can be seen in Figure 111 and Figure 112. For the Gulf Coast region, the area in which HPWHs can be an effective replacement has moved much further north, now including parts of Arkansas and North Carolina. There are also more sites in Arizona and California that have positive source energy savings. The reasons for this are similar to those in the case of a HPWH vs. an electric water heater: the system benefits from the relatively constant space temperature, which may allow the heat pump to be utilized for more of the year, while the change in space heating and cooling loads is less of a factor. In addition, in the case of a gas water heater, the heat is provided by a gas furnace and the cooling by an air conditioner, while in the electric water heater case both heating and cooling are provided by an ASHP. This means the source energy impact of a change in heating energy consumption is smaller than the impact of a change in cooling energy consumption since the site to source ratio for gas is about 1/3 that of electricity.

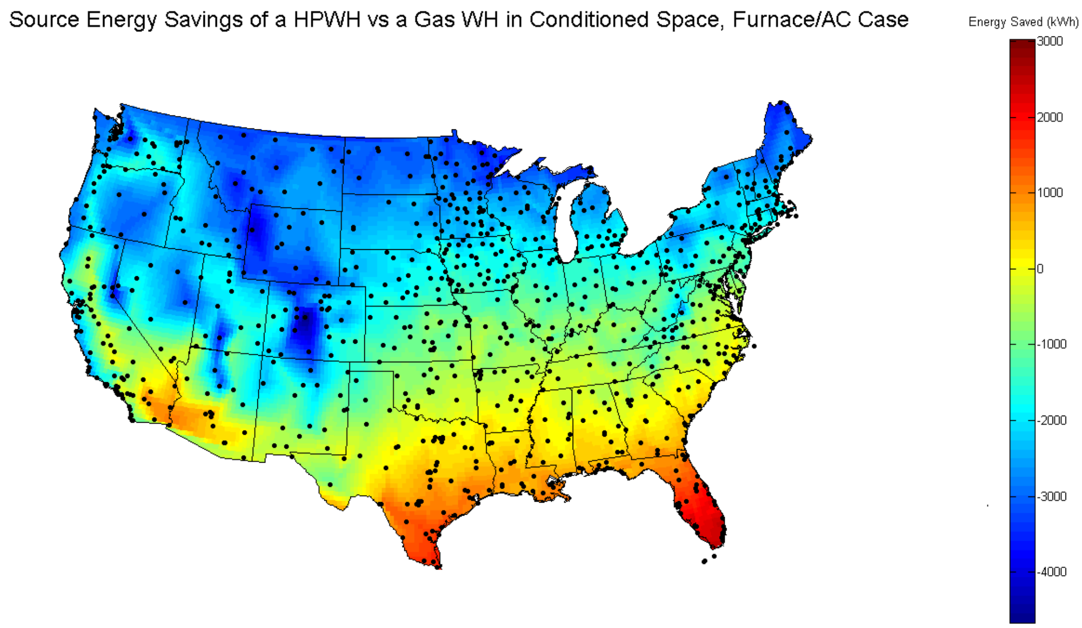


Figure 111: Source energy savings of a HPWH vs. a gas water heater in conditioned space

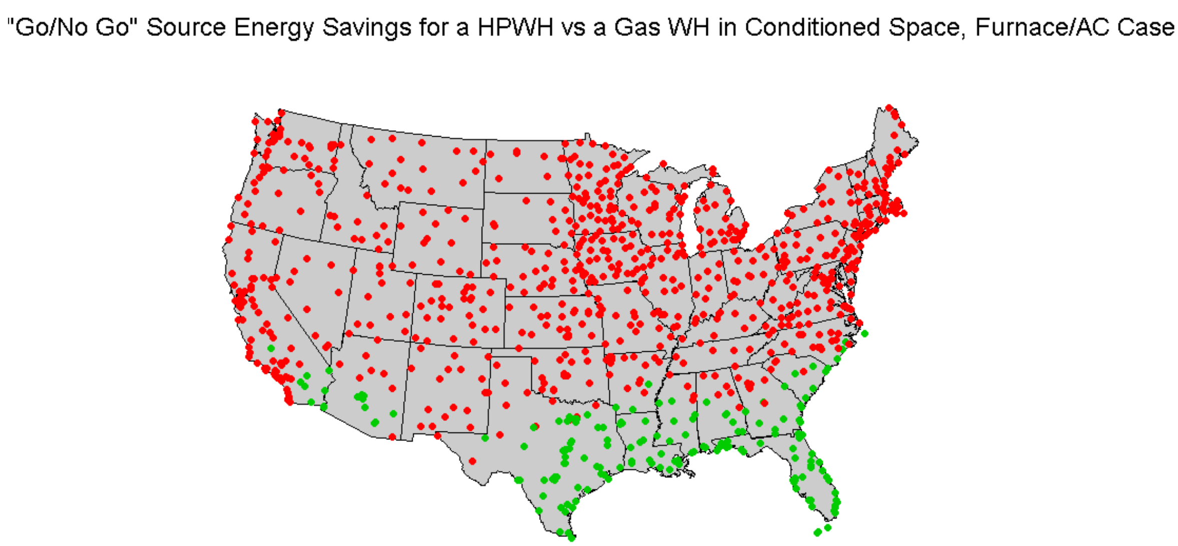


Figure 112: "Go/No Go" map of source energy savings for a HPWH vs. a gas water heater in conditioned space

8.4 Distribution System Study Results

For this distribution system study, two different types of water heaters were simulated: heat pump water heaters and electric storage water heaters. Since HPWHs tend to produce water at a lower average outlet temperature (refer to Chapter 4 for more details), their distribution losses are different than those of typical water heaters. The electric water heater was run as a baseline to compare the HPWH against so that changes in distribution losses in different locations and at different usage could be examined. It was theorized prior to these simulations that the HPWH may have significantly higher losses due to more wasted water (and therefore energy). However, the net distribution losses for HPWHs and standard water heaters were found to be fairly close in all cases.

There are two components to DHW distribution losses. The first of these is pipe losses, which is heat lost through the pipes of the distribution system. Pipe losses occur both during draws and immediately following draw events as the pipes are left full of hot water. Pipe losses are assumed to go to either conditioned or unconditioned space depending on where the pipe is located and can therefore impact space heating and cooling loads. The second component of the DHW distribution losses is wasted energy. For mixed draws, a minimum temperature of water (105 °F) is required for the draw to be useful. Any time this minimum temperature is not supplied, the water is not used and any energy that went into heating that water is wasted. This is intended to model typical behavior: occupants will wait for a shower or sink to get warm before using it, and they will stop using it if it gets too cold. These types of losses do not impact space heating and cooling loads as the wasted energy goes down the drain.

8.4.1 Distribution System Pipe Losses

For HPWHs, the pipe losses are less than for a standard water heater due to the sag in outlet temperature. The difference between the pipe losses for electric water heaters and HPWHs are very consistent across all climates, DHW use, and water heater installation location. In every case, the losses from the HPWH are about 14% lower than for an electric water heater. The distribution losses vary more in locations with colder mains temperatures as more energy is required to meet the load. In Chicago, where the most extreme variation in pipe losses occurs, the losses from the HPWH vary from 12.1% less than an electric water heater to 16.3% less. There is an increase in the HPWH losses relative to the electric water heater losses as the draw volume increase as higher draw volumes lead to more times where the HPWH has to switch to electric elements (electric elements only come on when the tank is fairly depleted, which corresponds to an even lower outlet temperature). The pipe losses in every case are provided in Table 79, while Figure 113 and Figure 114 show the pipe losses for cases in Chicago and different locations subjected to a medium use draw profile respectively.

Location	WH Location	Draw	Electric	HPWH	% Difference
Chicago	Conditioned	Low	442.0	378.2	-14.4%
Chicago	Conditioned	Med	605.3	513.8	-15.1%
Chicago	Conditioned	High	743.2	622.2	-16.3%
Chicago	Unconditioned	Low	452.0	397.1	-12.1%
Chicago	Unconditioned	Med	618.7	538.1	-13.0%
Chicago	Unconditioned	High	759.0	652.0	-14.1%
Seattle	Conditioned	Low	448.9	386.1	-14.0%
Seattle	Conditioned	Med	614.3	524.9	-14.6%
Seattle	Conditioned	High	754.7	636.2	-15.7%
Seattle	Unconditioned	Low	458.0	398.8	-12.9%
Seattle	Unconditioned	Med	626.5	538.6	-14.0%
Seattle	Unconditioned	High	769.0	651.0	-15.4%
Atlanta	Conditioned	Low	437.2	376.3	-13.9%
Atlanta	Conditioned	Med	597.1	511.3	-14.4%
Atlanta	Conditioned	High	735.8	621.8	-15.5%
Atlanta	Unconditioned	Low	444.5	384.3	-13.6%
Atlanta	Unconditioned	Med	606.7	520.2	-14.3%
Atlanta	Unconditioned	High	747.3	631.4	-15.5%
Los Angeles	Conditioned	Low	433.7	374.6	-13.6%
Los Angeles	Conditioned	Med	592.3	509.8	-13.9%
Los Angeles	Conditioned	High	729.2	621.8	-14.7%
Los Angeles	Unconditioned	Low	434.9	375.7	-13.6%
Los Angeles	Unconditioned	Med	593.6	510.2	-14.1%
Los Angeles	Unconditioned	High	730.9	619.9	-15.2%
Houston	Conditioned	Low	428.2	369.8	-13.6%
Houston	Conditioned	Med	584.4	502.9	-14.0%
Houston	Conditioned	High	720.7	613.6	-14.9%
Houston	Unconditioned	Low	425.1	368.2	-13.4%
Houston	Unconditioned	Med	579.5	499.9	-13.7%
Houston	Unconditioned	High	715.3	611.1	-14.6%
Phoenix	Conditioned	Low	426.0	369.0	-13.4%
Phoenix	Conditioned	Med	581.8	502.1	-13.7%
Phoenix	Conditioned	High	717.7	613.0	-14.6%
Phoenix	Unconditioned	Low	419.1	363.4	-13.3%
Phoenix	Unconditioned	Med	572.0	495.2	-13.4%
Phoenix	Unconditioned	High	706.7	605.4	-14.3%

Table 79: Pipe losses (in kWh) from the DHW distribution system for electric storage and HPWHs

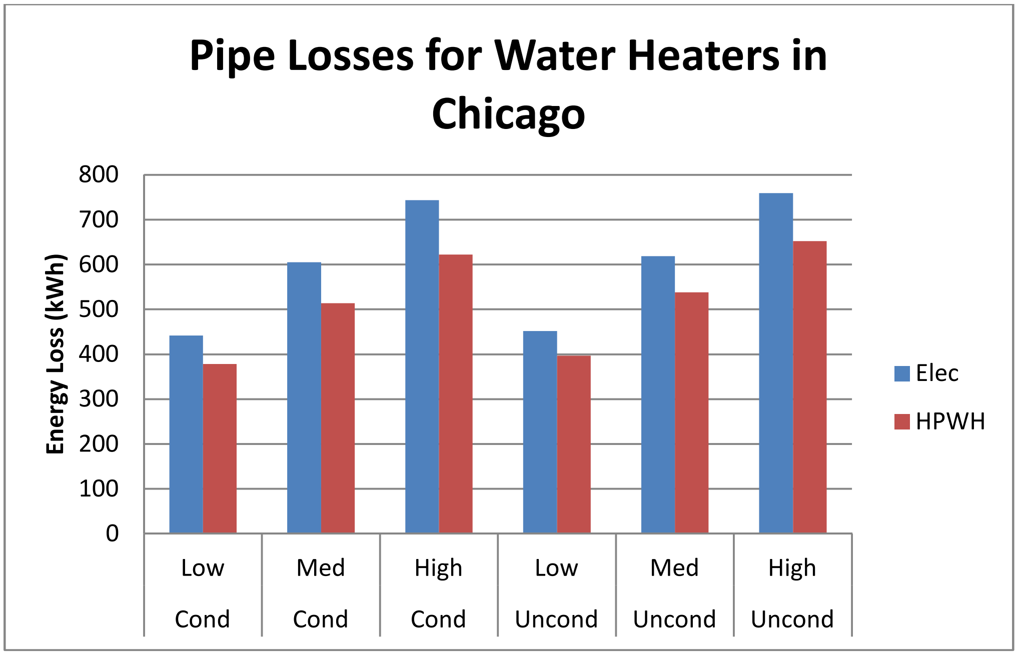


Figure 113: Pipe losses in Chicago for electric storage and HPWHs

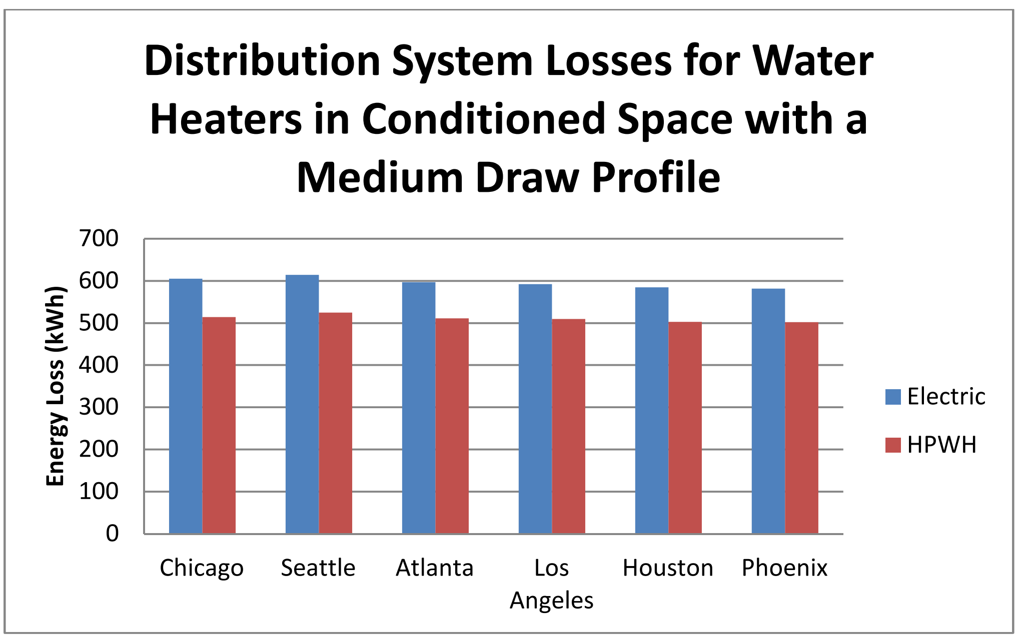


Figure 114: Pipe losses for electric storage and HPWHs under a medium draw profile

8.4.2 Distribution System Wasted Energy

The wasted energy is always larger for a HPWH than a standard electric storage water heater. This is also due to the sag that was observed with HPWHs, which leads to the temperature of water delivered to drop below the set point more often. However, there is much more variation in the wasted energy than pipe losses when looking at different locations and draw profiles. The amount of wasted energy depends on the draw volume, with larger draw volumes leading to higher wasted energy. In particular, for HPWHs a dramatic increase in the wasted energy is seen when going from medium to high draw profiles. This is due to the high draw profile leading to more situations where the heat pump cannot keep up with demand and the electric elements must turn on (this phenomenon can be seen by comparing Figure 69 and Figure 70 in the discussion of HPWH performance without distribution systems). There are also significant variations in the wasted energy in different locations. This is partially due to the lower draw that comes from having a warmer mains temperature. The higher mains temperature is also an important factor in tempering, which affects the wasted energy. With a higher mains temperature, less hot water is required to temper down to the use temperature. This leads to lower flow rates from the water heater, which makes it less likely to be depleted and makes it more likely to be able to provide water at a useful temperature for the entire duration of the draw. The wasted energy in every case is provided in Table 80, while Figure 115 and Figure 116 show the energy for all cases in Chicago and different locations subjected to a medium use draw profile respectively.

Location	WH Location	Draw	Electric	HPWH	% Difference
Chicago	Conditioned	Low	230.5	335.6	45.6%
Chicago	Conditioned	Med	323.0	468.7	45.1%
Chicago	Conditioned	High	435.4	731.7	68.0%
Chicago	Unconditioned	Low	221.9	331.4	49.4%
Chicago	Unconditioned	Med	316.2	460.7	45.7%
Chicago	Unconditioned	High	429.6	724.3	68.6%
Seattle	Conditioned	Low	198.8	293.9	47.9%
Seattle	Conditioned	Med	290.2	420.8	45.0%
Seattle	Conditioned	High	391.1	655.6	67.6%
Seattle	Unconditioned	Low	195.0	301.3	54.5%
Seattle	Unconditioned	Med	288.3	435.1	50.9%
Seattle	Unconditioned	High	390.1	691.5	77.3%
Atlanta	Conditioned	Low	120.3	187.9	56.1%
Atlanta	Conditioned	Med	182.8	267.6	46.4%
Atlanta	Conditioned	High	246.6	423.7	71.8%
Atlanta	Unconditioned	Low	116.2	196.5	69.2%
Atlanta	Unconditioned	Med	179.6	277.7	54.6%
Atlanta	Unconditioned	High	243.2	441.7	81.6%
Los Angeles	Conditioned	Low	115.9	175.4	51.3%
Los Angeles	Conditioned	Med	179.5	256.2	42.8%
Los Angeles	Conditioned	High	244.4	412.2	68.6%
Los Angeles	Unconditioned	Low	115.2	179.3	55.6%
Los Angeles	Unconditioned	Med	178.5	270.1	51.3%
Los Angeles	Unconditioned	High	242.9	430.5	77.2%
Houston	Conditioned	Low	68.1	112.1	64.6%
Houston	Conditioned	Med	112.5	166.3	47.8%
Houston	Conditioned	High	156.5	270.0	72.5%
Houston	Unconditioned	Low	67.7	111.4	64.5%
Houston	Unconditioned	Med	111.3	164.3	47.6%
Houston	Unconditioned	High	155.3	269.9	73.8%
Phoenix	Conditioned	Low	20.6	56.4	174.1%
Phoenix	Conditioned	Med	45.1	82.0	81.7%
Phoenix	Conditioned	High	71.3	154.8	117.0%
Phoenix	Unconditioned	Low	22.1	58.0	162.3%
Phoenix	Unconditioned	Med	47.0	89.9	91.3%
Phoenix	Unconditioned	High	73.1	155.1	112.3%

Table 80: Wasted energy (in kWh) from the DHW distribution system for electric storage and HPWHs

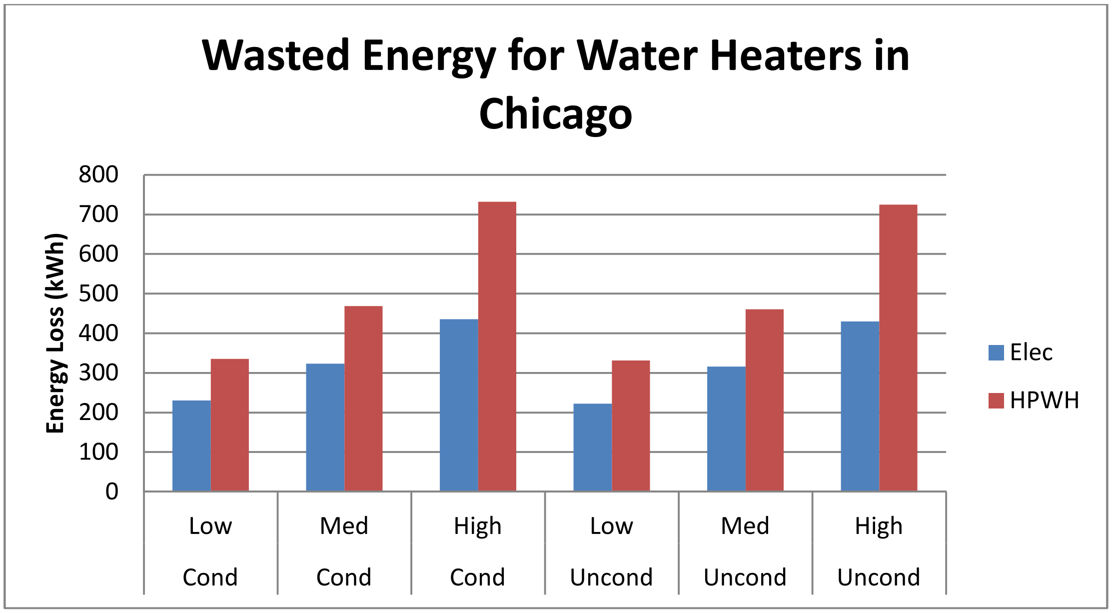


Figure 115: Wasted energy in Chicago for electric storage and HPWHs

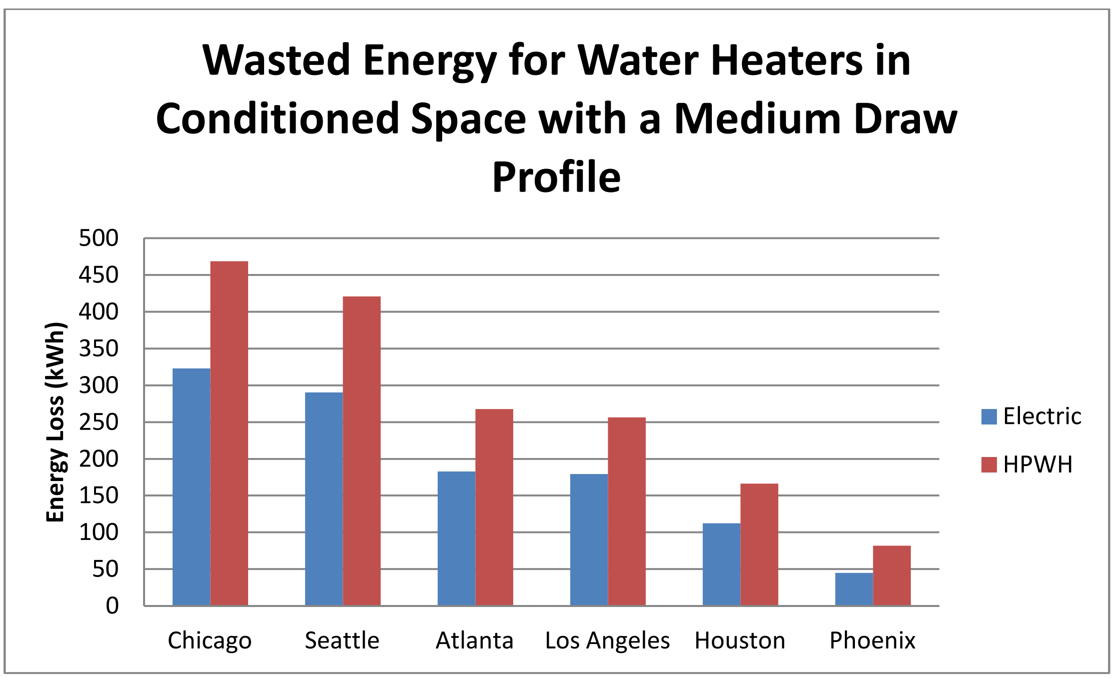


Figure 116: Wasted energy for electric storage and HPWHs under a medium draw profile

8.4.3 Net Distribution System Losses

The net distribution losses are simply the sum of the previous two factors. When looking at the net distribution system losses, it is important to remember that even if the net distribution system losses are similar to standard electric water heaters (as they often are) the two components of these losses are not necessarily the same. This is especially important when considering integrating these results into whole home simulations, as only the pipe losses impact the space heating and cooling loads.

In general, the net distribution system losses for a HPWH are indeed very similar to those of an electric storage water heater. The largest difference between the two is a 15.8%, (188 kWh) increase, increase in the distribution losses, while the smallest is a 6.8% (42.8 kWh) reduction. On average across all of the cases considered here, there is a 3% increase in the distribution system losses for a HPWH over a typical electric water heater. The largest factor influencing whether a HPWH has higher distribution losses is the wasted energy as the pipe losses for a HPWH relative to a standard water heater are fairly consistent in all cases. This leads to HPWHs having higher distribution losses in cold climates where the wasted energy is relatively large and smaller distribution losses in warm climates where the wasted energy is relatively small. The large jump when going from a medium to a large draw profile that was seen in the wasted energy is also apparent in these results. The net distribution system losses in every case is provided in Table 81, while Figure 117 and Figure 118 show the energy for all cases in Chicago and different locations subjected to a medium use draw profile respectively.

Location	WH Location	Draw	Electric	HPWH	% Difference
Chicago	Conditioned	Low	672.5	713.8	6.2%
Chicago	Conditioned	Med	928.3	982.5	5.8%
Chicago	Conditioned	High	1178.7	1353.8	14.9%
Chicago	Unconditioned	Low	673.9	728.5	8.1%
Chicago	Unconditioned	Med	934.9	998.8	6.8%
Chicago	Unconditioned	High	1188.6	1376.4	15.8%
Seattle	Conditioned	Low	647.7	680.0	5.0%
Seattle	Conditioned	Med	904.5	945.7	4.6%
Seattle	Conditioned	High	1145.8	1291.8	12.7%
Seattle	Unconditioned	Low	653.0	700.1	7.2%
Seattle	Unconditioned	Med	914.8	973.7	6.4%
Seattle	Unconditioned	High	1159.1	1342.5	15.8%
Atlanta	Conditioned	Low	557.5	564.2	1.2%
Atlanta	Conditioned	Med	779.9	778.9	-0.1%
Atlanta	Conditioned	High	982.4	1045.5	6.4%
Atlanta	Unconditioned	Low	560.7	580.8	3.6%
Atlanta	Unconditioned	Med	786.3	797.8	1.5%
Atlanta	Unconditioned	High	990.5	1073.1	8.3%
Los Angeles	Conditioned	Low	549.6	550.1	0.1%
Los Angeles	Conditioned	Med	771.7	766.0	-0.7%
Los Angeles	Conditioned	High	973.6	1034.0	6.2%
Los Angeles	Unconditioned	Low	550.2	555.0	0.9%
Los Angeles	Unconditioned	Med	772.1	780.3	1.1%
Los Angeles	Unconditioned	High	973.8	1050.4	7.9%
Houston	Conditioned	Low	496.3	481.9	-2.9%
Houston	Conditioned	Med	696.9	669.1	-4.0%
Houston	Conditioned	High	877.2	883.6	0.7%
Houston	Unconditioned	Low	492.8	479.6	-2.7%
Houston	Unconditioned	Med	690.8	664.2	-3.8%
Houston	Unconditioned	High	870.6	881.0	1.2%
Phoenix	Conditioned	Low	446.5	425.4	-4.7%
Phoenix	Conditioned	Med	626.9	584.1	-6.8%
Phoenix	Conditioned	High	789.1	767.7	-2.7%
Phoenix	Unconditioned	Low	441.2	421.5	-4.5%
Phoenix	Unconditioned	Med	619.0	585.1	-5.5%
Phoenix	Unconditioned	High	779.8	760.5	-2.5%

Table 81: Net distribution losses (in kWh) from the DHW distribution system for electric storage and HPWHs

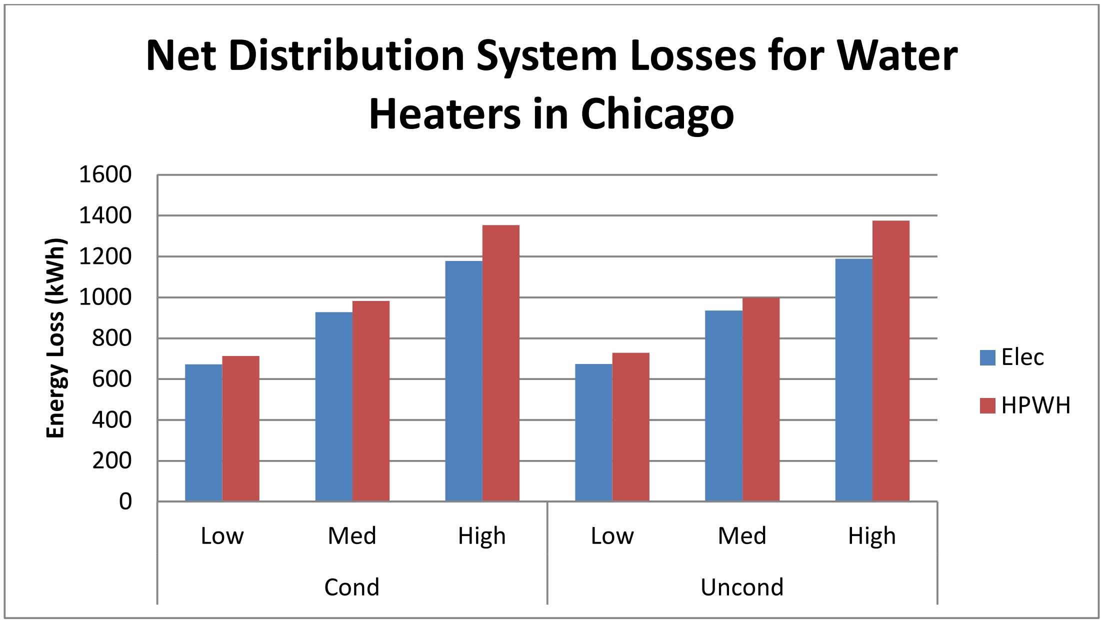


Figure 117: Net distribution losses in Chicago for electric storage and HPWHs

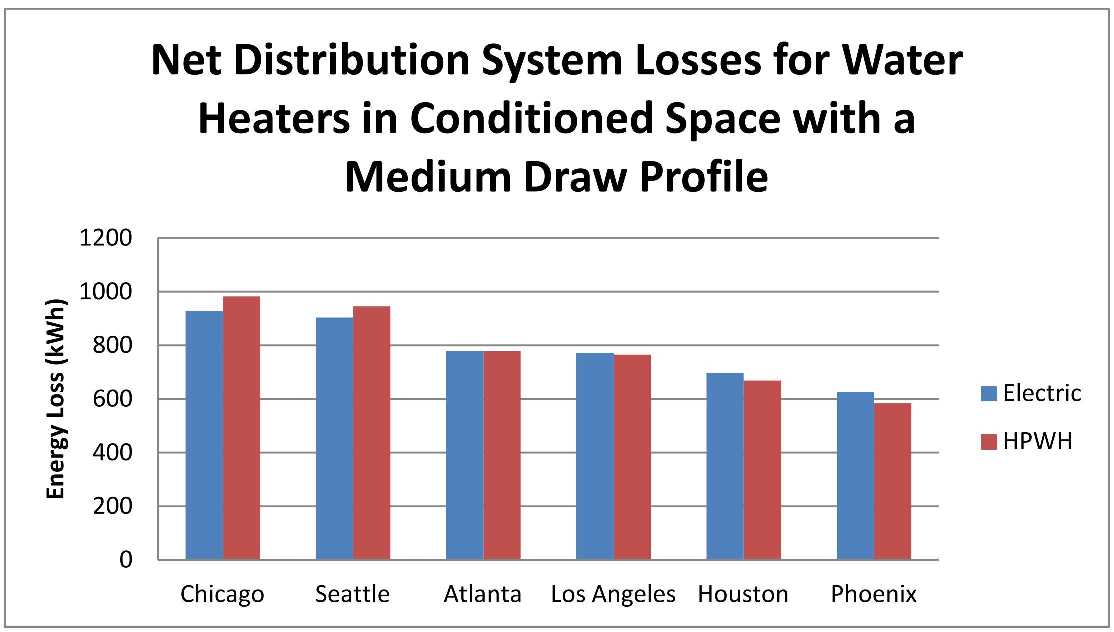


Figure 118: Net distribution losses for electric storage and HPWHs with a medium draw profile

Chapter 9: Conclusions and Future Work

9.1 Water Heater Modeling Conclusions and Future Work

In this work, models for several of the most common residential water heating technologies were presented. While many of these models were preexisting, new models of HPWHs and condensing water heaters were created and validated. The HPWH was extensively validated against both lab and field testing and was able to predict performance well in both situations. The condensing water heater underwent only a limited validation given the lack of information it could be validated against but performed fairly well.

For the HPWH model, although it performed very well in most of the validation studies presented here, there are some opportunities for future work. Icing of the evaporator is not accounted for in the model and was seen to lead to over prediction of energy savings in situations where icing occurs. Ideally, future models will be able to be account for this phenomenon. In addition, the performance maps for this model are based solely on wet bulb temperature instead of considering ambient dry temperature and humidity separately. While it would require extensive lab testing to derive a performance map that examines these factors independently, it could improve the current model. The model created here is also based on only one manufacturer's HPWH (out of the 5 integrated units currently available). In the future, models of these other HPWHs could be created and run through simulations similar to those performed here. This could give further insight into the current crop of HPWHs and could help to inform manufacturers on potential improvements to their products.

The condensing water heater model presented here was based on the very limited amount of data about these units that is available. If further testing of these units is performed, the model can be validated more stringently, leading to better predictions of the performance of this unit. The best case scenario would be to perform testing specifically to derive the necessary parameters as well as subjecting the unit to typical residential water heater draw profiles to provide results to validate against.

Finally, a model of a condensing tankless water heater needs to be created. There was currently not enough data available to derive all the necessary parameters and validate the model, which led to its exclusion in this work. Future lab testing of these units is needed since it is a gas water heater has a high potential for energy savings and could gain significant market penetration.

9.2 Annual Simulations Conclusions and Future Work

Simulations were performed for these units in a variety of climates, in both conditioned and unconditioned space with several realistic draw profiles. From these simulations, it is apparent that solar water heaters are the most energy efficient gas water heating option in almost every scenario, although tankless water heaters did have an edge in low use homes in cooling dominated climates. For electric water heaters, HPWHs were able to provide comparable, if not better, energy savings when compared to a solar water heater under many of the scenarios examined here. In the future, it would be useful to develop and run a tankless condensing water heater model to see how it compares to other gas water heaters.

Economic calculations were also performed for all of these technologies. HPWHs were often the most cost effective water heating option, even before any incentives were considered. This demonstrates their great potential to be significant cost and energy savers in homes that use electricity for water heating. For gas water heaters, there were no scenarios where a more efficient option was able to provide a LCC savings over a standard gas storage water heater without incentives with the installed costs assumed here. However, with the currently available incentives these units can compete with typical gas storage water heaters. Future work could run these simulations in more locations, determining how far in each climate zone these results can be extrapolated.

Works Cited

1. **US Energy Information Agency (EIA).** *Residential Energy Consumption Survey.* Washington, DC : US Department of Energy, 2005.

2. **US Department of Energy.** *Water Heater Market Profile.* s.l. : US Department of Energy, 2009.

3. **Cassard, H., Denholm, P. and Ong, S.** *Break-even Cost for Residential Solar Water Heating in the United States: Key Drivers and Sensitivities.* Golden, CO : National Renewable Energy Laboratory (NREL), 2011.

4. **EPA Office of Air and Radiation, Climate Protection Partnership Division.** *National Awareness of ENERGY STAR for 2008: Analysis of 2008 CEE Household Survey.* 2009.

5. **US Department of Energy.** *Energy Star Qualified Water Heaters-Partner Guide.* s.l. : US Department of Energy, 2009. pp. 1-13.

6. —. *Energy Conservation Program: Energy Conservation Standards for Residential Water Heaters, Direct Heating Equipment, and Pool Heaters; Final Rule.* s.l. : Federal Register, 2010.

7. *Condensing Natural Gas Water Heaters.* **Pescatore, P, Roth, K and Brodrick, J.** February 2004, ASHRAE, pp. 51-52.

8. **Pescatore, Peter.** *Development of a Market Optimized Condensing Gas Water Heater.* s.l. : TIAX, 2006.

9. **Dubay, K., Ayee, G. and Gereffi, G.** *Manufacturing Climate Solutions: Carbon-Reducing Technologies and U.S. Jobs-Chapter 6: Electric Heat Pump Water Heaters.* Durham : Manufacturing Climate Solutions, 2009.

10. **Bohac, D, et al.** *Actual Savings and Performance of Natural Gas Tankless Water Heaters.*

Minneapolis : Minnesota Center for Energy and Environment, 2010.

11. **US Department of Energy.** Energy Saver. *Water Heating.* [Online] Office of Energy Efficiency and Renewable Energy. [Cited: June 15, 2011.] http://www.energysavers.gov/your_home/water_heating/index.cfm/mytopic=12760.

12. **JohnnyOnEnergy.** Water Heaters. *JohnnyOnEnergy.* [Online] [Cited: June 15, 2011.] http://www.johnnyonenergy.com/water-heater-replacement/heat-pump-water-heater/.

13. **Hiller, C.** *Commentary on the Energy Star Water Heating Program.* Washington, DC : Energy Star, 2007.

14. **Benjamin Franklin Plumbers.** Tankless Water Heaters. *Benjamin Franklin Plumbers.* [Online] [Cited: June 15, 2011.] http://www.benfranklinplumbingcharlotte.com/tankless_water_heaters.html.

15. *Economics of residential gas furnaces and water heaters in US new construction market.* **Lekov, A., et al.** 2009, Energy Efficiency Online Publication.

16. **Hudon, K., Merrigan, T., Burch, J.** *Low Cost Solar Water Heating Research and Development Roadmap.* Golden, CO : NREL, 2011.

17. *A Review of Heat Pump Water Heating Systems.* **Hepbasli, Arif and Kalinci, Yildiz.** 2009, Renewable and Sustainable Energy Reviews, pp. 1211-1229.

18. **Hashimoto, K.** *Technology and Market Development of CO2 Heat Pump Water Heaters (ECO CUTE) in Japan.* s.l. : IEA Heat Pump Centre Newsletter, 2006.

19. **Rheem.** Hybrid Water Heater Efficiency Zones. *Hybrid Water Heaters.* [Online] [Cited: June 15, 2011.] http://www.rheem.com/Products/tank_water_heaters/Hybrid_electric/efficiency.

20. **Butti, Ken and Perlin, John.** *A Golden Thread: 2500 Years of Solar Architecture and Technology.* Palo Alto : Van Nostrand Reinhold Company, 1980.

21. **Ramlow, Bob and Nusz, Benjamin.** *Solar Water Heating: A Comprehensive Guide to Solar Water Heating and Space Heating Systems.* Gabriola Island : New Society Publishers, 2006.

22. **Goswami, Y., Kreith, F. and Kreider, J.** *Principles of Solar Engineering.* New York : Taylor & Francis Group, 2000.

23. **ASHRAE.** *ASHRAE Handbook-Heating, Ventilating, and Air-Conditioning Systems and Equipment.* s.l. : ASHRAE, 2008. pp. 31.1-31.7.

24. **Siegenthaler, John.** Active Solar Heating: New Opportunities for Hydronics Pros (Webinar). May 2008.

25. **Weiss, W., Bergmann, I. and Faninger, G.** *Solar Heat Worldwide: Markets and Contribution to the Energy Supply 2006.* Gleisdorf : IEA, 2008.

26. **Solar Rating and Certification Corporation.** *Methodology for Determining the Thermal Performance Rating for Solar Collectors.* Cocoa : SRCC, 1994.

27. *TRNSYS 16: A TRaNsient System Simulation Program, User Manual.* **Klein, S et al.** Madison, WI : Solar Energy Laboratory, University of Wisconsin, 2004.

28. **Thermal Energy System Specialists (TESS).** *Type 534: Vertical Cylinder Water Heater Documentation.* Madison, WI : TESS, 2004.

29. **Thermal Energy System Specialists (TESS)**. *Type 539: Solar Collector with Capacitance Effects and Controls to Maintain a Desired Outlet Temperature.* Madison, WI : TESS, 2010.

30. **Grant, Peter.** Evaluation of the Performance of Gas Tankless Water Heaters. Golden, CO : Master's Thesis, 2010.

31. *Using Ratings Data to Derive Simulation-Model Inputs for Storage-Tank Water Heaters.* **Burch, Jay and Erikson, Paul.** Portland, Oregon : s.n., 2004. Poceedings of the 29th Passive Solar Conference. pp. 393-398.

32. **US Department of Energy.** *10 CFR Part 430 Energy Conservation Program for Consumer Products: Test Procedure for Water Heaters.* s.l. : Federal Register Vol. 63, No. 90, 1998.

33. **Department of Energy.** *Residential Water Heater Final Rule Making Technical Support Document.* Washington, DC : Departmet of Energy , 2001.

34. **Healy, W.** Effect of Temperature Stratification Near Heating Elements on the Measured Energy Factors of Electric Water Heaters. *ASHRAE Transactions.* 2008, Vol. 114, 2.

35. *Deriving Simulation Parameters from Rating Data for Stratified Storage Tank Water Heaters.* **Burch, J.** s.l. : Proceedings of the 2010 Solar Conference, 2010.

36. **Grant, Peter.** Sub timestep issues with Type 201. *Personal Communication.* July 13, 2011.

37. **Solar Rating and Certification Corporation (SRCC).** *OG 300: Operating Guidelines and Minimum Standards for Certifying Solar Water Heating Systems.* Cocoa, FL : SRCC, 2010.

38. **Merrigan, T.** *Personal Communications on Solar Water Heater Sizes in the US.* October 2011.

39. —. *Personal Communications on Solar Water Heater Collector Sizing.* October 12, 2011.

40. **Hendron, R., Hancock, E. and Reeves, P.** *Building America Field Test and Analysis Report: CORE H1 and H2 Houses - Carbondale, CO.* Golden, CO : National Renewable Energy Laboratory, 2005.

41. **Hudon, K., et al.** Heat Pump Water Heater Technology Assessment Based on Laboratory Research and Energy Simulation Models. *ASHRAE Transactions.* 2011.

42. **Thermal Energy System Specialists (TESS).** *Documentation for Heat Pump Water Heater Type 938.* Madison, WI : TESS.

43. **Puttagunta, Srikanth.** *Measured Performance of Advanced Water Heating Strategies: Heat Pump Water Heaters.* Norwalk, CT : Stephen Winter Associates, 2011.

44. **Thermal Energy System Specialists (TESS).** TRNSYS-TESS Libraries. *TESS Library General Descriptions.* [Online] [Cited: July 20, 2011.] http://www.trnsys.com/tess-libraries/TESSLibs17_General_Descriptions.pdf.

45. **Heat Transfer Products.** Heat Transfer Products: Phoenix Water Heater. [Online] [Cited: October 22, 2011.] http://www.htproducts.com/literature/lp-191i.pdf.

46. **ANSI.** *Method of Test for Measuring Thermal Efficiency.* s.l. : ANSI, 2004. ANSI Z.21.10.3.

47. **PG&E Apllied Technology Services.** *Laboratory Testing of Residential Gas Water Heaters.* San Ramon, CA : PG&E, 2008.

48. **Incropera, F., Dewitt, D., Bergman, T., and Lavine, A.** *Introduction to Heat Transfer.* Hoboken, NJ : John Wiley and Sons, 2007.

49. *Correlating Equations for Laminar and Turbulent Free Convection from a Horizontal Cylinder.* **Churchill, S. and Chu, H.** s.l. : International Journal of Heat and Mass Transfer, 1975.

50. **Applied Energy Technology.** *Hot Water Distribution System Research - Phase 1.* Davis, CA : California Energy Commission Public Interest Energy Research Program, 2005.

51. **Hendron, R. and Engelbrecht, C.** *Building America House Simulation Protocol.* Golden, CO : National Renewable Energy Laboratory, 2010.

52. **Davis Energy Group.** *Prototype Floor Plans: Hot Water Distribution System Layouts.* Davis, CA : Davis Energy Group, 2006.

53. *Development of Standardized Domestic Hot Water Event Schedules for Residential Buildings.* **Hendron, R. and Burch, J.** Long Beach : National Renewable Energy Lab, 2007. Energy Sustainability. pp. 1-8.

54. *HWSIM: Development and Validation of a Residential Hot Water Distribution System Model.* **Springer, D., Rainer, L., Hoeschele, M., and Scott, R.** Pacific Grove, CA : ACEEE Summer

Study of Energy Efficiency in Buildings, 2008.

55. **Klein, S. et al.** *Multizone Building Modeling with Type 56 and TRNBUILD.* Madison, WI : TNRSYS, 2010.

56. **PNNL & ORNL.** *Building America Best Practice Series-Guid to Determining Climate Regions by County.* s.l. : US Department of Energy, 2007.

57. **ICC.** *International Energy Conservation Code.* Falls Church, VA : International Code Council, 2009.

58. **ASHRAE.** *ASHRAE Handbook: Fundamentals.* s.l. : ASHRAE, 1989.

59. **Oak Ridge National Laboratory (ORNL).** *1988.* Oak Ridge, Tennessee : ORNL, Building Foundation Design Handbook.

60. **Winklemann, F.** *Underground Surfaces: How to get Better Underground Surface Heat Transfer Calculations in DOE2.1E.* Berkeley, CA : LBNL, 2003.

61. **ASHRAE.** *Ventilation and Acceptable Indoor Air Quality in Low Rise Residential Buiildings.* Atlanta, GA : ASHRAE, 2007. 62.2.

62. **Casey, R.** *Performance Modeling of a Building Integrated Photovoltaic/Thermal Air Collector (Thesis).* Boulder, CO : CU Boulder, 2009.

63. **Wilcox, S. and Marion, W.** *Users Manual for TMY3 Data Sets.* Golden, CO : NREL, 2008.

64. **Solar Ratings and Certification Corporation (SRCC).** Solar Facts - System Ratings. *SRCC.* [Online] [Cited: October 31, 2011.] http://www.solar-rating.org/facts/system_ratings.html.

65. **NREL GIS.** Solar Radiation Map. *NREL GIS.* [Online] NREL, May 20, 2011. [Cited: October 25, 2011.] http://www.nrel.gov/gis/solar.html.

66. **US Department of Energy (DOE.** DSIRE Database of State Incentives for Renewables and

Efficiency. *DSIRE.* [Online] [Cited: November 10, 2011.] http://www.dsireusa.org/.

67. **Fuller, S., Peterson, S.** *Life-Cycle Costing Manual for the Federal Energy Management Program.* Gaithersburg, MD : National Institute of Standards and Technology (NIST), 1995.

68. **Rushing, A., Kneifel, J., and Lippiatt, B.** *Annual Supplement to NIST Handbook 135 and NBS Special Publication 709.* Washington, DC : US Department of Commerce, 2010.

69. **US Department of Energy (DOE).** *Residential Heating Products final Rule.* Washington, DC : US DOE, 2010. 75 FR 21981.

70. **Lawrence Berkeley National Laboratory.** *Therm 6.3/Window 6.3 NFRC Simulation Manual.* Berkeley, CA : LBNL, 2011.

71. **Thermal Energy System Specialists (TESS).** *Documentation for Type 607: Air Duct.* Madison, WI : TESS, 2011.

72. **Lawrence Berkeley National Laboratory (LBNL).** *EnergyPlus Engineering Reference V6.0.* Berkeley, CA : LBNL, 2010.

73. **Wisconsin Public Service.** Natural Gas Appliance Calculator. *Wisconsin Public Service.* [Online] 2012. [Cited: January 30, 2012.]

74. **US Department of Energy; US Environmental Protection Agency.** Energy Star. *Energy Star Products.* [Online] 2010. [Cited: May 10, 2010.] http://www.energystar.gov/index.cfm?c=products.pr_find_es_products.

75. **Building Science.** *Combo Space/Water Heating Systems - "Duo Diligence".* Somerville : Building Science Publishing, 2002.

76. *Optimizing the Performance of Radiant Heating Systems.* **D'Antonio, P.** 8, August 2006, HPAC Engineering, Vol. 78, pp. 24-33.

77. *Maximizing Small-Boiler Efficiency.* **D'Antonio, P.** 8, August 2006, HPAC Engineering, Vol.

78, pp. 50-51.

78. **Butcher, T.** *Hydronic Baseboard Thermal Distribution Sustem with Outdoor Reset Control to Enable the Use of a Condensing Boiler.* Upton : Brookhaven National Lab, 2004.

79. *Annual Energy Outllok 2010 Early Release.* **Department of Energy.** s.l. : Department of Energy, 2009. pp. 1-3.

80. **US Department Of Energy.** *Energy Star Residential Water Heater Criteria: Final Analysis.* s.l. : US Department of Energy, 2008.

81. *Evaluation of a High-Performance Multifamily Residential Building in Aspen, Colorado.* **Hendron, R, et al.** San Francisco, CO : s.n., 2009. Proceedings of the 3rd International Conference on Energy Sustainability. pp. 1-12.

82. *Maximization of Primary Energy Savings of Solar Heating and Cooling Systems by Transient Simulation and Design of Experiments.* **Calise, F., Palombo, A. and Vanoli, L.** 2010, Applied Energy, pp. 524-540.

83. *From Model Validation to Production of Reference Simulations: How to Increase Reliability and Applicability of Buildings and HVAC Simulation Models.* **Andre, P., et al.** 2008, Buiding Serv. Eng. Res. Technol., pp. 61-72.

84. *Temperature Stratification from Thermal Diodes in Solar Hot Water Storage Tank.* **Rhee, J, et al.** 2009, Solar Energy, pp. 507-511.

85. **US Department of Energy.** Annual Energy Outlook 2010 Early Release. [Online] 2010. http://www.eia.doe.gov/oiaf/aeo/.

86. *Seasonal Performance Rating of Heat Pump Water Heaters.* **Morrison, G., Anderson, T. and Behnia, M.** 2004, Solar Energy, pp. 147-152.

87. **Hoffman, J., Wells, J. and Guiney, W.** *Transforming the Market for Solar Water Heaters: A*

New Model to Build a Permanant Sales Force. s.l. : Renewable Energy Policy Product, 1998.

88. *Cost Effectiveness of a Photovoltaic-Powered Heat Pump Water Heating System vs. Solar Thermal Water Heating.* **Wilson, E. and Burkhardt, J.** San Francisco : ASME, 2009. 3rd International Conference of Energy Sustainability. pp. 293-300.

89. *Thermal Stratification Within the Water Tank.* **Han, Y., Wang, R. and Dai, Y.** 2008, Renewable and Sustainable Energy Reviews, pp. 1015-1023.

90. *System Optimization and Experimental Research on Air Source Heat Pump Water Heaters.* **Zhang, J., Wang, R. and Wu, J.** 2007, Applied Thermal Engineering, pp. 1029-1035.

91. **Magnusson, Lee.** *Solar Row Case Study-Results of One Year of Long-Term Monitoring.* Golden : National Renewable Energy Laboratory, 2009.

92. **Grossman, G., et al.** *Solar Energy for the Production of Heat-Summary and Recommendations of the 4th Assembly of rhe Energy Forum at SNI.* s.l. : Samuel Neaman Institute, 2007.

93. *Solar Water Heaters in China: A New Day Dawning.* **Han, J., Mol, A. and Lu, Y.** 2010, Energy Policy, pp. 383-391.

94. *Sensitivity of Long-Term Performance Simulations of Solar Energy Systems to the Degree of Stratification in the Thermal Storage Unit.* **Arias, D., McMahon, A. and Klein, S.** 2007, International Journal of Energy Research, pp. 242-254.

95. **Baxter, V., et al.** *Residential Heat Pump Water Heater Development Status-USA.* Oak Rigde : Oak Ridge National Laboratory, 2005.

96. **Baxter, V,, et al.** *Residential Heat Pump Water Heater Development Status-USA.* Oak Rigde : Oak Ridge National Laboratory, 2005.

97. **US Energy Information Agency.** *Annual Energy Review.* Washington DC : US Department

of Energy, 2009.

98. **National Association of Home Builders.** *National Green Building Standard.* Washington, DC : s.n., 2008.

99. **US Green Building Council.** *LEED for Homes Rating System.* Washington, DC : s.n., 2008.

100. —. LEED: What LEED Is. *United States Green Building Council.* [Online] 2011.

101. **National Association of Home Builder.** About NAHB. *National Association of Home Builders.* [Online] 2011 . [Cited: 2011 20-January.] http://www.nahb.org/page.aspx/landing/sectionID=5#.

102. **NAHB Research Center.** *Green Home Building Rating Systems-A Sample Comparison.* Uppper Marlboro : NAHB Research Center, 2008.

103. **ENERGY STAR.** Independent Inspection and Testing. *ENERGY STAR.* [Online] 2011. [Cited: 2011 23-January.] http://www.energystar.gov/ia/new_homes/features/HERSrater_062906.pdf.

104. —. ENERGY STAR. *History: ENERGY STAR.* [Online] 2011. [Cited: 2011 23-January.] http://www.energystar.gov/index.cfm?c=about.ab_history.

105. —. National Program Requirements, Version 3.0. *ENERGY STAR.* [Online] 2011 . [Cited: 2011 23-January.] http://www.energystar.gov/ia/partners/bldrs_lenders_raters/downloads/NationalProgramRequirements.pdf.

106. **AIA Cincinnati.** Comparison of LEED for Homes and Nation Green Building Standard. *AIA Cincinnati.* [Online] 2010. [Cited: 2001 23-January.] http://www.aiacincinnati.org/community/LEED_NAHB_Sum.cfm.

107. **D Bohac, B Schoenbauer, M Hewett, M Lobenstein, T Butcher.** *Actual Savings and*

Performance of Natural Gas Tankless Water Heaters. Minneapolis : Center for Energy and Environment, 2010.

108. **IMAC Systems, Inc.** IMAC Systems. *American Meter AC-250 Diaphragm Gas Meter.* [Online] [Cited: March 31, 2011.] http://www.imacsystems.com/ac250.htm.

109. **National Renewable Energy Laboratory.** National Residential Efficiency Measures Database. *Tank Water Heaters.* [Online] October 2010. [Cited: May 09, 2011.] http://www.nrel.gov/ap/retrofits/measures.cfm?gId=6&ctId=40&scId=493.

110. **US Energy Information Agency (EIA).** *Electric Sales, Revenue, and Average Price 2008.* Washington,DC : EIA, 2010.

111. **Garrabrant, M.** Gas-Fired Heat Pump Water Heater for Residential Applications. *Presentation at the ACEEE Hot Water Forums.* Berkeley, CA : s.n., 2011.

112. **Ramlow, B. and Nusz, B.** *Solar Water Heating: A Comprehensive Guide to Solar Water Heating and Space Heating Systems.* Gabriola Islande : New Society Publishers, 2010.

113. **Pacific Northwest National Laboratory (PNNL) and Oak Ridge National Laboratory (ORNL).** *Building America Best Practice Series: Your Guide to Determining Climate Zone by County.* Washington DC : US Department of Energy, 2010. PNNL-17211.

114. **Puttagunta, Srikanth.** HPWH discrepencies at Site 13. *Personal Communication.* October 25, 2011.

Appendix A: FORTRAN Code for HPWH model (Type 994)

```
      SUBROUTINE TYPE994(TIME,XIN,OUT,T,DTDT,PAR,INFO,ICNTRL,*)

C---------------------------------------------------------------------------
-------------------------------------------
C    DESCRIPTION:
C    THIS COMPONENT IS INTENDED TO MODEL A HEAT PUMP WATER HEATER.  IT IS
ASSUMED THAT THE CATALOG DATA DOES
C    NOT ACCOUNT FOR THE ADDITIONAL HEAT INPUT BY THE FAN; BOTH IN THE POWER
REPORTED AND THE IMPACT ON THE
C    NET CAPACITY. THIS VERSION OF THE HEAT PUMP MODEL TAKES NORMALIZED
CAPACITY AND POWER AS PARAMETERS.
C
C    THIS SUBROUTINE WAS WRITTEN BY JEFF THORNTON OF THERMAL ENERGY SYSTEM
SPECIALISTS
C---------------------------------------------------------------------------
-------------------------------------------
! Copyright © 2005 Thermal Energy System Specialists, LLC. All rights
reserved.

C---------------------------------------------------------------------------
-------------------------------------------
      !Export this subroutine for its use in external DLLs.
      !DEC$ATTRIBUTES DLLEXPORT :: TYPE994
C---------------------------------------------------------------------------
-------------------------------------------

C---------------------------------------------------------------------------
-------------------------------------------
C    ACCESS TRNSYS FUNCTIONS
      USE TrnsysConstants
      USE TrnsysFunctions
C---------------------------------------------------------------------------
-------------------------------------------

C---------------------------------------------------------------------------
-------------------------------------------
C    TRNSYS DECLARATIONS
      IMPLICIT NONE
      DOUBLE PRECISION XIN,OUT,TIME,PAR,T,DTDT,TIME0,TFINAL,DELT

      INTEGER*4 INFO(15),NP,NI,NOUT,ND,IUNIT,ITYPE,ICNTRL
      CHARACTER*3 YCHECK,OCHECK
C---------------------------------------------------------------------------
-------------------------------------------

C---------------------------------------------------------------------------
-------------------------------------------
C    USER DECLARATIONS
```

```
      PARAMETER (NP=13,NI=8,NOUT=18,ND=0)
C-----------------------------------------------------------------------------
-------------------------------------------

C-----------------------------------------------------------------------------
-------------------------------------------
C    REQUIRED TRNSYS DIMENSIONS
      DIMENSION XIN(NI),OUT(NOUT),PAR(NP),YCHECK(NI),OCHECK(NOUT)
C-----------------------------------------------------------------------------
-------------------------------------------

C-----------------------------------------------------------------------------
-------------------------------------------
C    DECLARATIONS AND DEFINITIONS FOR THE USER-VARIABLES
      INTEGER LU_DATA,N_WT,N_WB,N_DB,NX,NVAL(4),NY,STATUS,MODE,
      1   PSYCHMODE
      DOUBLE PRECISION PSYDAT(9),FLOW_FLUID,RHO_FLUID,CP_FLUID,CP_AIR,
      1   DP_AIR,T_K,BLOWPOW,CONTPOW,LPS_TOT,T_FLUID_IN,Q_EVAP,RH_AIR_IN,
     1   POWER_BLOWER,P_KPA,AIRPROPS(5),P_AIR_OUT,P_AIR_IN,POWER_CONT,
     1   Q_COND,FLOW_AIR,T_FLUID_OUT,T_AIR_OUT,W_AIR_OUT,RH_AIR_OUT,
     1   Q_TOT,Q_LAT,Q_SENS,REJECTED,POWER_COMPRESSOR,COP,EER,X(3),
     1   H_AIR_IN,W_AIR_IN,T_AIR_IN,WB_AIR_IN,RHO_AIR_DRY_IN,H_AIR_OUT,
     1   Y(5),ONSIG,POWER,RATED_TOT,RATED_SENS,RATED_POW,RATED_COND,
     1   Q_ENV, RATED_LPS_TOT, WB_AIR_OUT
      LOGICAL DEVICE_ON
      CHARACTER (LEN=maxMessageLength) MESSAGE1
C-----------------------------------------------------------------------------
-------------------------------------------

C-----------------------------------------------------------------------------
-------------------------------------------
C    ERROR MESSAGES
      MESSAGE1='The heat pump model was unable to correctly read from th
      1e supplied heat pump data files.  Please check the location and fo
     1rmat of the required data files and re-run the simulation.'
C-----------------------------------------------------------------------------
-------------------------------------------

C-----------------------------------------------------------------------------
-------------------------------------------
C    GET GLOBAL TRNSYS SIMULATION VARIABLES
      TIME0=getSimulationStartTime()
      TFINAL=getSimulationStopTime()
      DELT=getSimulationTimeStep()
C-----------------------------------------------------------------------------
-------------------------------------------

C-----------------------------------------------------------------------------
-------------------------------------------
C    SET THE VERSION INFORMATION FOR TRNSYS
      IF(INFO(7).EQ.-2) THEN
         INFO(12)=16
         RETURN 1
      ENDIF
```

```
C-----------------------------------------------------------------------------
-------------------------------------------

C-----------------------------------------------------------------------------
-------------------------------------------
C    DO ALL THE VERY LAST CALL OF THE SIMULATION MANIPULATIONS HERE
      IF (INFO(8).EQ.-1) THEN
         RETURN 1
      ENDIF
C-----------------------------------------------------------------------------
-------------------------------------------

C-----------------------------------------------------------------------------
-------------------------------------------
C    PERFORM ANY "AFTER-ITERATION" MANIPULATIONS THAT ARE REQUIRED
      IF(INFO(13).GT.0) THEN
         RETURN 1
      ENDIF
C-----------------------------------------------------------------------------
-------------------------------------------

C-----------------------------------------------------------------------------
-------------------------------------------
C    DO ALL THE VERY FIRST CALL OF THE SIMULATION MANIPULATIONS HERE
      IF (INFO(7).EQ.-1) THEN

C       RETRIEVE THE UNIT NUMBER AND TYPE NUMBER FOR THIS COMPONENT FROM THE
INFO ARRAY
         IUNIT=INFO(1)
         ITYPE=INFO(2)

C       SET SOME INFO ARRAY VARIABLES TO TELL THE TRNSYS ENGINE HOW THIS TYPE
IS TO WORK
         INFO(6)=NOUT
         INFO(9)=1
         INFO(10)=0

C       CALL THE TYPE CHECK SUBROUTINE TO COMPARE WHAT THIS COMPONENT
REQUIRES TO WHAT IS SUPPLIED
         CALL TYPECK(1,INFO,NI,NP,ND)

C       SET THE YCHECK AND OCHECK ARRAYS TO CONTAIN THE CORRECT VARIABLE
TYPES FOR THE INPUTS AND OUTPUTS
         DATA YCHECK /'TE1','MF1','TE1','DM1','PC1','MF1','PR4','DM1'/
         DATA OCHECK /'TE1','MF1','TE1','DM1','PC1','MF1','PR4','PW1',
      1               'PW1','PW1','PW1','PW1','PW1','PW1','DM1','DM1',
     1               'TE1','MF1'/

C       CALL THE RCHECK SUBROUTINE TO SET THE CORRECT INPUT AND OUTPUT TYPES
FOR THIS COMPONENT
         CALL RCHECK(INFO,YCHECK,OCHECK)

C       RETURN TO THE CALLING PROGRAM
         RETURN 1
```

```
      ENDIF
C-----------------------------------------------------------------------------
-------------------------------------------

C-----------------------------------------------------------------------------
-------------------------------------------
C    DO ALL OF THE INITIAL TIMESTEP MANIPULATIONS HERE - THERE ARE NO
ITERATIONS AT THE INTIAL TIME
      IF (TIME.LT.(TIME0+DELT/2.D0)) THEN

C       SET THE UNIT NUMBER FOR FUTURE CALLS
         IUNIT=INFO(1)

C       READ IN THE VALUES OF THE PARAMETERS IN SEQUENTIAL ORDER
         MODE=JFIX(PAR(1)+0.1)
         LU_DATA=JFIX(PAR(2)+0.1)
         N_WT=JFIX(PAR(3)+0.1)
         N_WB=JFIX(PAR(4)+0.1)
         N_DB=JFIX(PAR(5)+0.1)
         RHO_FLUID=PAR(6)             !KG/M^3
         CP_FLUID=PAR(7)              !KJ/KG.K
C        BLOWPOW=PAR(8)              !KJ/HR
         CONTPOW=PAR(8)               !KJ/HR
C        LPS_TOT=PAR(10)             !L/S
         RATED_TOT=PAR(9)             !KJ/H
         RATED_SENS=PAR(10)           !KJ/H
         RATED_POW=PAR(11)            !KJ/H
         RATED_COND=PAR(12)           !KJ/H
         RATED_LPS_TOT=PAR(13)        !KJ/H

C       CHECK THE PARAMETERS FOR PROBLEMS AND RETURN FROM THE SUBROUTINE IF
AN ERROR IS FOUND
         IF((MODE.LT.1).OR.(MODE.GT.2)) CALL TYPECK(4,INFO,0,1,0)
         IF(LU_DATA.LT.10) CALL TYPECK(4,INFO,0,2,0)
         IF(N_WT.LT.1) CALL TYPECK(4,INFO,0,3,0)
         IF(N_WB.LT.1) CALL TYPECK(4,INFO,0,4,0)
         IF(N_DB.LT.1) CALL TYPECK(4,INFO,0,5,0)
         IF(RHO_FLUID.LT.0.) CALL TYPECK(4,INFO,0,6,0)
         IF(CP_FLUID.LT.0.) CALL TYPECK(4,INFO,0,7,0)
C         IF(BLOWPOW.LT.0.) CALL TYPECK(4,INFO,0,8,0)
         IF(CONTPOW.LT.0.) CALL TYPECK(4,INFO,0,9,0)
C        IF(LPS_TOT.LE.0.) CALL TYPECK(-4,INFO,0,10,0)
         IF(RATED_TOT.LE.0.) CALL TYPECK(-4,INFO,0,11,0)
         IF(RATED_SENS.LE.0.) CALL TYPECK(-4,INFO,0,12,0)
         IF(RATED_POW.LE.0.) CALL TYPECK(-4,INFO,0,13,0)
         IF(RATED_COND.LE.0.) CALL TYPECK(-4,INFO,0,14,0)

C       PERFORM ANY REQUIRED CALCULATIONS TO SET THE INITIAL VALUES OF THE
OUTPUTS HERE
         OUT(1)=XIN(1)
         OUT(2)=0.
         OUT(3)=XIN(3)
         OUT(4)=XIN(4)
         OUT(5)=XIN(5)
         OUT(6)=0.
```

```
          OUT(7)=XIN(7)
          OUT(8:16)=0.
          OUT(17)=XIN(3)
          OUT(18)=0.

C        RETURN TO THE CALLING PROGRAM
          RETURN 1

      ENDIF
C---------------------------------------------------------------------------
-------------------------------------------

C---------------------------------------------------------------------------
-------------------------------------------
C    *** ITS AN ITERATIVE CALL TO THIS COMPONENT ***
C---------------------------------------------------------------------------
-------------------------------------------

C---------------------------------------------------------------------------
-------------------------------------------
C    RE-READ THE PARAMETERS IF ANOTHER UNIT OF THIS TYPE HAS BEEN CALLED
      IF(INFO(1).NE.IUNIT) THEN

C        RESET THE UNIT NUMBER
          IUNIT=INFO(1)
          ITYPE=INFO(2)

          MODE=JFIX(PAR(1)+0.1)
          LU_DATA=JFIX(PAR(2)+0.1)
          N_WT=JFIX(PAR(3)+0.1)
          N_WB=JFIX(PAR(4)+0.1)
          N_DB=JFIX(PAR(5)+0.1)
          RHO_FLUID=PAR(6)              !KG/M^3
          CP_FLUID=PAR(7)               !KJ/KG.K
C          BLOWPOW=PAR(8)               !KJ/HR
          CONTPOW=PAR(8)                !KJ/HR
C          LPS_TOT=PAR(10)              !L/S
          RATED_TOT=PAR(9)              !KJ/H
          RATED_SENS=PAR(10)            !KJ/H
          RATED_POW=PAR(11)             !KJ/H
          RATED_COND=PAR(12)            !KJ/H
          RATED_LPS_TOT=PAR(13)         !KJ/H

      ENDIF
C---------------------------------------------------------------------------
-------------------------------------------

C---------------------------------------------------------------------------
-------------------------------------------
C    RETRIEVE THE CURRENT VALUES OF THE INPUTS TO THIS MODEL FROM THE XIN
ARRAY IN SEQUENTIAL ORDER

      T_FLUID_IN=XIN(1)              !C
      FLOW_FLUID=XIN(2)              !KG/HR
      T_AIR_IN=XIN(3)           !C
```

```
      W_AIR_IN=XIN(4)          !%
      WB_AIR_IN=XIN(5)         !C
      P_AIR_IN=XIN(6)          !ATM
      DP_AIR=XIN(7)            !ATM
      ONSIG=XIN(8)


C     CHECK THE INPUTS FOR PROBLEMS
      IF(FLOW_FLUID.LT.0.) CALL TYPECK(-3,INFO,2,0,0)
      IF(W_AIR_IN.LT.0.) CALL TYPECK(-3,INFO,4,0,0)
      IF(W_AIR_IN.GT.1.) CALL TYPECK(-3,INFO,4,0,0)
      IF(P_AIR_IN.LT.0.) CALL TYPECK(-3,INFO,6,0,0)
      IF(DP_AIR.LT.0.) CALL TYPECK(-3,INFO,7,0,0)
      IF(DP_AIR.GT.P_AIR_IN) CALL TYPECK(-3,INFO,7,0,0)
C      IF(BLOWPOW.LT.0.) CALL TYPECK(4,INFO,0,8,0)
C      IF(LPS_TOT.LE.0.) CALL TYPECK(-4,INFO,0,10,0)

      IF(ErrorFound()) RETURN 1
C-----------------------------------------------------------------------------
-------------------------------------------

C-----------------------------------------------------------------------------
-------------------------------------------
C     PERFORM ALL THE CALCULATION HERE FOR THIS MODEL.

C     DETERMINE IF THE HEAT PUMP IS OPERATING IN COOLING MODE
      IF(ONSIG.GE.0.5) THEN
         ONSIG=1.
         DEVICE_ON=.TRUE.
      ELSE
         ONSIG=0.
         DEVICE_ON=.FALSE.
      ENDIF

C     SET THE CORRECT INLET STATE
      IF(MODE.EQ.1) THEN
         PSYCHMODE=1
      ELSE
         PSYCHMODE=1
      ENDIF

C     CALL THE PSYCH ROUTINE TO DETERMINE THE PROPERTIES OF THE RETURN AIR
      PSYDAT(1)=P_AIR_IN
      PSYDAT(2)=T_AIR_IN
      PSYDAT(3)=WB_AIR_IN
      PSYDAT(6)=W_AIR_IN
      CALL PSYCHROMETRICS(TIME,INFO,1,PSYCHMODE,0,PSYDAT,2,STATUS,*10)
      CALL LINKCK('TYPE 994','PSYCHROMETRICS',1,994)
10    IF(ErrorFound()) RETURN 1
      P_AIR_IN=PSYDAT(1)
      T_AIR_IN=PSYDAT(2)
      WB_AIR_IN=PSYDAT(3)
      RH_AIR_IN=PSYDAT(4)
      W_AIR_IN=PSYDAT(6)
```

```
      H_AIR_IN=PSYDAT(7)
      RHO_AIR_DRY_IN=PSYDAT(9)                        !KG DRY AIR/M^3

C     CHECK FOR NO-FLOW CONDITIONS
      IF((ONSIG.LT.0.5).OR.(FLOW_FLUID.LE.0.)) THEN

         T_FLUID_OUT=T_FLUID_IN

         P_AIR_OUT=P_AIR_IN-DP_AIR
         T_AIR_OUT=T_AIR_IN
         W_AIR_OUT=W_AIR_IN
         WB_AIR_OUT=WB_AIR_IN
         H_AIR_OUT=H_AIR_IN
         FLOW_AIR=0.0

         Q_TOT=0.
         Q_SENS=0.
         Q_LAT=0.
         REJECTED=0.
         Q_EVAP=0.
         Q_COND=0.
         Q_ENV=0.

         POWER_COMPRESSOR=0.
         POWER_BLOWER=0.
         POWER_CONT=0.
         COP=0.
         EER=0.

         CP_AIR=1.007

      ELSE

C        CALCULATE THE SPECIFIC HEAT OF THE AIR STREAM
         T_K=T_AIR_IN+273.15
         P_KPA=P_AIR_IN*101.325
         CALL AIRPROP(T_K,P_KPA,AIRPROPS)
         CP_AIR=AIRPROPS(5)

C        GET THE RATED CATALOG PERFORMANCE
         NX=3
         NVAL(3)=N_WT
         NVAL(2)=N_WB
         NVAL(1)=N_DB
         NY=5
         X(3)=T_FLUID_IN
         X(2)=WB_AIR_IN
         X(1)=T_AIR_IN
         CALL DYNAMICDATA(LU_DATA,NX,NVAL,NY,X,Y,INFO,*40)
         CALL LINKCK('TYPE 994','DYNAMICDATA',1,994)
40       IF(ErrorFound()) THEN
            CALL MESSAGES(-1,MESSAGE1,'FATAL',IUNIT,ITYPE)
            RETURN 1
         ENDIF
         Q_TOT=RATED_TOT*Y(1)              !KJ/H
```

```
         Q_SENS=RATED_SENS*Y(2)              !KJ/H
         POWER=RATED_POW*Y(3)                !KJ/H
         REJECTED=RATED_COND*Y(4)            !KJ/H
         LPS_TOT=RATED_LPS_TOT*Y(5)          !L/S
C       CALCULATE THE ENERGY THAT IS LOST FROM THE COMPRESSOR TO THE
SURROUNDINGS
         Q_ENV=Q_TOT+POWER-REJECTED

C       CALCULATE THE FAN POWER FROM A CURVE FIT OF FAN VELOCITY
        BLOWPOW=0.007234*LPS_TOT*LPS_TOT+0.2299*LPS_TOT-2.807

C       DETERMINE THE DRY AIR MASS FLOW RATE
         FLOW_AIR=LPS_TOT*3.6*RHO_AIR_DRY_IN               !KG/H

C       CALCULATE THE OUTLET AIR CONDITIONS FROM THE CAPACITIES
         T_AIR_OUT=T_AIR_IN-Q_SENS/CP_AIR/FLOW_AIR
         H_AIR_OUT=H_AIR_IN-Q_TOT/FLOW_AIR
         P_AIR_OUT=P_AIR_IN+DP_AIR

         IF(P_AIR_OUT.LE.0.) THEN
            CALL TYPECK(-3,INFO,29,0,0)
            RETURN 1
         ENDIF

C       CALL PSYCH TO GET THE REMAINING PROPERTIES
         PSYDAT(1)=P_AIR_OUT
         PSYDAT(2)=T_AIR_OUT
         PSYDAT(7)=H_AIR_OUT
         CALL PSYCHROMETRICS(TIME,INFO,1,5,0,PSYDAT,0,STATUS,*60)
60       IF(ErrorFound()) RETURN 1
         T_AIR_OUT=PSYDAT(2)
         W_AIR_OUT=PSYDAT(6)
         RH_AIR_OUT=PSYDAT(4)
         H_AIR_OUT=PSYDAT(7)

C       CHECK TO MAKE SURE THE ROUTINE RETURNED A REASONABLE ANSWER
         IF(W_AIR_OUT.GT.W_AIR_IN) THEN
            W_AIR_OUT=W_AIR_IN
            H_AIR_OUT=H_AIR_IN-Q_TOT/FLOW_AIR

            PSYDAT(1)=P_AIR_OUT
            PSYDAT(6)=W_AIR_OUT
            PSYDAT(7)=H_AIR_OUT
            CALL PSYCHROMETRICS(TIME,INFO,1,7,0,PSYDAT,0,STATUS,*70)
70          IF(ErrorFound()) RETURN 1
            T_AIR_OUT=PSYDAT(2)
            W_AIR_OUT=PSYDAT(6)
            RH_AIR_OUT=PSYDAT(4)
            H_AIR_OUT=PSYDAT(7)
         ELSE IF(RH_AIR_OUT.GE.0.999) THEN
            H_AIR_OUT=H_AIR_IN-Q_TOT/FLOW_AIR

            PSYDAT(1)=P_AIR_OUT
            PSYDAT(4)=RH_AIR_OUT
            PSYDAT(7)=H_AIR_OUT
```

```
            CALL PSYCHROMETRICS(TIME,INFO,1,8,0,PSYDAT,0,STATUS,*80)
80          IF(ErrorFound()) RETURN 1
            T_AIR_OUT=PSYDAT(2)
            W_AIR_OUT=PSYDAT(6)
            RH_AIR_OUT=PSYDAT(4)
            H_AIR_OUT=PSYDAT(7)
         ENDIF

C       ADD IN THE FAN POWER
C        W_AIR_OUT=W_AIR_OUT
C        H_AIR_OUT=H_AIR_OUT+BLOWPOW/FLOW_AIR

C         PSYDAT(1)=P_AIR_OUT
C        PSYDAT(6)=W_AIR_OUT
C         PSYDAT(7)=H_AIR_OUT
C         CALL PSYCHROMETRICS(TIME,INFO,1,7,0,PSYDAT,0,STATUS,*90)
C90       IF(ErrorFound()) RETURN 1
C         T_AIR_OUT=PSYDAT(2)
C         W_AIR_OUT=PSYDAT(6)
C         RH_AIR_OUT=PSYDAT(4)
C         H_AIR_OUT=PSYDAT(7)

C       RE-CALCULATE THE HEAT TRANSFER
         Q_TOT=FLOW_AIR*(H_AIR_IN-H_AIR_OUT)
         Q_SENS=FLOW_AIR*CP_AIR*(T_AIR_IN-T_AIR_OUT)
         IF(Q_SENS.GT.Q_TOT) THEN
            Q_TOT=Q_SENS
            Q_LAT=Q_TOT-Q_SENS
         ELSE
            Q_LAT=Q_TOT-Q_SENS
         ENDIF

C       CALCULATE THE POWER REQUIREMENTS
         POWER_BLOWER=BLOWPOW
         POWER_CONT=CONTPOW
         POWER_COMPRESSOR=POWER
         IF(POWER_COMPRESSOR.LT.0) POWER_COMPRESSOR=0.

C       CALCULATE THE OUTLET FLUID TEMPERATURE
         T_FLUID_OUT=T_FLUID_IN + REJECTED/FLOW_FLUID/CP_FLUID

C    ** NOTE THAT THE ENERGY BALANCE FOR THE HEAT PUMP CANNOT BE SIMPLY
PERFORMED AS THE COMPRESSOR
C       POWER INCLUDES ATHE ENERGY THAT IS CONVECTED/RADIATED TO THE
SURROUNDINGS AND NOT PART OF THE
C       REFRIGERWNT CYCLE.

C       CALCULATE THE COP AND EER
         COP=Q_TOT/DMAX1(0.0001,(POWER_COMPRESSOR+POWER_BLOWER+
      1      POWER_CONT))
         EER=COP*3.413

      ENDIF
C----------------------------------------------------------------------------
------------------------------------------
```

```
C---------------------------------------------------------------------------
-------------------------------------------
C    SET THE OUTPUTS FROM THIS MODEL IN SEQUENTIAL ORDER AND GET OUT
      OUT(1)=T_FLUID_OUT
      OUT(2)=FLOW_FLUID
      OUT(3)=T_AIR_OUT
      OUT(4)=W_AIR_OUT
      OUT(5)=RH_AIR_OUT*100.
      OUT(6)=FLOW_AIR
      OUT(7)=P_AIR_OUT
      OUT(8)=Q_TOT
      OUT(9)=Q_SENS
      OUT(10)=OUT(8)-OUT(9)
      OUT(11)=REJECTED
      OUT(12)=POWER_COMPRESSOR
      OUT(13)=POWER_COMPRESSOR+POWER_BLOWER+POWER_CONT
      OUT(14)=Q_ENV
      OUT(15)=COP
      OUT(16)=EER
      OUT(17)=T_AIR_OUT
      OUT(18)=FLOW_AIR*(W_AIR_IN-W_AIR_OUT)

C---------------------------------------------------------------------------
-------------------------------------------

C---------------------------------------------------------------------------
-------------------------------------------
C    EVERYTHING IS DONE - RETURN FROM THIS SUBROUTINE AND MOVE ON
      RETURN 1
      END
C---------------------------------------------------------------------------

-------------------------------------------
```

Appendix B: Performance Map for the HPWH model (Type 994)

This performance map is based on the following rated conditions:
Rated Cooling: 2729.4 kJ/hr
Rated Sensible Cooling: 2457.8 kJ/hr
Rated Compressor Power: 1738.9 kJ/hr
Rated Heat Rejected: 4623.6 kJ/hr
Rated Air Flow: 64.25 kJ/hr

5 10 15 20 25 30 35 40 45 50 55 ! Water Temperature
0 3 6.1 10.3 14.1 16.2 20.1 23.9 29 32 35 !Air WB Temperature
-20 80 !Air DB Temperature
0.106241894 0.077498162 0.421002991 0.822127995 0.927854366 !Total Cooling Sens Cooling Compressor Power Heat Rejected Fan Power

0.106241894	0.077498162	0.421002991	0.822127995	0.927854366
0.455617803	0.332350458	0.488335406	0.901876703	1.015280331
0.455617803	0.332350458	0.488335406	0.901876703	1.015280331
0.77652928	0.566439371	0.549705909	0.988791158	1.135371623
0.77652928	0.566439371	0.549705909	0.988791158	1.135371623
1.111526929	0.896648974	0.591665697	1.03367815	1.103914241
1.111526929	0.896648974	0.591665697	1.03367815	1.103914241
1.503149052	1.234845007	0.670146843	1.138049874	1.414290177
1.503149052	1.234845007	0.670146843	1.138049874	1.414290177
1.518042307	1.356220805	0.68964466	1.195830748	1.297686132
1.518042307	1.356220805	0.68964466	1.195830748	1.297686132
1.670632984	1.314428671	0.67115149	1.227707794	1.443164171
1.670632984	1.314428671	0.67115149	1.227707794	1.443164171
1.844126134	1.368114079	0.669811045	1.235319005	1.483031742
1.844126134	1.368114079	0.669811045	1.235319005	1.483031742
1.890316365	1.199032995	0.672292343	1.302732117	1.583388037
1.890316365	1.199032995	0.672292343	1.302732117	1.583388037
1.866134658	0.997041929	0.635428818	1.306085669	1.624773346
1.866134658	0.997041929	0.635428818	1.306085669	1.624773346
1.808589823	0.741930831	0.598529523	1.309966402	1.663449384
1.808589823	0.741930831	0.598529523	1.309966402	1.663449384
0.192699451	0.149223975	0.415027932	0.663355986	0.413614936
0.192699451	0.149223975	0.415027932	0.663355986	0.413614936
0.564465734	0.437115001	0.492899454	0.822043645	0.773426991
0.564465734	0.437115001	0.492899454	0.822043645	0.773426991
0.912744406	0.706817523	0.566583678	1.000916534	1.172352323
0.912744406	0.706817523	0.566583678	1.000916534	1.172352323

1.21713131	1.022682216	0.61211277	1.051710473	1.201504261
1.21713131	1.022682216	0.61211277	1.051710473	1.201504261
1.576979656	1.380497149	0.710916139	1.294334867	1.872247977
1.576979656	1.380497149	0.710916139	1.294334867	1.872247977
1.697773207	1.631176178	0.716736466	1.349685568	1.878533946
1.697773207	1.631176178	0.716736466	1.349685568	1.878533946
1.779113296	1.44571417	0.718820789	1.38778601	1.879344098
1.779113296	1.44571417	0.718820789	1.38778601	1.879344098
1.830100645	1.41812126	0.718936778	1.443363272	1.90044966
1.830100645	1.41812126	0.718936778	1.443363272	1.90044966
1.867667156	1.219466291	0.718056003	1.476398445	1.947799654
1.867667156	1.219466291	0.718056003	1.476398445	1.947799654
1.772205949	0.918768555	0.682953481	1.468890889	1.821067739
1.772205949	0.918768555	0.682953481	1.468890889	1.821067739
1.650316952	0.571866978	0.644403483	1.444033694	1.678008941
1.650316952	0.571866978	0.644403483	1.444033694	1.678008941
0.10944256	0.093228375	0.192006992	1.480920448	0.122229367
0.10944256	0.093228375	0.192006992	1.480920448	0.122229367
0.10944256	0.093228375	0.345377149	1.4053403	0.579047186
0.10944256	0.093228375	0.345377149	1.4053403	0.579047186
0.567326113	0.48327535	0.463129039	1.502931449	0.98753609
0.567326113	0.48327535	0.463129039	1.502931449	0.98753609
1.192961849	1.008084344	0.640139124	1.032558949	1.310409921
1.192961849	1.008084344	0.640139124	1.032558949	1.310409921
1.583536088	1.443681197	0.749810652	1.29374718	1.88450673
1.583536088	1.443681197	0.749810652	1.29374718	1.88450673
1.742986958	1.754637756	0.773000739	1.380488099	1.889153266
1.742986958	1.754637756	0.773000739	1.380488099	1.889153266
1.873844014	1.601703927	0.780527357	1.449693192	1.881799495
1.873844014	1.601703927	0.780527357	1.449693192	1.881799495
1.91695564	1.503611674	0.781536272	1.505400285	1.901807651
1.91695564	1.503611674	0.781536272	1.505400285	1.901807651
1.988107785	1.302911086	0.783635857	1.55148096	1.92598876
1.988107785	1.302911086	0.783635857	1.55148096	1.92598876
1.804542644	0.894975991	0.706389744	1.739533211	1.708944282
1.804542644	0.894975991	0.706389744	1.739533211	1.708944282
1.60088267	0.437673228	0.627711848	1.884256719	1.485898276
1.60088267	0.437673228	0.627711848	1.884256719	1.485898276
0.170713189	0.136274625	0.405861191	0.496019242	0.930873787
0.170713189	0.136274625	0.405861191	0.496019242	0.930873787
0.539819248	0.430919638	0.512542905	0.700339937	1.202676581
0.539819248	0.430919638	0.512542905	0.700339937	1.202676581
0.886549408	0.707702719	0.597875916	0.901006167	1.314200568
0.886549408	0.707702719	0.597875916	0.901006167	1.314200568
1.26833424	1.111465859	0.730579621	1.099764344	1.898721105

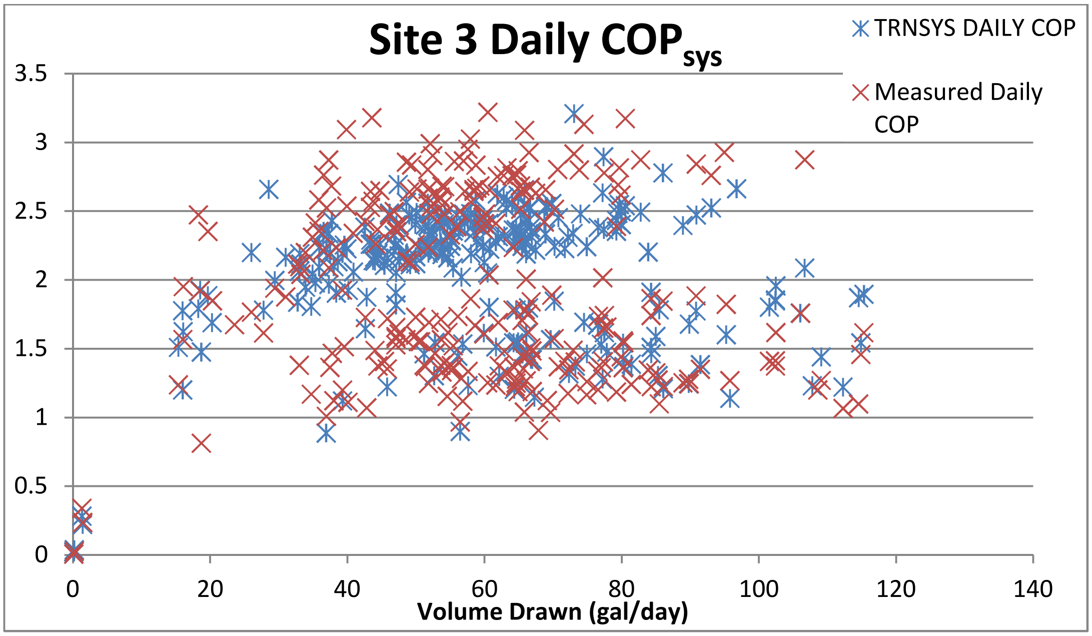
1.26833424	1.111465859	0.730579621	1.099764344	1.898721105
1.533003766	1.436209776	0.784002599	1.268784274	1.888011383
1.533003766	1.436209776	0.784002599	1.268784274	1.888011383
1.739372576	1.828672324	0.820128492	1.379829852	1.891471852
1.739372576	1.828672324	0.820128492	1.379829852	1.891471852
1.910047628	1.747128365	0.851016747	1.508403536	1.885326116
1.910047628	1.747128365	0.851016747	1.508403536	1.885326116
2.011571299	1.659087143	0.856806048	1.563291088	1.90183535
2.011571299	1.659087143	0.856806048	1.563291088	1.90183535
2.110909192	1.417364754	0.860991553	1.645805717	1.919444112
2.110909192	1.417364754	0.860991553	1.645805717	1.919444112
2.099207201	1.109038266	0.825316533	1.659170394	1.730709276
2.099207201	1.109038266	0.825316533	1.659170394	1.730709276
2.052733132	0.718531095	0.783346604	1.653228415	1.568154626
2.052733132	0.718531095	0.783346604	1.653228415	1.568154626
0.20327308	0.161964347	0.514228861	0.36692101	1.270078294
0.20327308	0.161964347	0.514228861	0.36692101	1.270078294
0.532231442	0.424072474	0.597299719	0.590840283	1.443605795
0.532231442	0.424072474	0.597299719	0.590840283	1.443605795
0.868775513	0.692224758	0.678124216	0.805870714	1.512312351
0.868775513	0.692224758	0.678124216	0.805870714	1.512312351
1.190189313	1.057224051	0.761450865	1.057864376	1.89469812
1.190189313	1.057224051	0.761450865	1.057864376	1.89469812
1.443262725	1.393027231	0.818819229	1.225112659	1.889024862
1.443262725	1.393027231	0.818819229	1.225112659	1.889024862
1.699869646	1.825251527	0.855788095	1.347251244	1.880548997
1.699869646	1.825251527	0.855788095	1.347251244	1.880548997
1.9050848	1.865554264	0.913583288	1.498999808	1.886305665
1.9050848	1.865554264	0.913583288	1.498999808	1.886305665
2.055485453	1.81789204	0.937052818	1.618321142	1.902135635
2.055485453	1.81789204	0.937052818	1.618321142	1.902135635
2.210616434	1.539898448	0.945701358	1.701415121	1.917886176
2.210616434	1.539898448	0.945701358	1.701415121	1.917886176
2.283016742	1.31206152	0.948564457	1.743499115	1.800691221
2.283016742	1.31206152	0.948564457	1.743499115	1.800691221
2.316289822	0.962771248	0.938169062	1.758060981	1.701043464
2.316289822	0.962771248	0.938169062	1.758060981	1.701043464
0.384072618	0.303571252	0.603314171	0.497900889	2.231672403
0.384072618	0.303571252	0.603314171	0.497900889	2.231672403
0.621810674	0.491479569	0.665859752	0.669899897	2.099895261
0.621810674	0.491479569	0.665859752	0.669899897	2.099895261
0.860959367	0.680502855	0.735315393	0.852259809	1.919765776
0.860959367	0.680502855	0.735315393	0.852259809	1.919765776
1.161998452	1.05347902	0.79335228	1.036053627	1.896398061
1.161998452	1.05347902	0.79335228	1.036053627	1.896398061

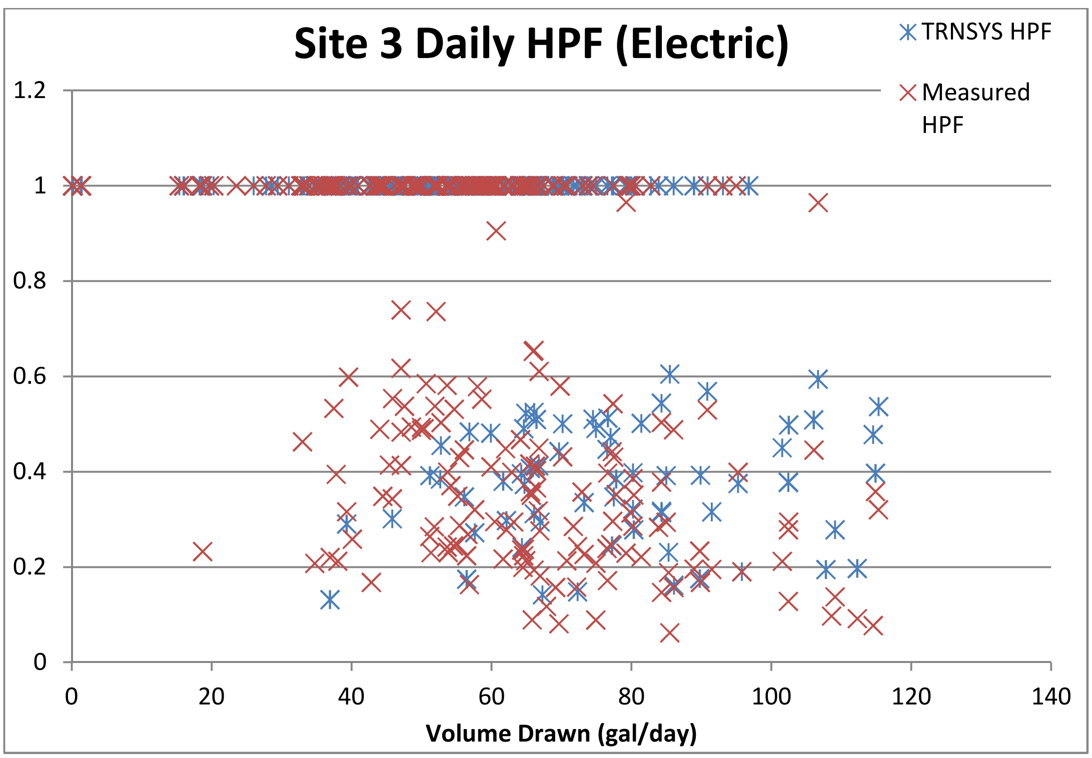
1.367131892	1.350819602	0.855150724	1.221641663	1.881744923
1.367131892	1.350819602	0.855150724	1.221641663	1.881744923
1.62707662	1.781613198	0.895240052	1.323711477	1.877790969
1.62707662	1.781613198	0.895240052	1.323711477	1.877790969
1.76817963	1.787977911	0.953351101	1.488553055	1.612879628
1.76817963	1.787977911	0.953351101	1.488553055	1.612879628
2.023310114	1.917155602	1.013381186	1.653541536	1.645553339
2.023310114	1.917155602	1.013381186	1.653541536	1.645553339
2.287228058	1.730513003	1.039843485	1.78605977	1.917184634
2.287228058	1.730513003	1.039843485	1.78605977	1.917184634
2.43449677	1.613386085	1.07556994	1.888689186	1.900158271
2.43449677	1.613386085	1.07556994	1.888689186	1.900158271
2.571797217	1.390321712	1.097791983	1.968853478	1.990610518
2.571797217	1.390321712	1.097791983	1.968853478	1.990610518
0.369710654	0.208183911	0.664962516	0.555172143	2.449390961
0.369710654	0.208183911	0.664962516	0.555172143	2.449390961
0.583610891	0.472414372	0.714346291	0.688010461	2.238745168
0.583610891	0.472414372	0.714346291	0.688010461	2.238745168
0.79797751	0.645937303	0.765938036	0.82612917	1.90808007
0.79797751	0.645937303	0.765938036	0.82612917	1.90808007
1.053753092	0.986886513	0.83021718	1.00037488	1.890174113
1.053753092	0.986886513	0.83021718	1.00037488	1.890174113
1.296396842	1.2894424	0.891862655	1.169282144	1.736192534
1.296396842	1.2894424	0.891862655	1.169282144	1.736192534
1.564814309	1.737735884	0.936534749	1.284662004	1.86926957
1.564814309	1.737735884	0.936534749	1.284662004	1.86926957
1.675269132	1.632821778	0.986634754	1.426000343	1.270843778
1.675269132	1.632821778	0.986634754	1.426000343	1.270843778
1.824884098	1.668752839	1.047042239	1.582011791	1.007402498
1.824884098	1.668752839	1.047042239	1.582011791	1.007402498
2.224461269	1.782017593	1.130174272	1.813579476	1.53178104
2.224461269	1.782017593	1.130174272	1.813579476	1.53178104
2.342901441	1.652519355	1.17507221	1.934039714	1.314820624
2.342901441	1.652519355	1.17507221	1.934039714	1.314820624
2.492992353	1.51108795	1.221011038	2.059000553	1.334309615
2.492992353	1.51108795	1.221011038	2.059000553	1.334309615
0.337513476	0.27502678	0.699007125	0.521151115	2.369330554
0.337513476	0.27502678	0.699007125	0.521151115	2.369330554
0.533339182	0.434597633	0.748823243	0.654520316	2.17637719
0.533339182	0.434597633	0.748823243	0.654520316	2.17637719
0.723682456	0.589701063	0.799237657	0.791748786	1.893119402
0.723682456	0.589701063	0.799237657	0.791748786	1.893119402
0.972022556	0.918805292	0.869751735	0.965173075	1.87911059
0.972022556	0.918805292	0.869751735	0.965173075	1.87911059
1.205932665	1.187486776	0.92409191	1.111931974	1.358115024

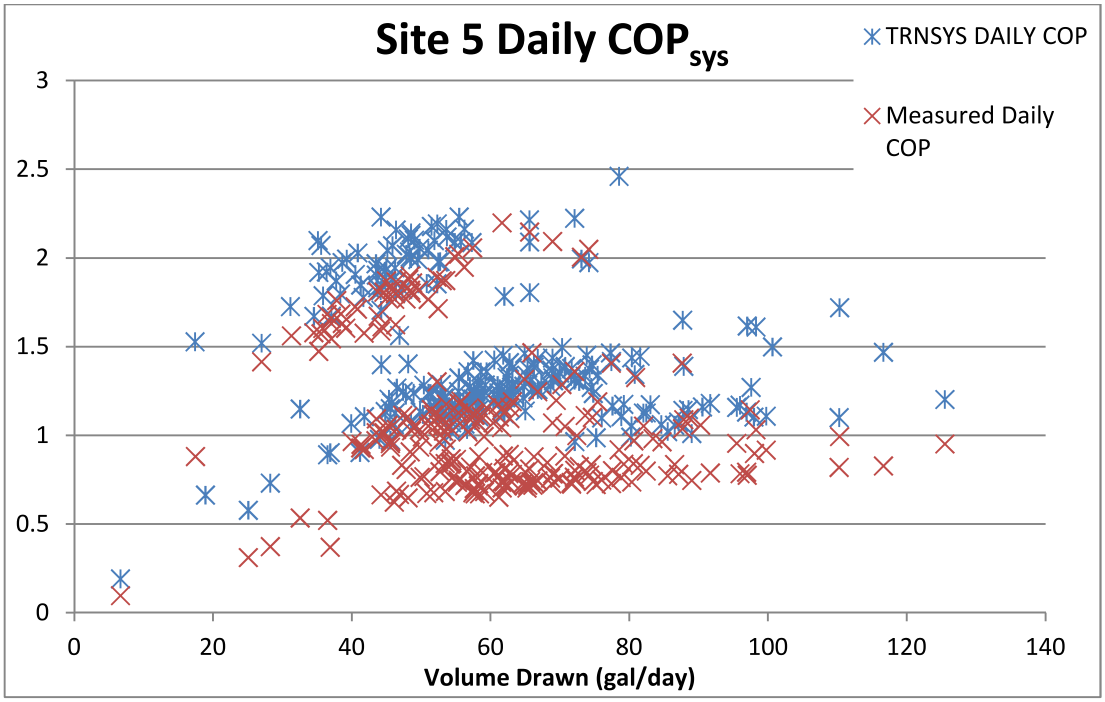
1.205932665	1.187486776	0.92409191	1.111931974	1.358115024
1.440424106	1.599599801	0.976486961	1.240880088	1.571094374
1.440424106	1.599599801	0.976486961	1.240880088	1.571094374
1.535601275	1.499521897	1.022167264	1.361688199	1.032344687
1.535601275	1.499521897	1.022167264	1.361688199	1.032344687
1.687002372	1.529156666	1.08486532	1.512339939	0.85114868
1.687002372	1.529156666	1.08486532	1.512339939	0.85114868
2.073779879	1.480040014	1.169906477	1.713580551	0.724771843
2.073779879	1.480040014	1.169906477	1.713580551	0.724771843
2.184276776	1.316136827	1.214902561	1.821263271	0.501641564
2.184276776	1.316136827	1.214902561	1.821263271	0.501641564
2.330024365	1.106648349	1.261516647	1.929289163	0.348097902
2.330024365	1.106648349	1.261516647	1.929289163	0.348097902
0.273657694	0.226516764	0.715454351	0.471947148	2.360409538
0.273657694	0.226516764	0.715454351	0.471947148	2.360409538
0.454535563	0.376236179	0.773756893	0.61039913	2.111032807
0.454535563	0.376236179	0.773756893	0.61039913	2.111032807
0.62235679	0.515148119	0.8350079	0.759144375	1.88281114
0.62235679	0.515148119	0.8350079	0.759144375	1.88281114
0.873623835	0.824026514	0.903815791	0.913526909	1.526873671
0.873623835	0.824026514	0.903815791	0.913526909	1.526873671
1.131684944	1.133337557	0.965358696	1.06574817	1.312765129
1.131684944	1.133337557	0.965358696	1.06574817	1.312765129
1.30276184	1.44672501	1.014256414	1.176714481	1.247844639
1.30276184	1.44672501	1.014256414	1.176714481	1.247844639
1.435260759	1.4303805	1.067271337	1.308142881	0.980802231
1.435260759	1.4303805	1.067271337	1.308142881	0.980802231
1.557126866	1.40770371	1.128910883	1.454339558	0.749935217
1.557126866	1.40770371	1.128910883	1.454339558	0.749935217
1.996785026	1.336333392	1.194510444	1.602130476	0.594555541
1.996785026	1.336333392	1.194510444	1.602130476	0.594555541
2.107210332	1.16147353	1.235950411	1.702076945	0.475192826
2.107210332	1.16147353	1.235950411	1.702076945	0.475192826
2.268265173	0.935403575	1.273274113	1.789489339	0.386119905
2.268265173	0.935403575	1.273274113	1.789489339	0.386119905
0.268484456	0.224309342	0.743633166	0.443571058	2.500858366
0.268484456	0.224309342	0.743633166	0.443571058	2.500858366
0.414651034	0.34642639	0.803196509	0.573640566	2.129558786
0.414651034	0.34642639	0.803196509	0.573640566	2.129558786
0.547323853	0.457269875	0.868904063	0.713694623	1.885007502
0.547323853	0.457269875	0.868904063	0.713694623	1.885007502
0.784464506	0.758893405	0.933745544	0.859838237	1.26400682
0.784464506	0.758893405	0.933745544	0.859838237	1.26400682
1	1	1	1	1
1	1	1	1	1

1.188347907	1.319667636	1.053776691	1.116510659	1.06066462
1.188347907	1.319667636	1.053776691	1.116510659	1.06066462
1.290640523	1.291872016	1.109752244	1.229086935	0.842196258
1.290640523	1.291872016	1.109752244	1.229086935	0.842196258
1.435381947	1.352603052	1.179555349	1.37713516	0.73400632
1.435381947	1.352603052	1.179555349	1.37713516	0.73400632
1.889116822	1.259172542	1.244805275	1.517412669	0.554343134
1.889116822	1.259172542	1.244805275	1.517412669	0.554343134
2.018435223	1.13737647	1.292840792	1.614454348	0.633355591
2.018435223	1.13737647	1.292840792	1.614454348	0.633355591
2.204086064	0.956125133	1.334583163	1.699725623	0.695095889
2.204086064	0.956125133	1.334583163	1.699725623	0.695095889
0.295519387	0.248095453	0.750994162	0.413962165	2.469291691
0.295519387	0.248095453	0.750994162	0.413962165	2.469291691
0.404468328	0.339560643	0.818335548	0.539580542	2.236261008
0.404468328	0.339560643	0.818335548	0.539580542	2.236261008
0.499809828	0.419602067	0.888330592	0.680487198	1.891460345
0.499809828	0.419602067	0.888330592	0.680487198	1.891460345
0.703433001	0.702660243	0.971203553	0.804664701	1.198186534
0.703433001	0.702660243	0.971203553	0.804664701	1.198186534
0.909192856	0.942918378	1.040051953	0.956181267	0.977720415
0.909192856	0.942918378	1.040051953	0.956181267	0.977720415
1.111129027	1.233915596	1.090087913	1.069252384	1.021432762
1.111129027	1.233915596	1.090087913	1.069252384	1.021432762
1.153232546	1.207627732	1.157559214	1.177860948	0.790199187
1.153232546	1.207627732	1.157559214	1.177860948	0.790199187
1.311521654	1.245944057	1.231144561	1.319701429	0.667342224
1.311521654	1.245944057	1.231144561	1.319701429	0.667342224
1.807021248	1.200703744	1.301411939	1.457872263	0.534954456
1.807021248	1.200703744	1.301411939	1.457872263	0.534954456
1.943599866	1.082742678	1.350506838	1.554085774	0.817869667
1.943599866	1.082742678	1.350506838	1.554085774	0.817869667
2.153091933	0.921062654	1.393269857	1.638365032	0.757440072
2.153091933	0.921062654	1.393269857	1.638365032	0.757440072

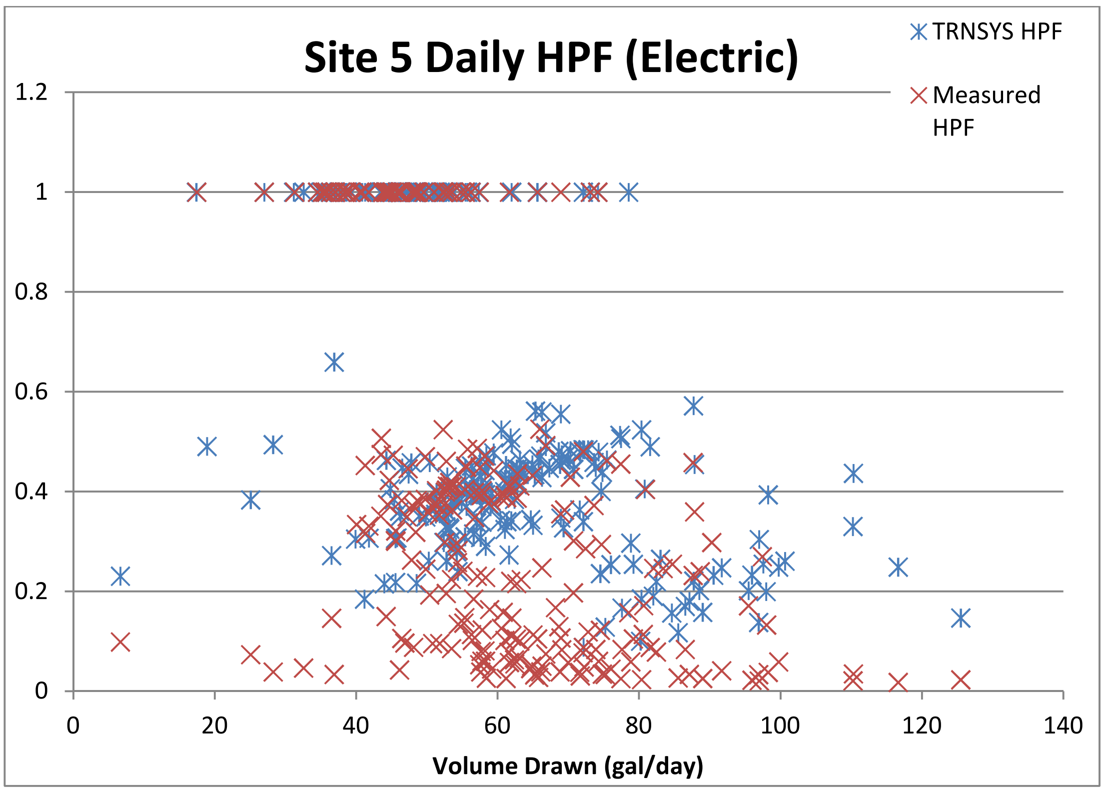
Appendix C: Comparison of System COP and Heat Pump Fraction for HPWH Field Test Data



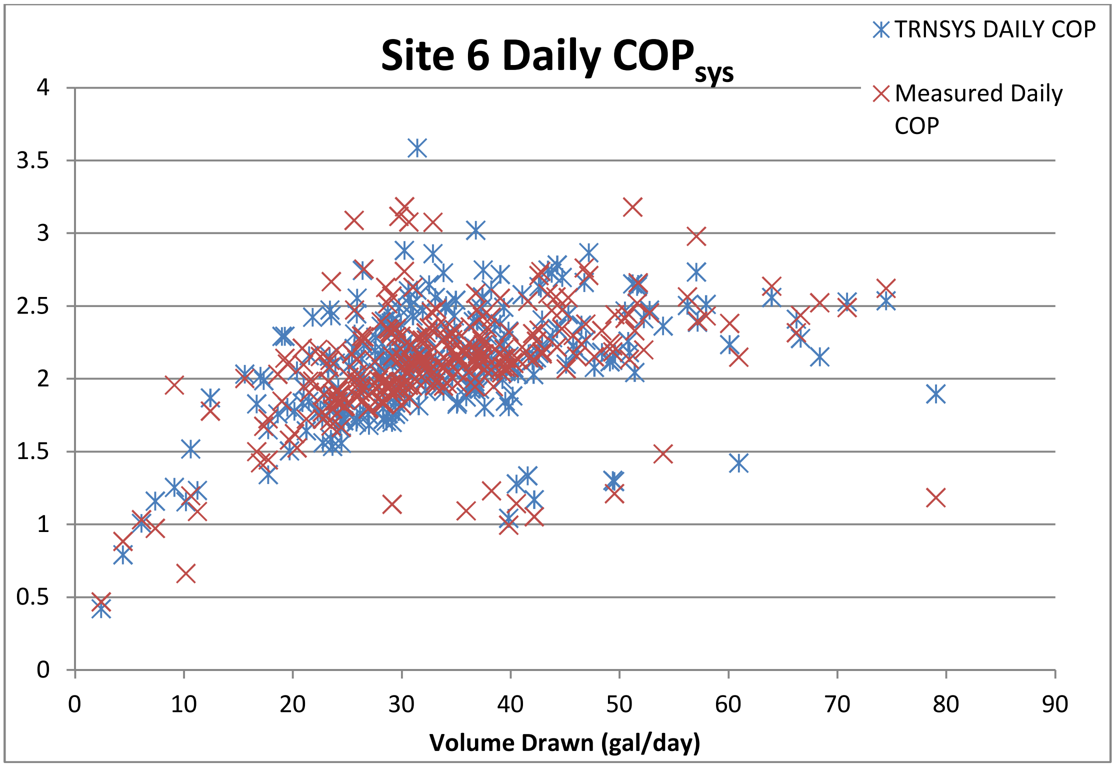




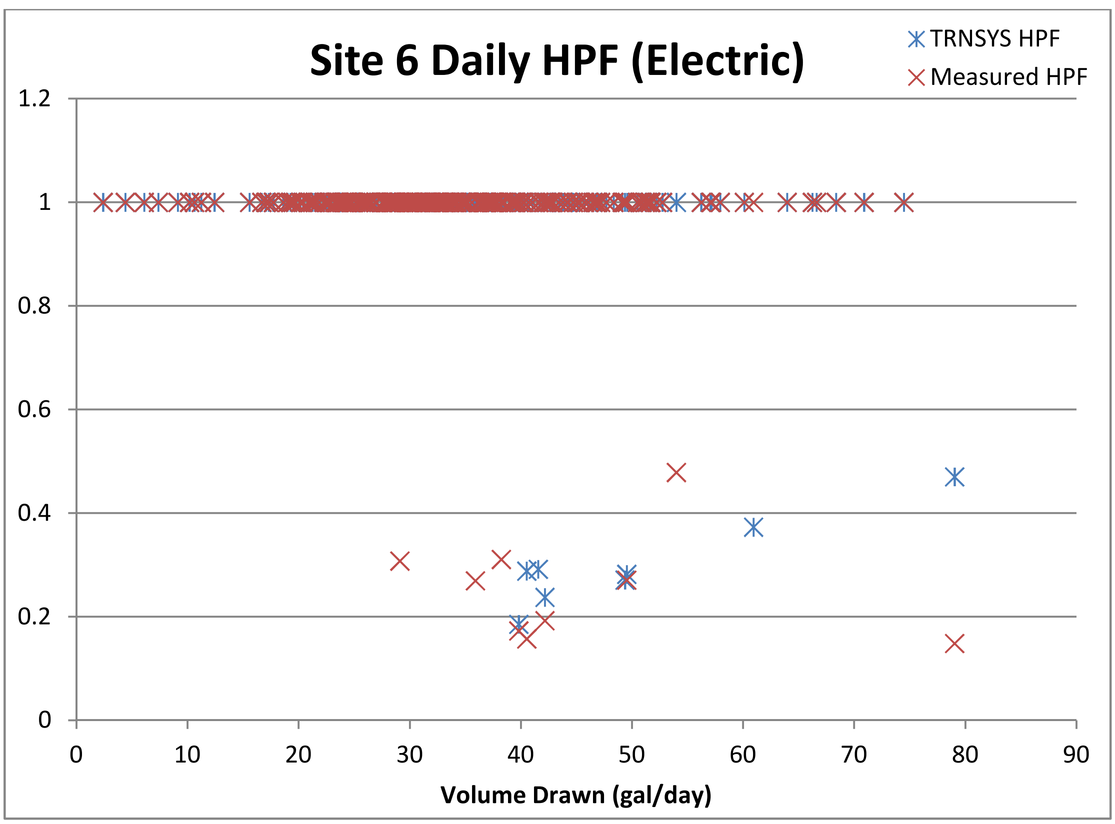
Site 5 Daily COP$_{sys}$
TRNSYS DAILY COP
Measured Daily COP
3
2.5
2
1.5
1
0.5
0
0
20
40
60
80
100
120
140
Volume Drawn (gal/day)



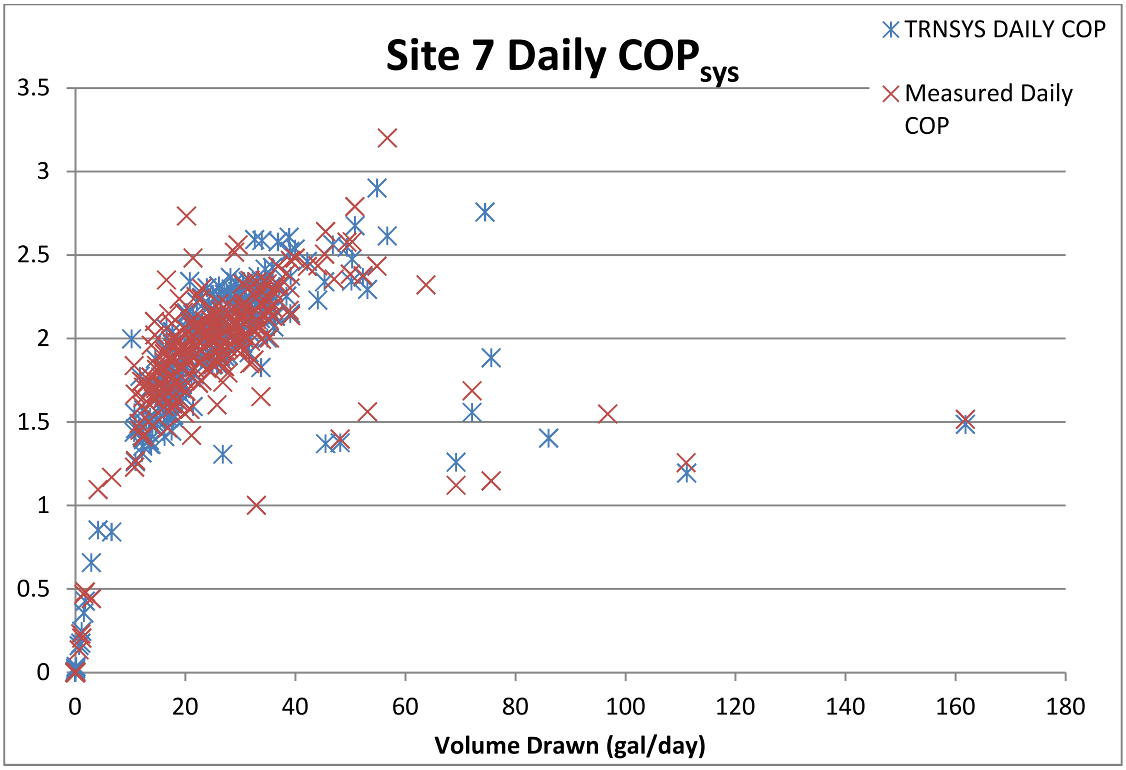
Site 5 Daily HPF (Electric)
TRNSYS HPF
Measured HPF
1.2
1
0.8
0.6
0.4
0.2
0
0
20
40
60
80
100
120
140
Volume Drawn (gal/day)

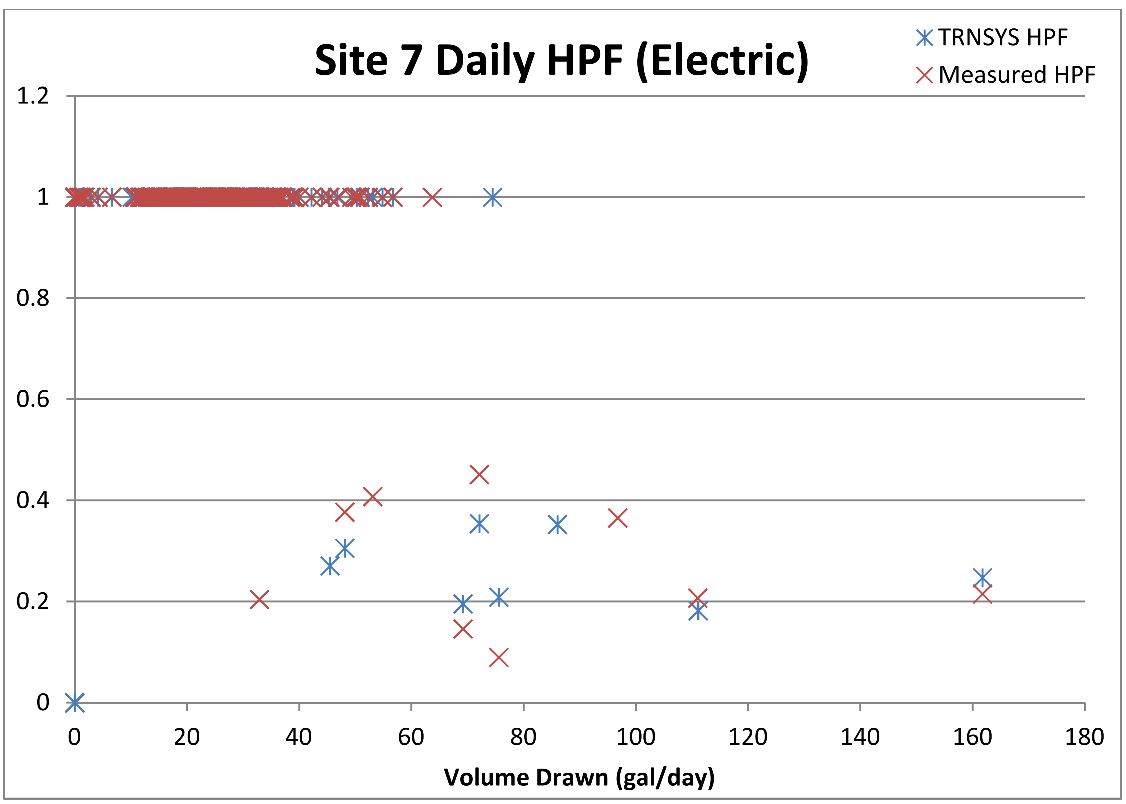


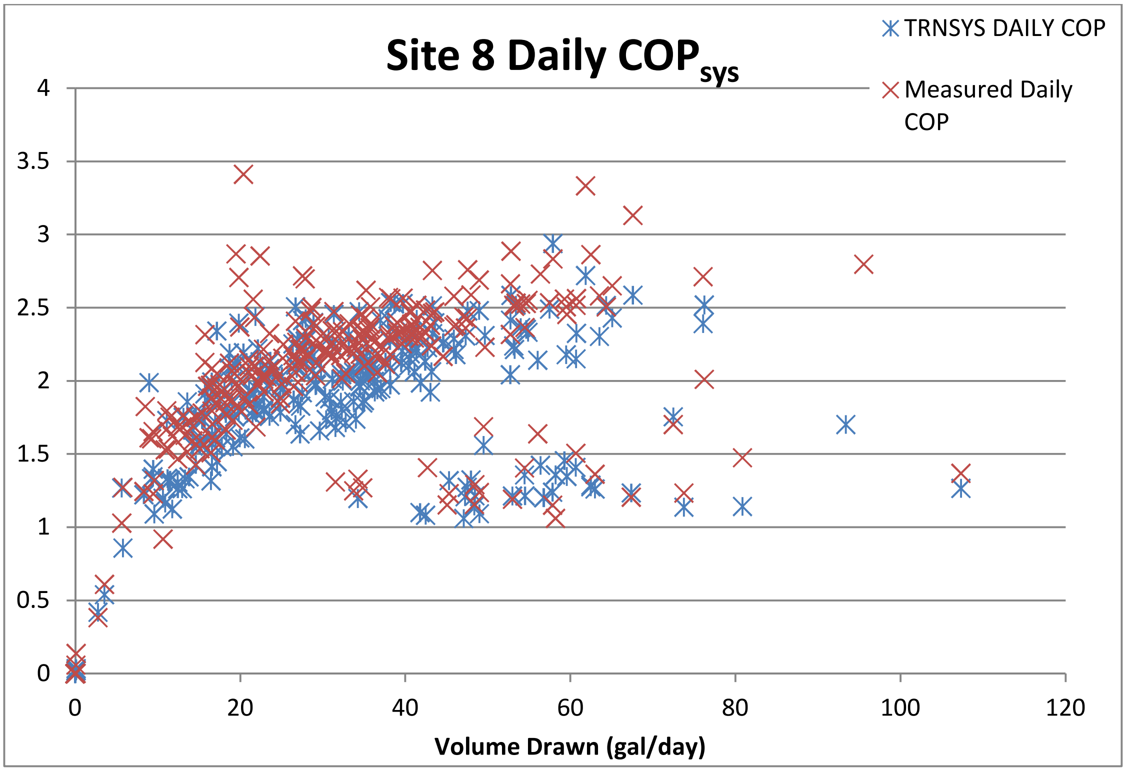
Site 6 Daily COP_{sys}
TRNSYS DAILY COP
Measured Daily COP
4
3.5
3
2.5
2
1.5
1
0.5
0
0
10
20
30
40
50
60
70
80
90
Volume Drawn (gal/day)

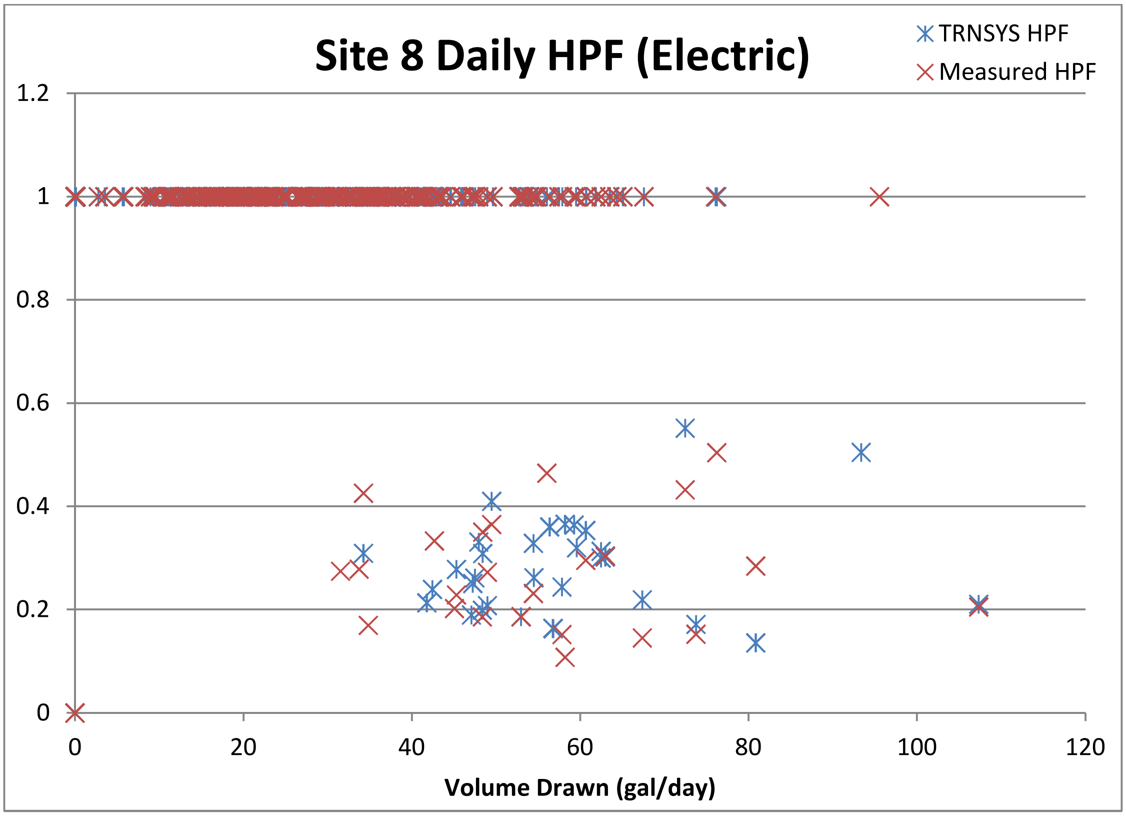


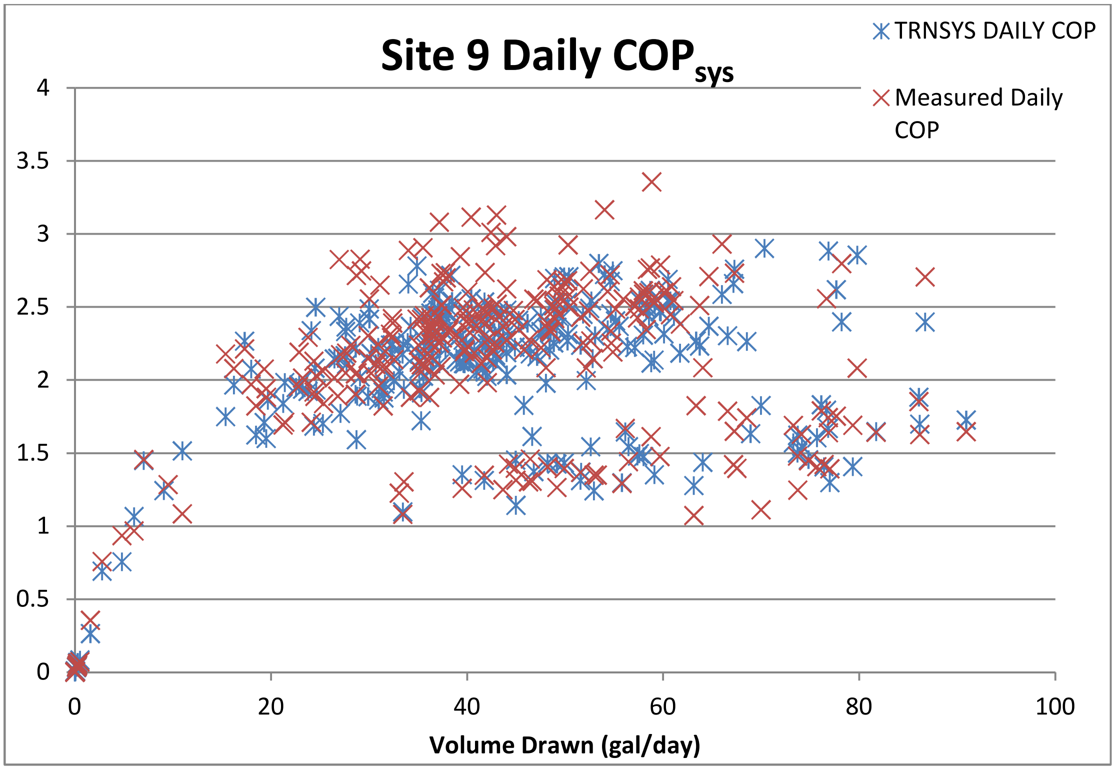
Site 6 Daily HPF (Electric)
TRNSYS HPF
Measured HPF
1.2
1
0.8
0.6
0.4
0.2
0
0
10
20
30
40
50
60
70
80
90
Volume Drawn (gal/day)

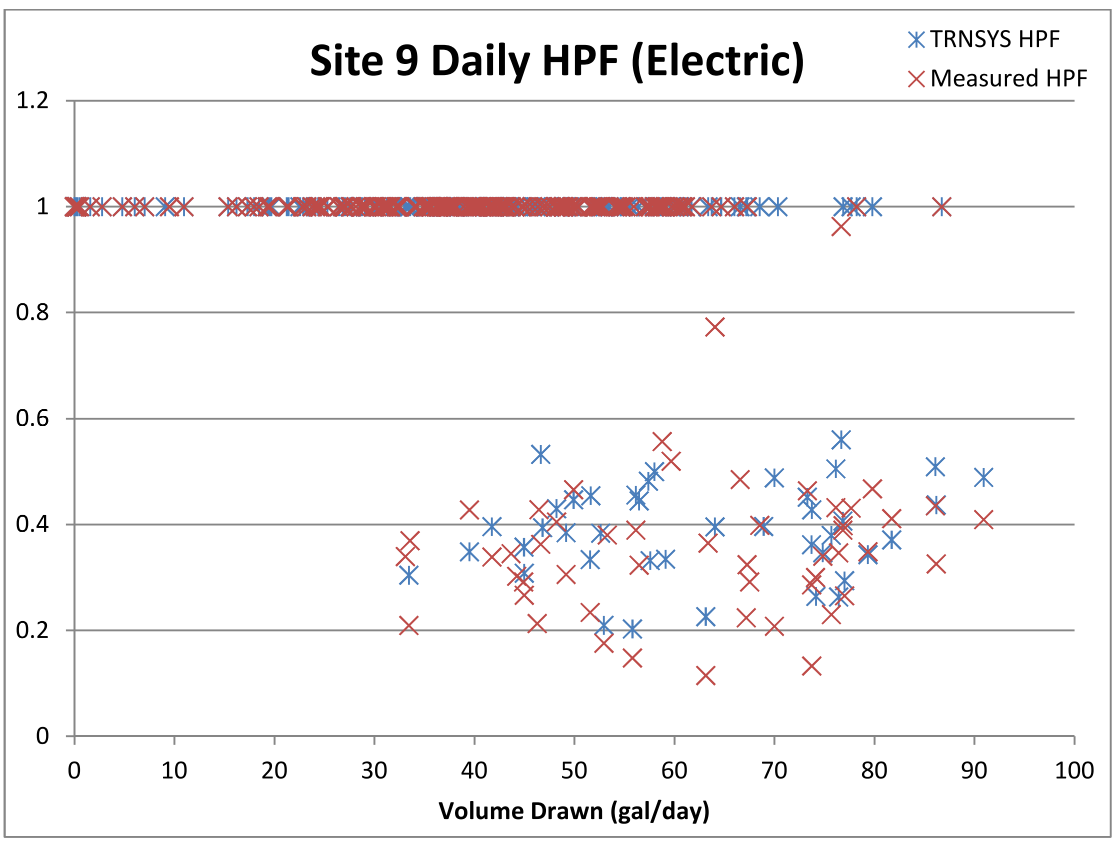

Site 7 Daily COP_{sys}
TRNSYS DAILY COP
Measured Daily COP
3.5
3
2.5
2
1.5
1
0.5
0
0
20
40
60
80
100
120
140
160
180
Volume Drawn (gal/day)

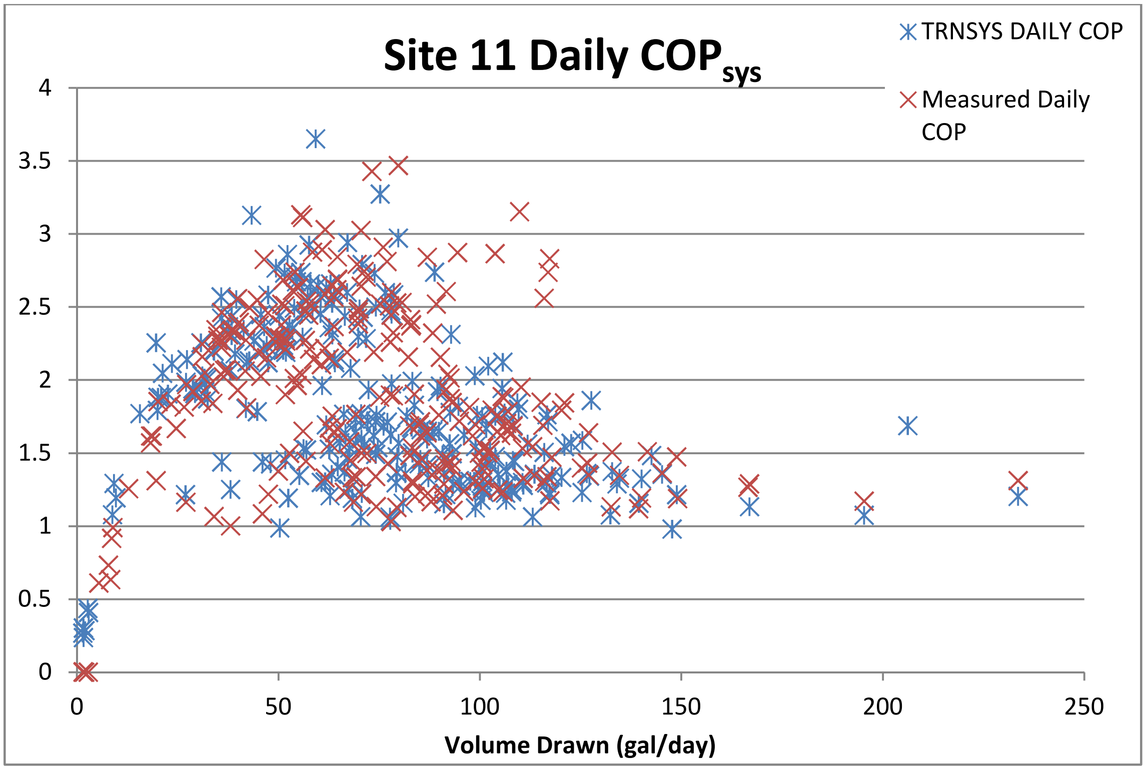

Site 7 Daily HPF (Electric)
TRNSYS HPF
Measured HPF
1.2
1
0.8
0.6
0.4
0.2
0
0
20
40
60
80
100
120
140
160
180
Volume Drawn (gal/day)

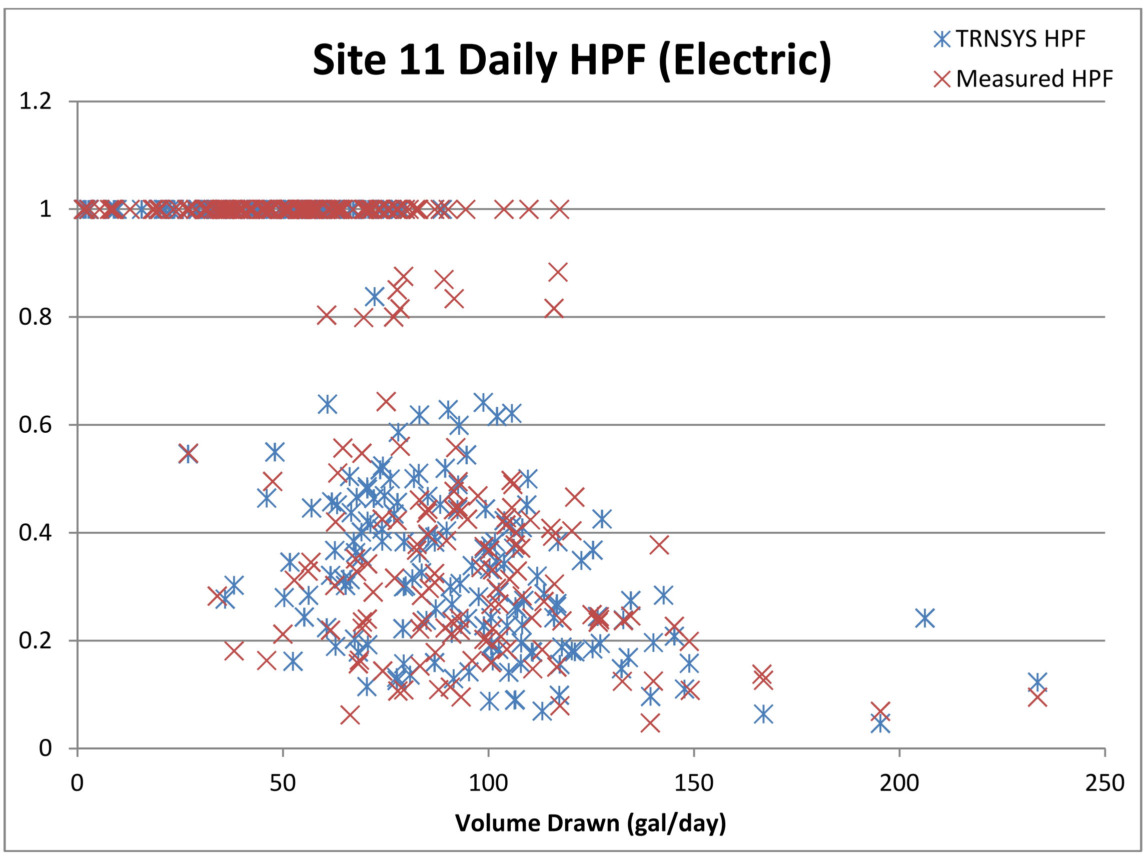

Site 8 Daily COP_{sys}
TRNSYS DAILY COP
Measured Daily COP
4
3.5
3
2.5
2
1.5
1
0.5
0
0
20
40
60
80
100
120
Volume Drawn (gal/day)

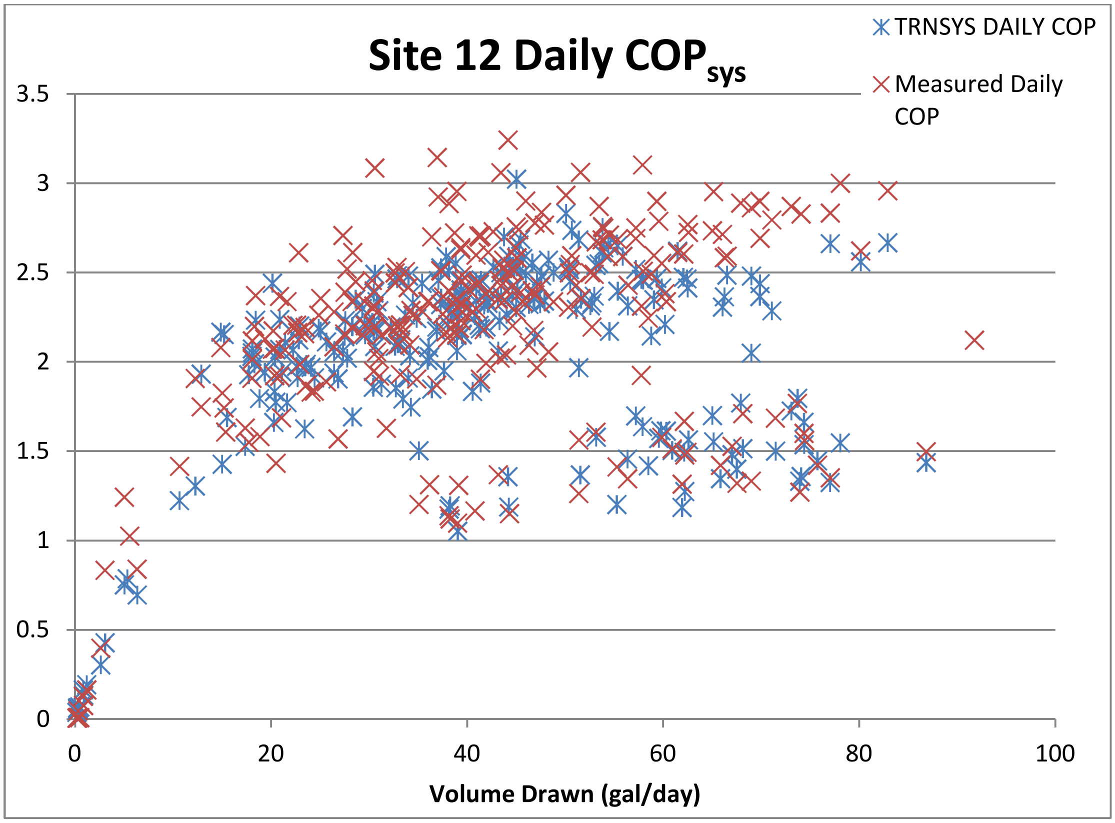

Site 8 Daily HPF (Electric)
TRNSYS HPF
Measured HPF
1.2
1
0.8
0.6
0.4
0.2
0
0
20
40
60
80
100
120
Volume Drawn (gal/day)

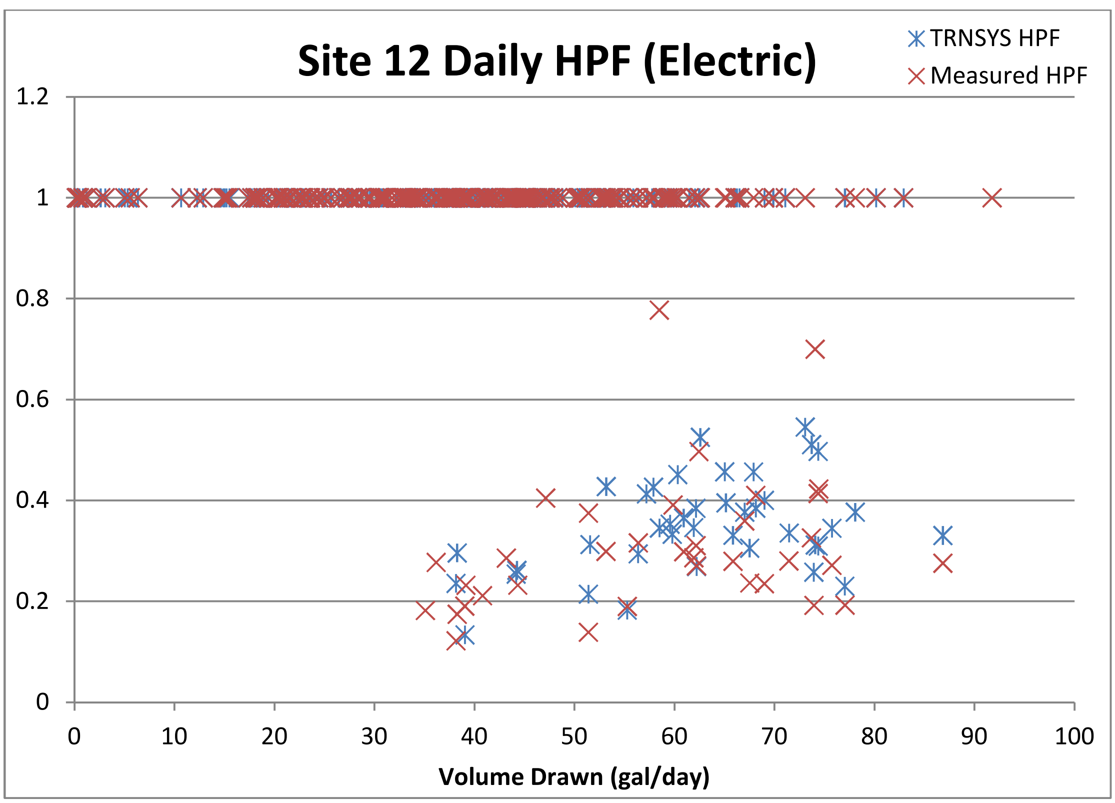

Site 9 Daily COP$_{sys}$
TRNSYS DAILY COP
Measured Daily COP
4
3.5
3
2.5
2
1.5
1
0.5
0
0
20
40
60
80
100
Volume Drawn (gal/day)

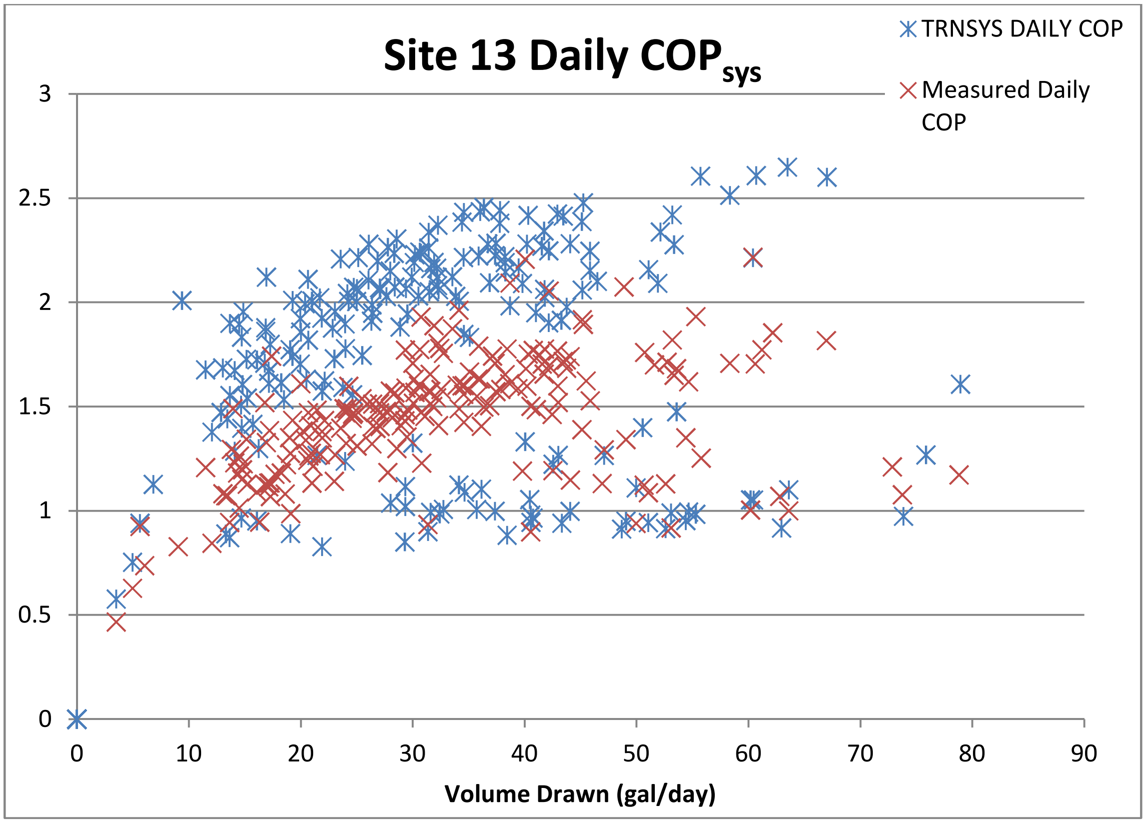

Site 9 Daily HPF (Electric)
TRNSYS HPF
Measured HPF
1.2
1
0.8
0.6
0.4
0.2
0
0
10
20
30
40
50
60
70
80
90
100
Volume Drawn (gal/day)


Site 11 Daily COP_{sys}
TRNSYS DAILY COP
Measured Daily COP
4
3.5
3
2.5
2
1.5
1
0.5
0
0
50
100
150
200
250
Volume Drawn (gal/day)

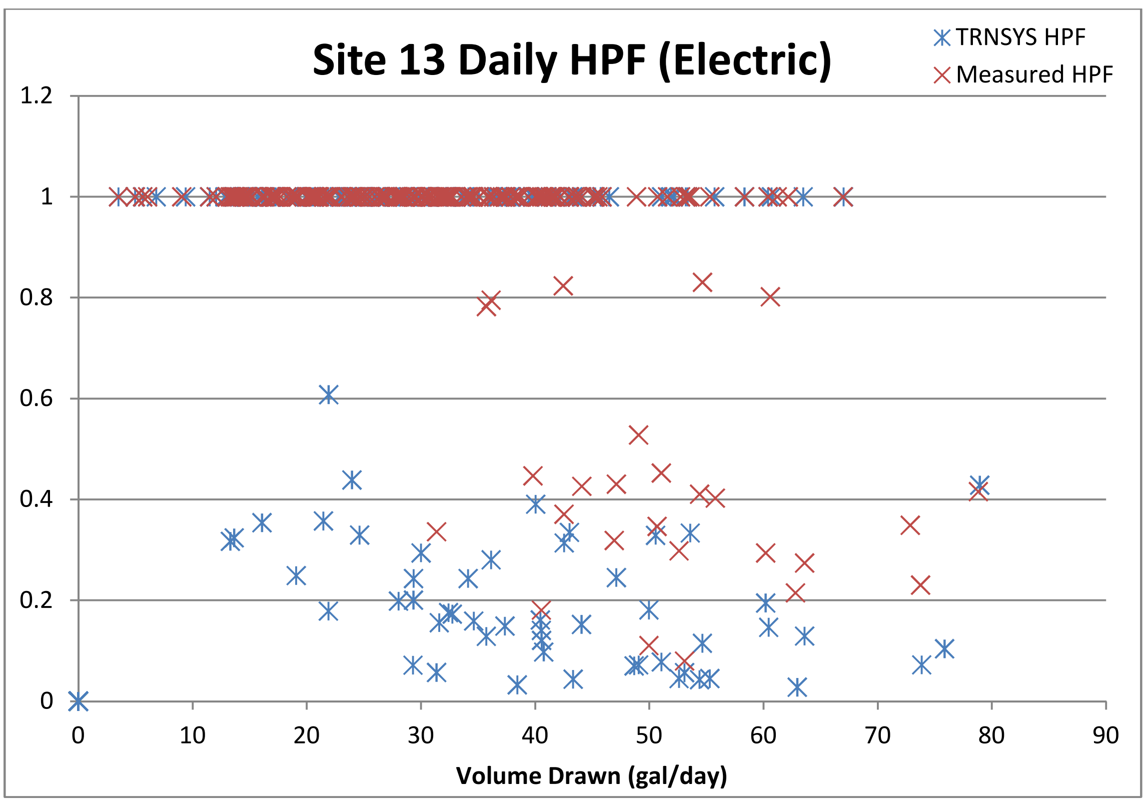

Site 11 Daily HPF (Electric)
TRNSYS HPF
Measured HPF
1.2
1
0.8
0.6
0.4
0.2
0
0
50
100
150
200
250
Volume Drawn (gal/day)


Site 12 Daily COP$_{sys}$
TRNSYS DAILY COP
Measured Daily COP
3.5
3
2.5
2
1.5
1
0.5
0
0
20
40
60
80
100
Volume Drawn (gal/day)

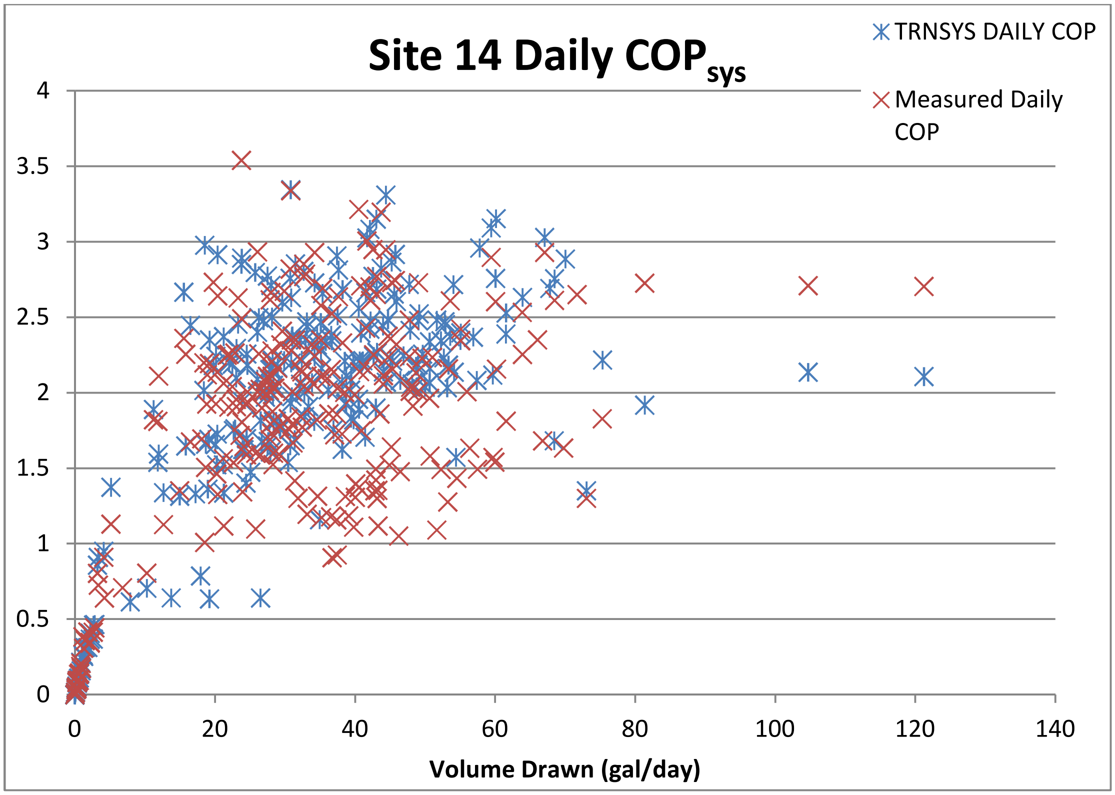

Site 12 Daily HPF (Electric)
TRNSYS HPF
Measured HPF
1.2
1
0.8
0.6
0.4
0.2
0
0
10
20
30
40
50
60
70
80
90
100
Volume Drawn (gal/day)

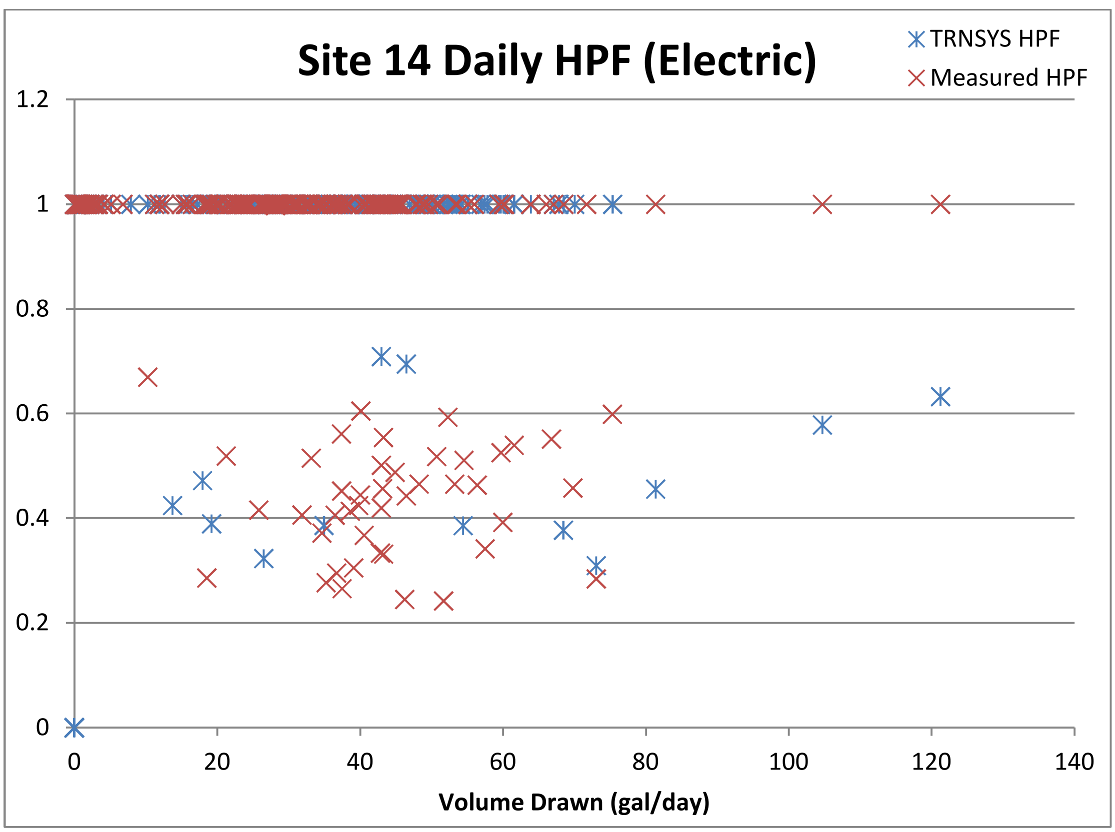


Site 13 Daily COP$_{sys}$
TRNSYS DAILY COP
Measured Daily COP
3
2.5
2
1.5
1
0.5
0
0
10
20
30
40
50
60
70
80
90
Volume Drawn (gal/day)



Site 13 Daily HPF (Electric)
TRNSYS HPF
Measured HPF
1.2
1
0.8
0.6
0.4
0.2
0
0
10
20
30
40
50
60
70
80
90
Volume Drawn (gal/day)


Site 14 Daily COP_{sys}
TRNSYS DAILY COP
Measured Daily COP
4
3.5
3
2.5
2
1.5
1
0.5
0
0
20
40
60
80
100
120
140
Volume Drawn (gal/day)


Site 14 Daily HPF (Electric)
TRNSYS HPF
Measured HPF
1.2
1
0.8
0.6
0.4
0.2
0
0
20
40
60
80
100
120
140
Volume Drawn (gal/day)

Appendix D: Condensing Water Heater TRNSYS Performance Map

0.254 0.466 0.625 0.783 1 !Part Load
14.62951852 21.111 27.59248148 34.07396296 40.55544445 47.03692593 53.51840741
59.99988889 66.48137037 72.96285185 79.444333
85.92581482 !Temperature
0.935
0.935
0.935
0.935
0.9345
0.929
0.918
0.907
0.901
0.9
0.898
0.896
0.943
0.943
0.943
0.943
0.9425
0.936
0.923
0.911
0.905
0.903
0.901
0.899
0.9525
0.9525
0.9515
0.951
0.949
0.942
0.93
0.918
0.91
0.907
0.906
0.903
0.963
0.962

0.961
0.96
0.957
0.948
0.935
0.923
0.915
0.911
0.91
0.907
0.973
0.971
0.9695
0.966
0.963
0.953
0.939
0.926
0.919
0.916
0.913
0.908

Appendix E: TRNSYS Condensing Water Heater Heat Source Model (Type 1228)

```
SUBROUTINE TYPE1228(TIME,XIN,OUT,T,DTDT,PAR,INFO,ICNTRL,*)

!---------------------------------------------------------------------------
-------------------------------------------
! This subroutine models a heating device for a condensing storage tank.
!
! Modifications:
!    November 1st, 2007 : Original Coding
!    June 2nd, 2011 : Added Performance Map for Condensing Units JBM
!
! Copyright © 2007 Thermal Energy System Specialists, LLC. All rights
reserved.
!---------------------------------------------------------------------------
-------------------------------------------

!---------------------------------------------------------------------------
-------------------------------------------
 USE TrnsysConstants
 USE TrnsysFunctions
!---------------------------------------------------------------------------
-------------------------------------------

!DEC$ATTRIBUTES DLLEXPORT :: TYPE1228

!---------------------------------------------------------------------------
-------------------------------------------
!TRNSYS Declarations
 IMPLICIT NONE
 DOUBLE PRECISION XIN,OUT,TIME,PAR,T,DTDT,TIME0,TFINAL,TimeStep,Stored
 INTEGER*4 INFO(15),NP,NI,NOUT,ND,IUNIT,ITYPE,ICNTRL,NStored
 CHARACTER*3 YCHECK,OCHECK
!---------------------------------------------------------------------------
-------------------------------------------

!---------------------------------------------------------------------------
-------------------------------------------
!User Declarations
 PARAMETER (NP=3,NI=3,NOUT=3,ND=0,NStored=0)
!---------------------------------------------------------------------------
-------------------------------------------

!---------------------------------------------------------------------------
-------------------------------------------
!Required TRNSYS Dimensions
 DIMENSION
XIN(NI),OUT(NOUT),PAR(NP),YCHECK(NI),OCHECK(NOUT),T(ND),DTDT(ND),Stored(NStor
ed)
!---------------------------------------------------------------------------
-------------------------------------------
```

```
!-----------------------------------------------------------------------------
-------------------------------------------
!Declarations & Definitions for the User Variables
 INTEGER N_T,N_PL,NX,NY,LU_DATA,NVAL(2)
 DOUBLE PRECISION Q_Fluid,Q_Input,H_Cap,Eta,Control,X(2),T_TANK_IN,Y
 CHARACTER (LEN=maxMessageLength) MESSAGE1
!-----------------------------------------------------------------------------
-------------------------------------------

!-----------------------------------------------------------------------------
-------------------------------------------
!Get Global TRNSYS Simulation Variables
 TIME0=getSimulationStartTime()
 TFINAL=getSimulationStopTime()
 TimeStep=getSimulationTimeStep()
!-----------------------------------------------------------------------------
-------------------------------------------

!-----------------------------------------------------------------------------
-------------------------------------------
!Set the Version Information
 IF(INFO(7).eq.-2) THEN
    INFO(12)=16
    RETURN 1
 ENDIF
!-----------------------------------------------------------------------------
-------------------------------------------

!-----------------------------------------------------------------------------
-------------------------------------------
!Do All of the Very Last Call Manipulations Here
 IF (INFO(8).eq.-1) THEN
      RETURN 1
 ENDIF
!-----------------------------------------------------------------------------
-------------------------------------------

!-----------------------------------------------------------------------------
-------------------------------------------
!Do All of the After-Convergence Manipulations Here
 IF(INFO(13).GT.0) THEN
      RETURN 1
 ENDIF
!-----------------------------------------------------------------------------
-------------------------------------------

!-----------------------------------------------------------------------------
-------------------------------------------
!Do All of the First Call Manipulations Here
 IF (INFO(7).eq.-1) THEN

!  Get the Unit Number and Type Number
    IUNIT=INFO(1)
    ITYPE=INFO(2)
```

```
!  Set the INFO Array Variables to Tell TRNSYS How This Type Should Work
    INFO(6)=NOUT
    INFO(9)=1
    INFO(10)=0

!  Call the TYPECK Subroutine to Compare What This Component Wants to What is
Supplied in the Input File
    CALL TYPECK(1,INFO,NI,NP,ND)

!  Set the Variable Types for the Inputs and the Outputs
    DATA YCHECK/'PW1','CF1','TE1'/
    DATA OCHECK/'PW1','PW1','DM1'/

!  Call the RCHECK subroutine to set the correct input and output types
    CALL RCHECK(INFO,YCHECK,OCHECK)

!  Set the Size of the Storage Array
      CALL SetStorageSize(NStored,INFO)

!  Return to the TRNSYS Engine
    RETURN 1

 ENDIF
!-----------------------------------------------------------------------------
-------------------------------------------

!-----------------------------------------------------------------------------
-------------------------------------------
!DO ALL OF THE INITIAL TIMESTEP MANIPULATIONS HERE - THERE ARE NO ITERATIONS
AT THE INTIAL TIME
 IF (TIME.LT.(TIME0+TimeStep/2.D0)) THEN

!  Get the Unit Number and Type Number
    IUNIT=INFO(1)
    ITYPE=INFO(2)

!  Set the initial values of the Outputs
    OUT(1)=0.
    OUT(2)=0.

!  Read in the parameter values in sequential order
    LU_DATA=PAR(1)
    N_PL=PAR(2)
    N_T=PAR(3)

!  Return to the TRNSYS Engine
    RETURN 1

ENDIF
!-----------------------------------------------------------------------------
-------------------------------------------

!-----------------------------------------------------------------------------
-------------------------------------------
!    *** ITS AN ITERATIVE CALL TO THIS COMPONENT ***
```

```
!-----------------------------------------------------------------------------
-------------------------------------------

!-----------------------------------------------------------------------------
-------------------------------------------
!Reread the Parameters If Another Unit of this Type Has Been Called
  IF(INFO(1).NE.IUNIT) THEN

!  Get the Unit Number and Type Number
    IUNIT=INFO(1)
    ITYPE=INFO(2)

 ENDIF
!-----------------------------------------------------------------------------
-------------------------------------------

!-----------------------------------------------------------------------------
-------------------------------------------
!Get the Values of the Inputs to the Model at the Current Iteration
 H_Cap=XIN(1)                 !kJ/h
 Control=XIN(2)              !0..1
 T_TANK_IN=XIN(3)

!Check the inputs for problems
 IF(H_Cap<0.) CALL TYPECK(-3,INFO,1,0,0)
 IF(Control<0.) CALL TYPECK(-3,INFO,3,0,0)
 IF(Control>1.) CALL TYPECK(-3,INFO,3,0,0)
 IF(ErrorFound()) RETURN 1

!-----------------------------------------------------------------------------
-------------------------------------------

!-----------------------------------------------------------------------------
-------------------------------------------
!Perform all of the iterative call calculations here.
!       GET THE RATED CATALOG PERFORMANCE
         NX=2
         NVAL(2)=N_PL
         NVAL(1)=N_T
         NY=1
         X(2)=Control
         X(1)=T_TANK_IN
         CALL DYNAMICDATA(LU_DATA,NX,NVAL,NY,X,Y,INFO,*40)
         CALL LINKCK('TYPE 1228','DYNAMICDATA',1,1228)
40       IF(ErrorFound()) THEN
            CALL MESSAGES(-1,MESSAGE1,'FATAL',IUNIT,ITYPE)
            RETURN 1
         ENDIF
 Eta=Y
 Q_Fluid=H_Cap*Eta*Control
 Q_Input=H_Cap*Control

!-----------------------------------------------------------------------------
-------------------------------------------
```

```
!---------------------------------------------------------------------------
-------------------------------------------
!Set the Outputs from the Model
 OUT(1)=Q_Fluid
 OUT(2)=Q_Input
 OUT(3)=Eta
!---------------------------------------------------------------------------
-------------------------------------------

!---------------------------------------------------------------------------
-------------------------------------------
!Everything is Done at this Iteration, Return to the Engine
 RETURN 1
 END
!---------------------------------------------------------------------------
-------------------------------------------
```

Appendix F: Comparison between TRNSYS and Building America Benchmark Homes

In many ways, the home used here are very similar to the Benchmark home described in the Building America House Simulation Protocol (HSP) (51). However, there are several differences between these two homes. Because of these discrepancies the home here is considered “Benchmark-like” instead of being a Benchmark home. These discrepancies are:

- Monthly Lighting Schedules
- Natural Ventilation
- DHW Distribution System Gains
- Heating and Cooling Seasons
- Vacation Periods
- Windows in Warm Climates
- Space Conditioning Equipment: Part Load Performance
- Timestep Size
- Ducts
- Dehumidification
- Outside Convection Coefficient
- Basement Infiltration

Many of these differences came from difficulty including these features in TRNSYS. Ideally, a version of BEopt using the TRNSYS simulation engine would be created that has addressed these issues and can simulate Building America Benchmark buildings easily. However, this is a large undertaking and would require a separate project. The list of discrepancies provided here, as well as the detailed description of these differences provided below, is intended to aid any future work on this undertaking.

Several occupant behaviors were not simulated in TRNSYS (or a simplified version is included). The first of these is the use of monthly lighting schedules. In the HSP, different lighting use profiles are used each month, reflecting seasonal changes in lighting needs. In TRNSYS, an average lighting profile (shown in Figure 53) is used every day for all homes. In addition, there is no exterior lighting simulated in TRNSYS, as the focus was on space conditioning energy consumption only, not whole home energy consumption. Natural ventilation was also not included in TRNSYS, although it is included in the HSP for Benchmark homes if certain conditions are met. In addition, no domestic hot water distribution system gains were simulated (excluding the runs where the entire distribution system was simulated) in TRNSYS, while standard gains are prescribed in the HSP.

Other occupant behaviors not simulated include heating and cooling seasons and vacation periods. The HSP gives criteria for which months heating and cooling equipment should be enabled and which months it will be disabled. This does lead to some periods where the home's temperature is not kept within the temperature set by the thermostat. In TRNSYS, the heating and cooling equipment is always enabled, ensuring there are never any times where the home has an unmet heating or cooling load. Vacation periods in TRNSYS are only reflected the domestic hot water draw profile: during these periods, none of the internal gains change for the home. In the HSP, there are changes to the internal gains during these periods to reflect the lack of occupants.

Along with differences in how occupancy is modeled, there are differences in both the envelope and the space conditioning equipment. In the TRNSYS simulations, all of the windows were the same (U=0.35 Btu/h·ft^2·°F, SHGC=0.35) in all climates. In the HSP, this corresponds to a cold climate window: in warm climates, different performance (U=0.40 Btu/h·ft^2·°F, SHGC=0.30) is prescribed. New windows will need to be created in TRNSYS. This can be done with the WINDOW 6 Program (70), although even with this tool it is difficult to get precisely the U and SHGC values prescribed.

Another cause of differences between TRNSYS and BEopt is the modeling of space heating and cooling equipment. In BEopt, hourly simulations are run and part load performance curves are utilized to determine the performance of space conditioning equipment. TRNSYS uses smaller timesteps (either 6 seconds or 1 minute depending on the case and the number of runs required) and does not have any part load effects. BEopt is controlled by load set points while TRNSYS uses temperature set points. Performance of air conditioners and air source heat pumps in BEopt uses biquadratic performance curves, while TRNSYS uses a detailed performance map with linear interpolation between user provided data points. These differences can lead to disparities between the space conditioning energy consumption between the two simulation engines.

In addition to differences in how the space conditioning equipment is modeled, there are a few other differences that can impact the heating and cooling loads. No ducts are modeled in TRNSYS, although they are prescribed in the HSP. Ducts could be added to the TRNSYS simulations in the future using preexisting models (71). In addition, no dehumidification equipment is modeled in TRNSYS. This could also be added to future simulations with preexisting models. Finally, there are differences in how the outside convection coefficient for homes is modeled. In TRNSYS, one outside convection coefficient is modeled using the approach outlined in the ASHRAE fundamentals handbook (58). In the EnergyPlus implementation of BEopt, the outside convection coefficient is calculated for each surface using the MoWiTT algorithm (72). This could be added to TRNSYS as well, although it would require significantly more effort than adding ducts or dehumidification equipment. Finally, infiltration was added to basements in TRNSYS, even though this is not used in BEopt. This infiltration was added to ensure that the basement humidity for cases with a HPWH would never reach 0: in the HSP there are no moisture sources in the basement, and the HPWH acts as a moisture sink, so the humidity could reach 0 if no infiltration or other moisture source was added.

To determine how important all of these differences are when comparing the space conditioning energy consumption of a Benchmark home to the "Benchmark-like" home in TRNSYS, simulations were done using BEopt E+ (the EnergyPlus implementation of BEopt) of homes in all of the locations used in the parametric study of different water heaters. Whenever possible, the BEopt model was made to be consistent with the TRNSYS building, even if this meant deviating from the HSP. A list of the differences between the BEopt model and the TRNSYS model is provided below.

- Monthly Lighting Schedules
- DHW Distribution System Gains
- Vacation Periods
- Windows in Warm Climates
- Space Conditioning Equipment: Part Load Performance
- Timestep Size
- Outside Convection Coefficient

Note that difference in natural ventilation, heating and cooling seasons, ducts, and dehumidification were removed by having the BEopt simulations vary from the Benchmark guidelines. Even with these differences removed, significant variation in energy consumption between TRNSYS and BEopt exist. The heating, cooling, and fan energy consumption in each case is provided in Figure 119 and Table 82. In general, TRNSYS tends to over predict the energy consumption: it only under predicts the heating energy consumption in Los Angeles. The full impact of each of the differences listed above has not been explored. Future work would be required to rectify the differences between these two simulation engines and develop a TRNSYS implementation of the Benchmark building and/or BEopt.

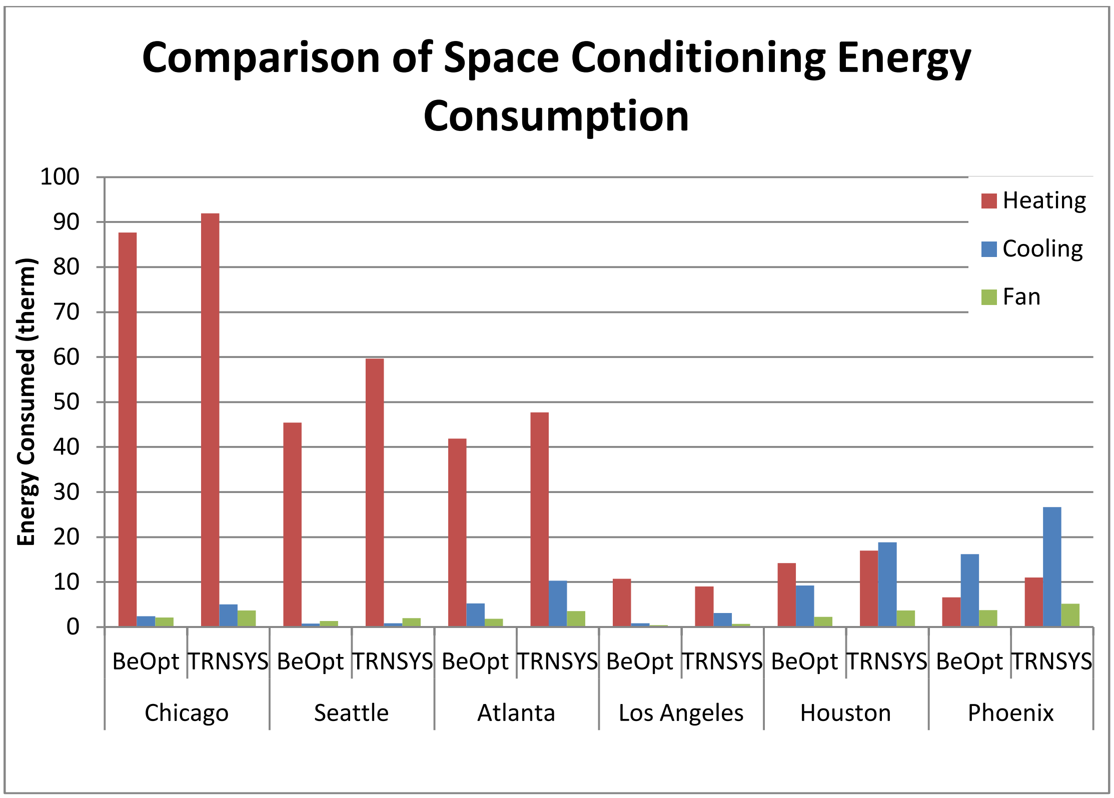


Figure 119: Comparison of space heating, cooling and fan energy consumption from BEopt E+ and TRNSYS

Location	Simulation Engine	Heating Energy	Cooling Energy	Fan Energy
Chicago	BeOpt	87.70	2.45	2.12
	TRNSYS	91.98	5.07	3.71
	% Difference	4.7%	51.8%	42.9%
Seattle	BeOpt	45.42	0.75	1.36
	TRNSYS	59.64	0.85	2.00
	% Difference	23.8%	11.7%	32.2%
Atlanta	BeOpt	41.85	5.25	1.87
	TRNSYS	47.68	10.33	3.55
	% Difference	12.2%	49.2%	47.2%
Los Angeles	BeOpt	10.71	0.86	0.42
	TRNSYS	9.03	3.12	0.74
	% Difference	-18.6%	72.5%	43.7%
Houston	BeOpt	14.23	9.21	2.27
	TRNSYS	16.98	18.82	3.69
	% Difference	16.2%	51.1%	38.5%
Phoenix	BeOpt	6.64	16.23	3.76
	TRNSYS	11.01	26.68	5.16
	% Difference	39.7%	39.2%	27.2%

Table 82: Space heating, cooling and fan energy consumption in BEopt E+ and TRNSYS

Appendix G: Impact of a Pilot Light on Gas Water Heater Performance

To determine the impact of the pilot on the annual energy consumption of a gas water heater (either used by itself or as a backup for a solar water heater), simulations were performed both with and without a standing pilot light. The pilot light modeled here consumes 750 Btu/hr, a typical size for a gas water heater (73), and has the same efficiency as the conversion efficiency of the water heater (82.3% for this particular water heater). Simulations were run for all draw profiles, climates, and installation locations. The inclusion of the pilot light had a very minimal impact on the gas water heater annual energy consumption in most cases but had a significant impact on the annual energy consumption of solar water heating systems using a gas water heater.

Adding a standing pilot has two major impacts on the annual energy consumption of a gas water heater. Since the standing pilot is running continuously for the full year (8760 hours) and consuming 750 Btu/hr, for all gas water heaters there is a minimum energy consumption of 6570000 Btu/yr (65.7 therm/yr). However, this energy is not wasted in a regular gas storage water heater. 617 Btu/hr goes into the tank from the pilot light (617 Btu/hr is the product of the pilot energy consumption and the conversion efficiency). This energy addition into the tank helps offset standby losses. To help illustrate this, consider an energy balance on the water heater during times where there is no energy from either the pilot or the burner added into the tank. Using the UA value of this gas water heater and assuming the tank is isothermal at 120 °F (48.89 °C) in a 70 °F (21.11 °C) room, the standby losses will be:

$$\dot{Q} = UA\Delta T \rightarrow \dot{Q} = \left(0.46\ \frac{Btu}{hr \cdot ft^2 \cdot {}^\circ F}\right)(20.48\ ft^2)(50\ {}^\circ F) = 471\ \frac{Btu}{hr}$$

This means that the pilot light will slightly heat the tank if the water heater is in conditioned space. Using the previous equation with the pilot light power and solving for the temperature difference this power input can sustain yields:

$$\dot{Q} = UA\Delta T \rightarrow 671\,\frac{Btu}{hr} = \left(0.46\ \frac{Btu}{hr\cdot ft^2\cdot {}^{\circ}F}\right)(20.48\ ft^2)(\Delta T) \rightarrow \Delta T = 71.2^{\circ}F$$

This means the pilot light can keep the tank roughly 70 °F warmer than the surrounding air. However, this is a simple steady state analysis that only applies during periods of no draw. After each draw, the water heater needs to recover using the burner if a large enough volume is drawn. If the water heater is in conditioned space, this can lead to slightly larger energy consumption by the water heater as shown in Figure 120. However, this can increase the energy consumption of a gas water heater by as much as 19% in the case of installing a gas water heater in unconditioned space in a hot climate (Phoenix) with very low use as the tank will heat to a temperature over the set point during periods of standby when the ambient air is hot. The increase in energy consumption will be smaller for water heaters in unconditioned space in cold climates as the tank losses are generally larger due to the lower average ambient air temperature. On average across all cases, adding a pilot light increases the annual energy consumption by 4.43 therms or 3.6% over the base case. The additional energy consumption associated with including a pilot light in the gas water heater model is shown in Figure 121 and Figure 122 for a gas water heater in conditioned and unconditioned space respectively.

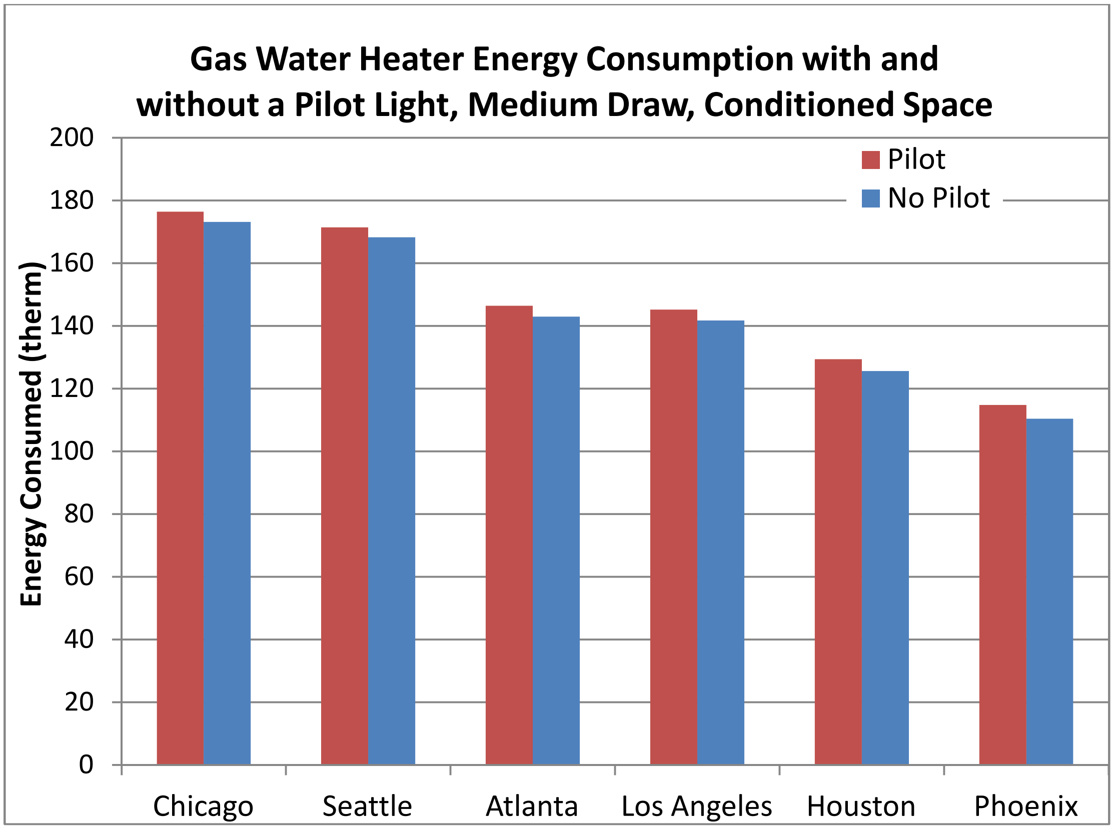


Figure 120: Additional energy consumed for a gas water heater with a pilot light in conditioned space

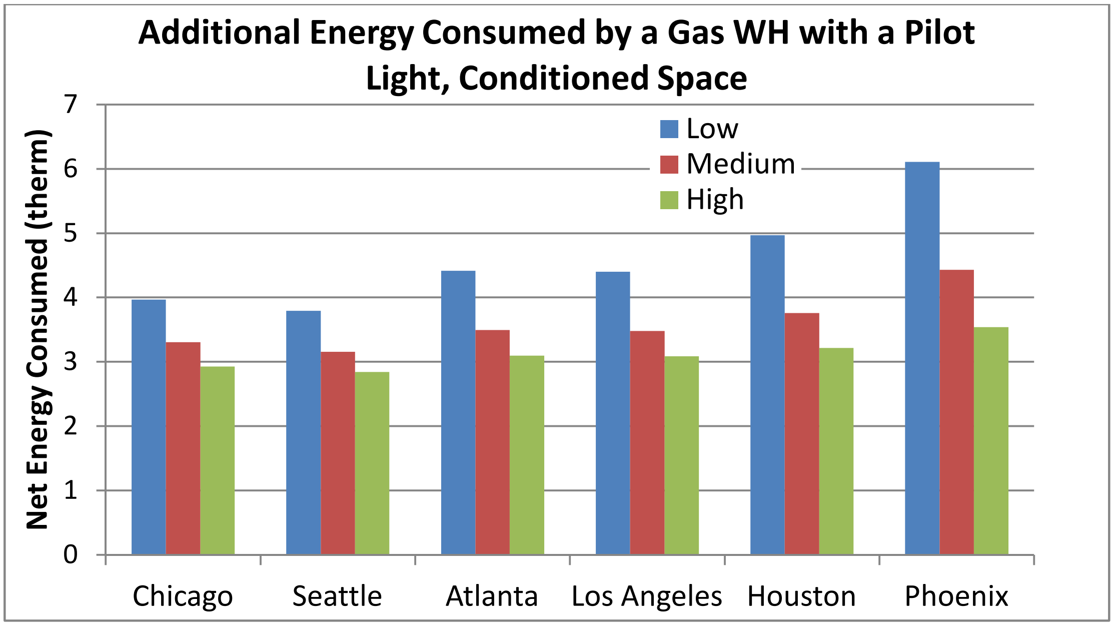


Figure 121: Additional energy consumed for a gas water heater with a pilot light in conditioned space

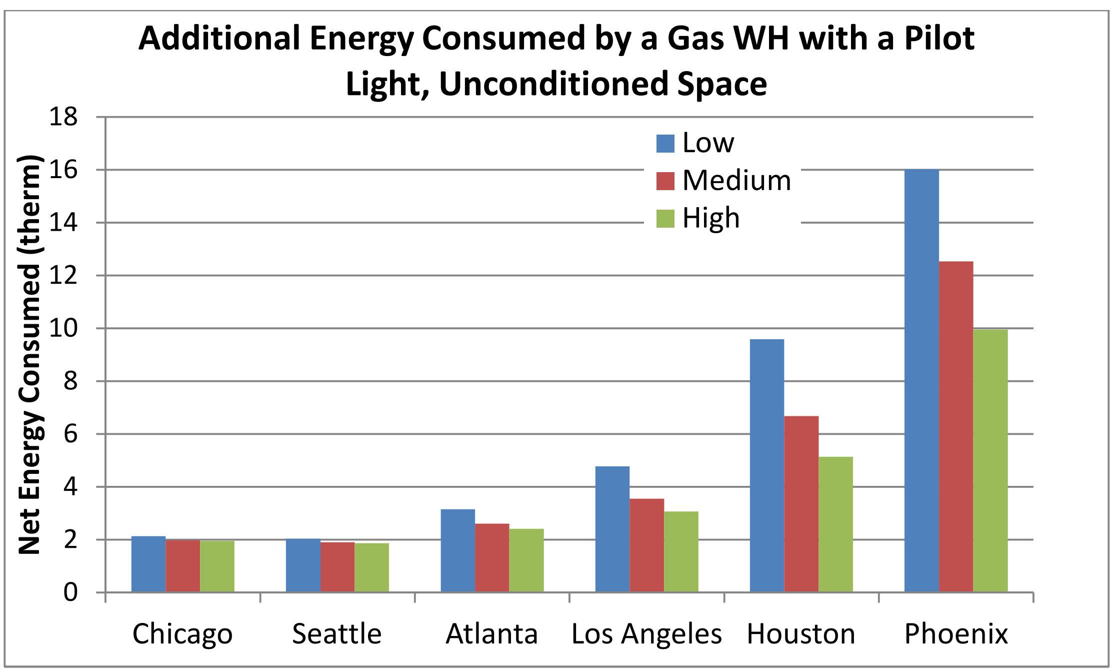


Figure 122: Additional energy consumed for a gas water heater with a pilot light in unconditioned space

For the case where a gas storage water heater is used as the backup to a solar water in a two tank system, the difference in energy consumption can be much larger if a standing pilot light is included. Since in this case the gas water heater will get preheated water from the solar storage tank, the gas water heater without a pilot light may consume significantly less energy than the amount the pilot light will consume over the course of a year (65.7 therms/yr) if there is a significant solar resource. For example, in Phoenix a gas water heater without a pilot light in conditioned space under average use will only consume 31.4 therm/yr. This means the energy consumption of the gas water heater will increase by at least 48% if a standing pilot is used. In addition, the pilot light can heat the gas water heater over its set point during periods where there is no draw, further increasing the gas water heater energy consumption. In locations with a relatively low amount of solar energy available (Seattle is a particularly good example) the impact of the pilot light is only slightly larger than in the case of a regular gas storage water heater as shown in Figure 123. However, given the large impact of a pilot light in the locations where solar water heaters are most attractive, it is recommended that for two tank solar water heaters using a gas water heater as a backup that a gas water heater that does not have a standing pilot light is used. The additional energy consumption of a gas water heater with a pilot light for solar two tank systems in conditioned and unconditioned space is given in Figure 124 and Figure 125 respectively.

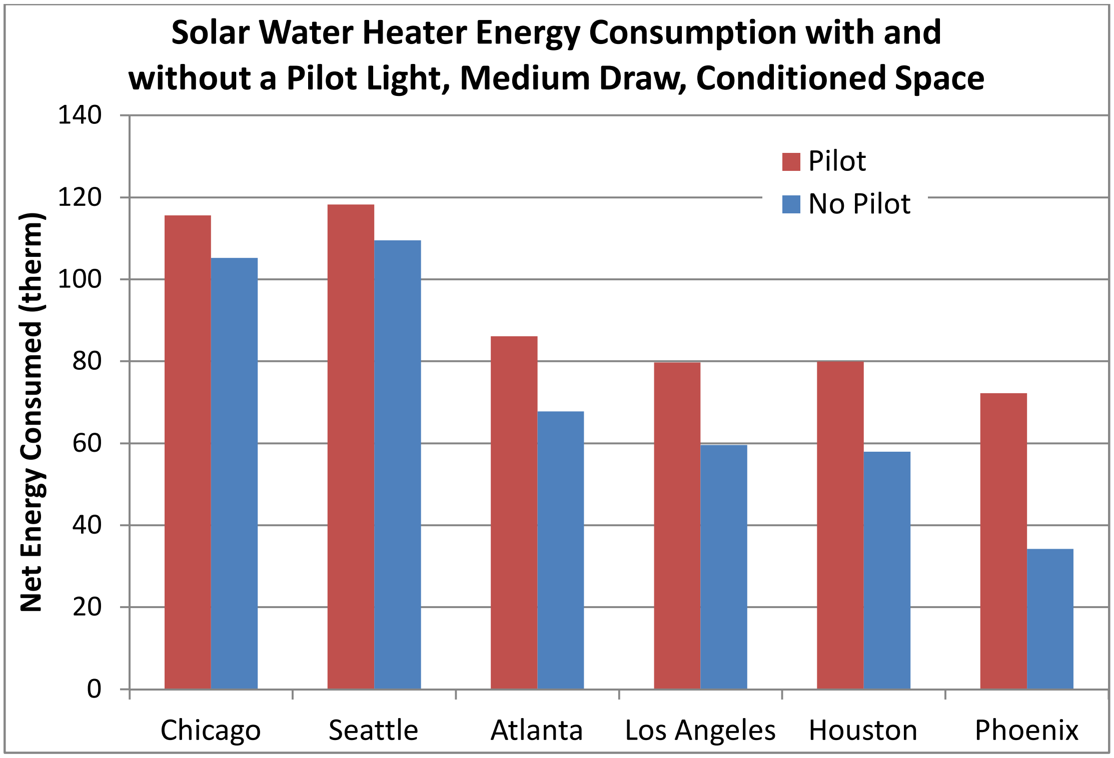


Figure 123: Comparison of energy consumed for a solar water heater with gas backup with and without a pilot light

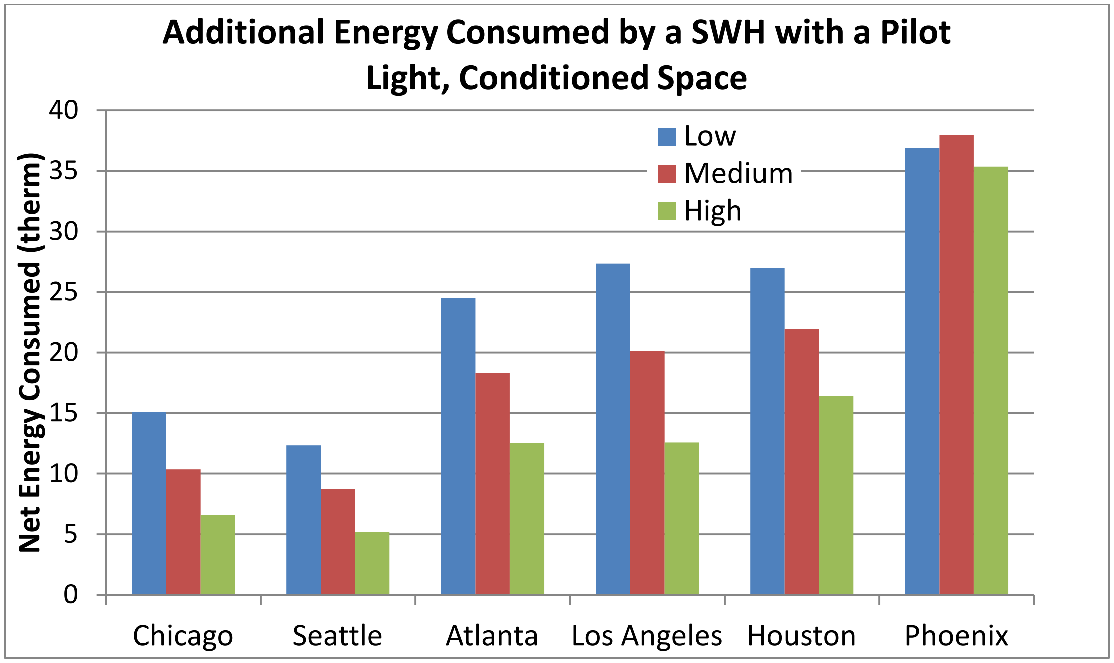


Figure 124: Additional energy consumed by a solar water heater with a pilot light in conditioned space

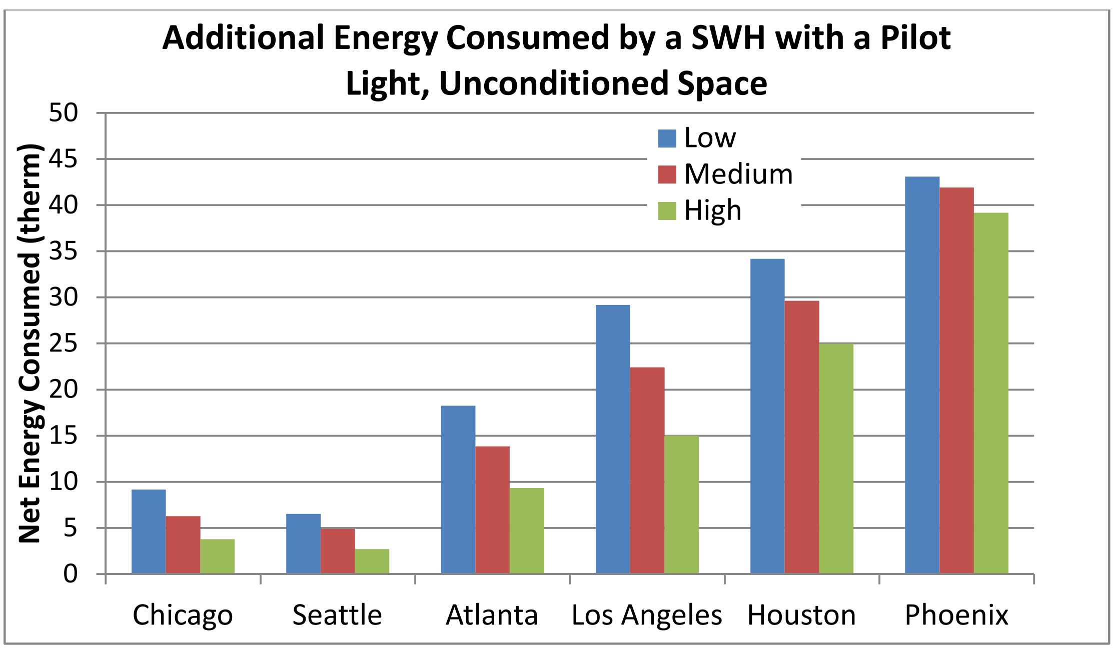


Figure 125: Additional energy consumed by a solar water heater with a pilot light in unconditioned space

Appendix H: Water Heater Draw Volume in gal/day for All Simulations in Parametric Study

Location	Installation Space	Draw Volume	Gas	Tankless	Condensing	Solar Gas	Elec	HPWH	Solar Elec
Chicago	**Cond**	**Low**	36.6	38.0	37.1	36.3	35.1	38.8	35.5
Chicago	**Cond**	**Med**	52.2	54.2	53.0	52.0	50.0	55.4	51.4
Chicago	**Cond**	**High**	70.4	73.2	71.9	70.2	67.9	75.9	70.8
Chicago	**Uncond**	**Low**	36.7	38.2	37.1	36.4	35.2	38.7	35.6
Chicago	**Uncond**	**Med**	52.4	54.5	53.1	52.1	50.1	55.2	51.5
Chicago	**Uncond**	**High**	70.6	73.5	71.8	70.4	67.9	75.7	70.9
Seattle	**Cond**	**Low**	36.4	37.9	36.9	36.1	34.8	38.5	35.6
Seattle	**Cond**	**Med**	51.9	54.1	52.7	51.7	49.6	55.0	51.3
Seattle	**Cond**	**High**	70.0	73.1	71.4	69.8	67.3	75.4	70.4
Seattle	**Uncond**	**Low**	36.4	38.1	36.9	36.2	34.8	38.6	35.7
Seattle	**Uncond**	**Med**	52.0	54.3	52.8	51.8	49.7	55.2	51.4
Seattle	**Uncond**	**High**	70.1	73.4	71.4	69.9	67.4	75.8	70.5
Atlanta	**Cond**	**Low**	34.3	36.4	34.9	33.6	32.6	36.6	31.3
Atlanta	**Cond**	**Med**	48.9	51.8	49.8	48.4	46.4	52.2	45.9
Atlanta	**Cond**	**High**	65.8	69.8	67.3	65.6	62.7	71.5	64.1
Atlanta	**Uncond**	**Low**	34.4	36.5	34.9	33.8	32.6	36.6	31.5
Atlanta	**Uncond**	**Med**	49.0	52.0	49.8	48.5	46.4	52.4	46.1
Atlanta	**Uncond**	**High**	65.9	70.1	67.3	65.7	62.8	71.7	64.2
Los Angeles	**Cond**	**Low**	34.4	36.4	34.9	33.6	32.6	36.6	31.0
Los Angeles	**Cond**	**Med**	49.0	52.0	49.9	48.6	46.5	52.3	46.3
Los Angeles	**Cond**	**High**	66.0	70.0	67.6	65.8	63.0	71.7	64.6
Los Angeles	**Uncond**	**Low**	34.4	36.4	34.9	33.5	32.6	36.7	31.1
Los Angeles	**Uncond**	**Med**	49.0	51.9	49.9	48.4	46.5	52.5	46.4
Los Angeles	**Uncond**	**High**	66.0	70.0	67.6	65.7	63.0	72.0	64.7
Houston	**Cond**	**Low**	32.5	34.9	33.2	31.8	30.7	34.8	29.0
Houston	**Cond**	**Med**	46.3	49.6	47.3	45.7	43.6	49.5	42.6
Houston	**Cond**	**High**	62.2	66.8	63.7	61.8	58.9	67.8	59.3
Houston	**Uncond**	**Low**	32.4	34.6	33.1	31.2	30.7	34.8	29.0
Houston	**Uncond**	**Med**	46.1	49.2	47.1	44.6	43.6	49.6	42.6
Houston	**Uncond**	**High**	62.0	66.4	63.6	60.7	58.9	67.9	59.2
Phoenix	**Cond**	**Low**	29.6	32.4	30.2	28.3	27.7	31.7	24.7
Phoenix	**Cond**	**Med**	41.8	45.9	42.9	40.0	39.2	45.0	35.7
Phoenix	**Cond**	**High**	56.1	61.6	57.7	54.1	52.7	61.3	49.6
Phoenix	**Uncond**	**Low**	29.4	31.8	30.1	27.1	27.7	31.8	24.7
Phoenix	**Uncond**	**Med**	41.6	45.1	42.6	37.9	39.2	45.1	35.7
Phoenix	**Uncond**	**High**	55.8	60.7	57.3	51.0	52.7	61.6	49.5

Appendix I: Annual Water Heater Operating Costs

Water Heater Annual Operating Costs in 2011$

Location	WH Loc	Draw	Gas	Tankless	Condensing	Solar Gas	Elec	HPWH	Solar Elec
Chicago	**Cond**	**Low**	169	117	134	96	288	129	136
Chicago	**Cond**	**Med**	217	164	172	129	394	175	215
Chicago	**Cond**	**High**	272	219	216	174	515	256	320
Chicago	**Uncond**	**Low**	187	121	144	111	304	194	145
Chicago	**Uncond**	**Med**	235	167	182	143	410	260	226
Chicago	**Uncond**	**High**	290	221	227	188	531	352	331
Seattle	**Cond**	**Low**	175	118	135	112	229	100	116
Seattle	**Cond**	**Med**	223	165	173	148	311	133	183
Seattle	**Cond**	**High**	279	221	218	195	408	196	266
Seattle	**Uncond**	**Low**	193	119	145	128	239	118	123
Seattle	**Uncond**	**Med**	240	166	183	163	322	158	190
Seattle	**Uncond**	**High**	296	222	228	211	418	226	274
Atlanta	**Cond**	**Low**	204	127	152	91	201	83	59
Atlanta	**Cond**	**Med**	256	179	194	117	271	106	100
Atlanta	**Cond**	**High**	316	238	242	162	354	153	166
Atlanta	**Uncond**	**Low**	218	127	160	100	209	94	62
Atlanta	**Uncond**	**Med**	270	179	201	127	278	120	104
Atlanta	**Uncond**	**High**	330	239	250	171	361	171	170
Los Angeles	**Cond**	**Low**	141	97	117	64	298	118	68
Los Angeles	**Cond**	**Med**	178	133	146	82	401	149	126
Los Angeles	**Cond**	**High**	220	176	180	112	523	217	221
Los Angeles	**Uncond**	**Low**	141	97	117	50	305	134	71
Los Angeles	**Uncond**	**Med**	177	133	146	66	408	174	131
Los Angeles	**Uncond**	**High**	220	176	180	96	530	251	226
Houston	**Cond**	**Low**	151	94	119	73	229	88	63
Houston	**Cond**	**Med**	187	130	148	88	305	110	102
Houston	**Cond**	**High**	229	172	181	117	396	156	170
Houston	**Uncond**	**Low**	139	94	112	51	230	97	65
Houston	**Uncond**	**Med**	175	130	141	64	306	122	104
Houston	**Uncond**	**High**	216	171	174	90	396	172	172
Phoenix	**Cond**	**Low**	179	98	131	65	157	66	24
Phoenix	**Cond**	**Med**	215	135	160	67	205	81	40
Phoenix	**Cond**	**High**	259	178	195	84	263	108	72
Phoenix	**Uncond**	**Low**	155	97	118	47	152	69	25
Phoenix	**Uncond**	**Med**	192	134	147	54	200	87	41
Phoenix	**Uncond**	**High**	235	177	182	74	258	116	73

Water Heater Annual Normalization Costs in 2011$

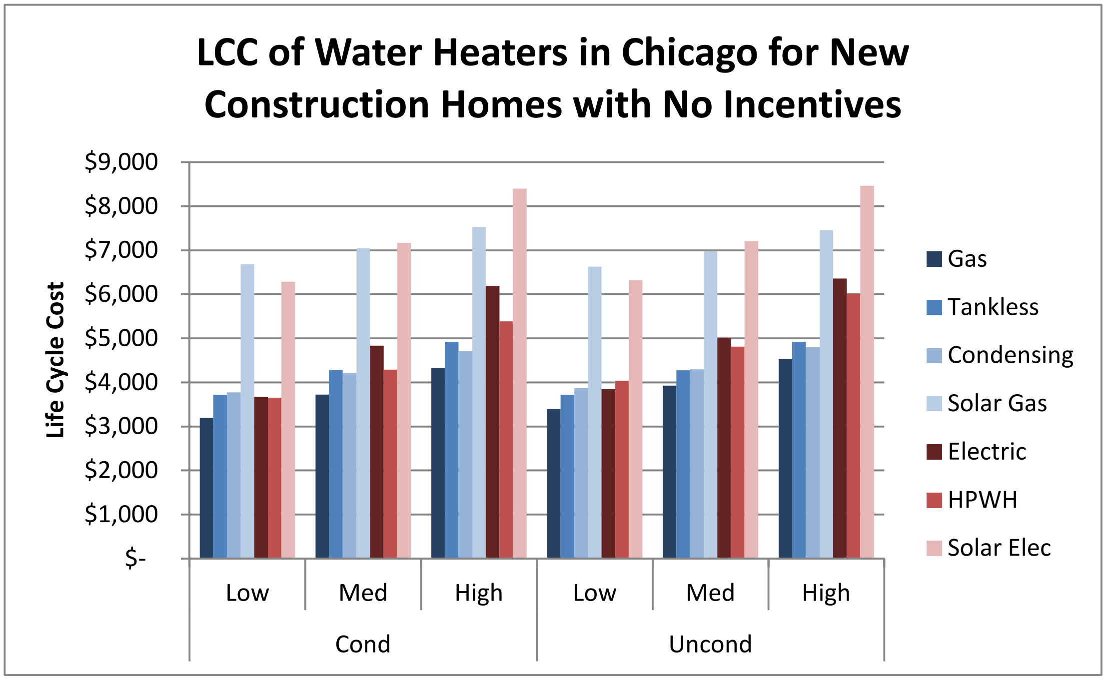
Location	WH Loc	Draw	Gas	Tankless	Condensing	Solar Gas	Elec	HPWH	Solar Elec
Chicago	**Cond**	**Low**	0.59	10.90	1.55	0.35	1.33	14.13	3.92
Chicago	**Cond**	**Med**	0.88	15.11	2.28	0.38	1.47	17.72	5.39
Chicago	**Cond**	**High**	1.14	19.31	3.97	0.48	3.83	35.98	13.79
Chicago	**Uncond**	**Low**	0.69	13.83	1.44	0.38	1.42	14.02	4.21
Chicago	**Uncond**	**Med**	0.96	18.89	2.38	0.51	1.49	18.19	5.75
Chicago	**Uncond**	**High**	1.29	23.75	3.57	0.67	3.91	35.99	14.37
Seattle	**Cond**	**Low**	0.57	11.45	1.59	0.39	0.62	10.72	3.35
Seattle	**Cond**	**Med**	0.81	15.78	2.53	0.41	0.92	13.09	4.60
Seattle	**Cond**	**High**	1.16	20.04	4.10	0.57	2.89	27.68	10.90
Seattle	**Uncond**	**Low**	0.72	14.33	1.60	0.48	0.67	11.92	3.61
Seattle	**Uncond**	**Med**	0.97	19.43	2.45	0.51	0.93	15.43	4.80
Seattle	**Uncond**	**High**	1.29	24.38	3.69	0.76	2.94	31.17	11.31
Atlanta	**Cond**	**Low**	0.56	14.19	1.79	0.32	0.16	8.88	0.83
Atlanta	**Cond**	**Med**	0.85	19.29	2.59	0.46	0.54	10.81	1.05
Atlanta	**Cond**	**High**	0.68	24.24	3.87	0.47	0.94	21.61	4.01
Atlanta	**Uncond**	**Low**	0.72	16.48	1.76	0.39	0.18	9.52	0.88
Atlanta	**Uncond**	**Med**	0.99	22.22	2.52	0.51	0.55	12.45	1.15
Atlanta	**Uncond**	**High**	0.81	27.69	3.81	0.58	0.97	23.81	4.15
Los Angeles	**Cond**	**Low**	0.42	9.52	1.16	0.28	0.09	12.44	0.68
Los Angeles	**Cond**	**Med**	0.60	12.90	1.74	0.45	0.73	15.52	1.48
Los Angeles	**Cond**	**High**	0.53	16.26	2.66	0.51	1.87	31.13	5.58
Los Angeles	**Uncond**	**Low**	0.38	9.50	1.24	0.29	0.09	14.22	0.72
Los Angeles	**Uncond**	**Med**	0.61	12.81	1.82	0.42	0.74	18.28	1.51
Los Angeles	**Uncond**	**High**	0.53	16.15	2.80	0.46	1.89	36.10	5.71
Houston	**Cond**	**Low**	0.40	10.83	1.33	0.37	0.03	9.13	0.67
Houston	**Cond**	**Med**	0.53	14.54	1.90	0.51	0.43	10.98	0.94
Houston	**Cond**	**High**	0.37	18.17	2.33	0.53	0.45	20.91	2.91
Houston	**Uncond**	**Low**	0.34	9.25	1.06	0.26	0.04	9.87	0.71
Houston	**Uncond**	**Med**	0.47	12.39	1.64	0.36	0.43	12.52	0.97
Houston	**Uncond**	**High**	0.33	15.83	2.25	0.36	0.45	23.01	2.99
Phoenix	**Cond**	**Low**	0.44	13.78	1.57	0.37	0.02	6.32	0.23
Phoenix	**Cond**	**Med**	0.47	18.27	2.14	0.42	0.07	7.97	0.28
Phoenix	**Cond**	**High**	0.43	22.61	2.71	0.47	0.11	13.64	0.83
Phoenix	**Uncond**	**Low**	0.39	11.27	1.40	0.26	0.02	7.11	0.25
Phoenix	**Uncond**	**Med**	0.39	14.98	1.69	0.32	0.07	9.35	0.29
Phoenix	**Uncond**	**High**	0.36	18.90	2.12	0.37	0.12	16.32	0.88

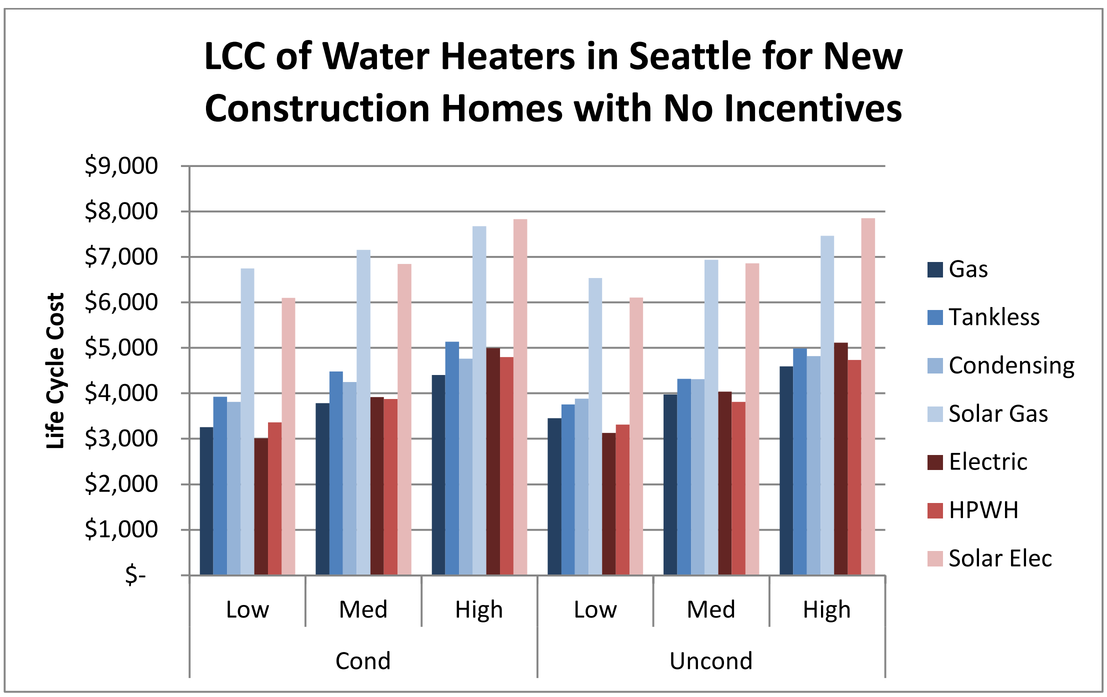
Water Heater Annual Costs for Change in Space Conditioning Energy Consumption in 2011$. A negative value denotes savings relative to the base case

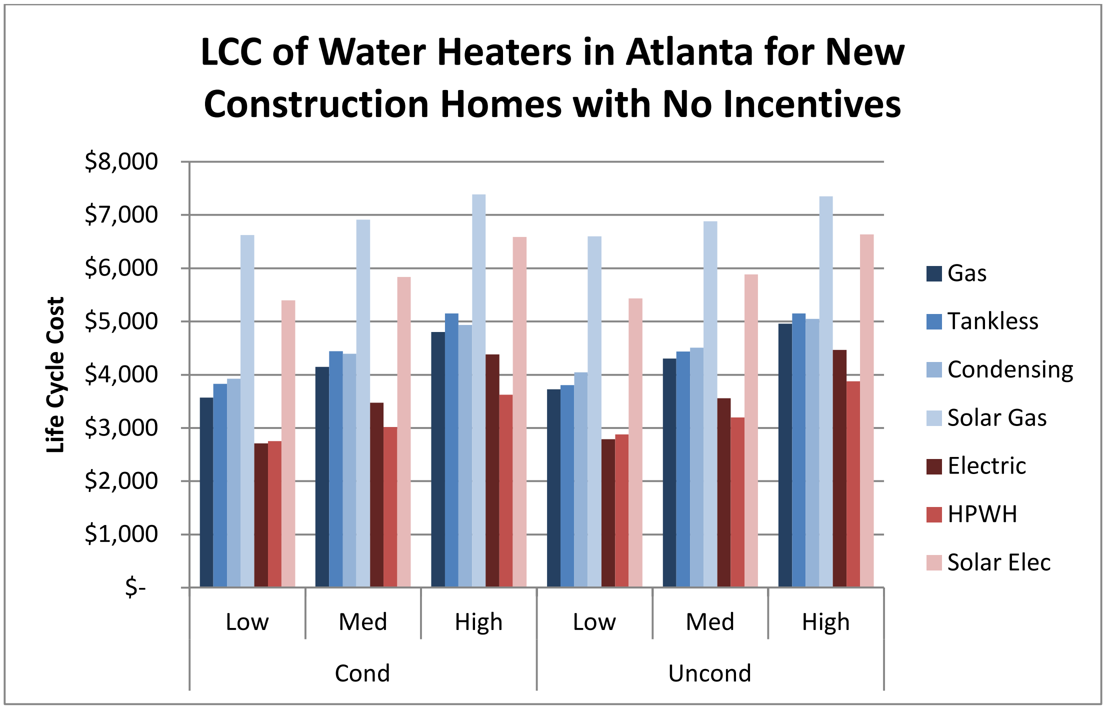
Location	WH Loc	Draw	Gas	Tankless	Condensing	Solar Gas	Elec	HPWH	Solar Elec
Chicago	**Cond**	**Low**	0	9.75	1.92	-4.61	0	21.32	-1.16
Chicago	**Cond**	**Med**	0	10.11	2.36	-4.20	0	30.38	-1.71
Chicago	**Cond**	**High**	0	9.73	2.52	-5.03	0	31.70	-1.84
Chicago	**Uncond**	**Low**	0	3.48	0.23	-24.22	0	-7.29	-7.22
Chicago	**Uncond**	**Med**	0	2.87	0.29	-24.74	0	-7.01	-8.05
Chicago	**Uncond**	**High**	0	2.90	0.48	-25.39	0	-7.92	-7.50
Seattle	**Cond**	**Low**	0	27.14	4.45	-13.56	0	27.81	2.30
Seattle	**Cond**	**Med**	0	26.86	5.72	-12.64	0	39.40	2.59
Seattle	**Cond**	**High**	0	26.63	5.91	-12.32	0	45.58	2.69
Seattle	**Uncond**	**Low**	0	8.34	1.30	-49.50	0	4.06	-4.16
Seattle	**Uncond**	**Med**	0	8.13	1.32	-48.02	0	5.93	-3.61
Seattle	**Uncond**	**High**	0	7.92	1.54	-47.33	0	7.08	-3.25
Atlanta	**Cond**	**Low**	0	6.82	-2.39	-4.25	0	-8.80	-2.29
Atlanta	**Cond**	**Med**	0	6.28	-2.30	-4.86	0	-9.77	-3.00
Atlanta	**Cond**	**High**	0	6.29	-2.20	-5.66	0	-12.10	-3.33
Atlanta	**Uncond**	**Low**	0	2.30	0.49	-15.79	0	-8.37	-2.16
Atlanta	**Uncond**	**Med**	0	2.24	0.59	-16.42	0	-8.99	-2.96
Atlanta	**Uncond**	**High**	0	2.21	0.44	-17.53	0	-9.41	-3.56
Los Angeles	**Cond**	**Low**	0	-8.18	-0.54	22.35	0	17.60	3.65
Los Angeles	**Cond**	**Med**	0	-7.98	-0.79	18.12	0	25.59	3.20
Los Angeles	**Cond**	**High**	0	-7.65	-1.01	16.18	0	31.62	2.92
Los Angeles	**Uncond**	**Low**	0	0.16	0.01	1.13	0	0.23	-0.12
Los Angeles	**Uncond**	**Med**	0	0.14	-0.02	0.89	0	-0.12	-0.23
Los Angeles	**Uncond**	**High**	0	0.15	-0.02	0.90	0	0.40	-0.21
Houston	**Cond**	**Low**	0	-17.03	-1.60	30.87	0	-16.68	7.71
Houston	**Cond**	**Med**	0	-16.64	-2.28	27.00	0	-26.75	7.09
Houston	**Cond**	**High**	0	-16.14	-3.03	24.77	0	-29.67	6.28
Houston	**Uncond**	**Low**	0	-0.72	-0.23	0.88	0	-0.02	14.02
Houston	**Uncond**	**Med**	0	-0.74	-0.26	0.82	0	-0.63	13.17
Houston	**Uncond**	**High**	0	-0.71	-0.28	0.62	0	-0.84	12.46
Phoenix	**Cond**	**Low**	0	-7.74	0.00	26.74	0	-13.05	6.51
Phoenix	**Cond**	**Med**	0	-7.30	-0.46	27.20	0	-18.53	7.08
Phoenix	**Cond**	**High**	0	-6.92	-0.53	28.40	0	-23.16	7.69
Phoenix	**Uncond**	**Low**	0	0.13	-0.17	0.46	0	0.10	12.85
Phoenix	**Uncond**	**Med**	0	0.00	-0.19	0.59	0	-0.02	13.16
Phoenix	**Uncond**	**High**	0	-0.10	-0.16	0.53	0	-0.33	13.44

Appendix J: Life Cycle Costs for All Simulations in Parametric Study

New Construction, No Incentives



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Cond
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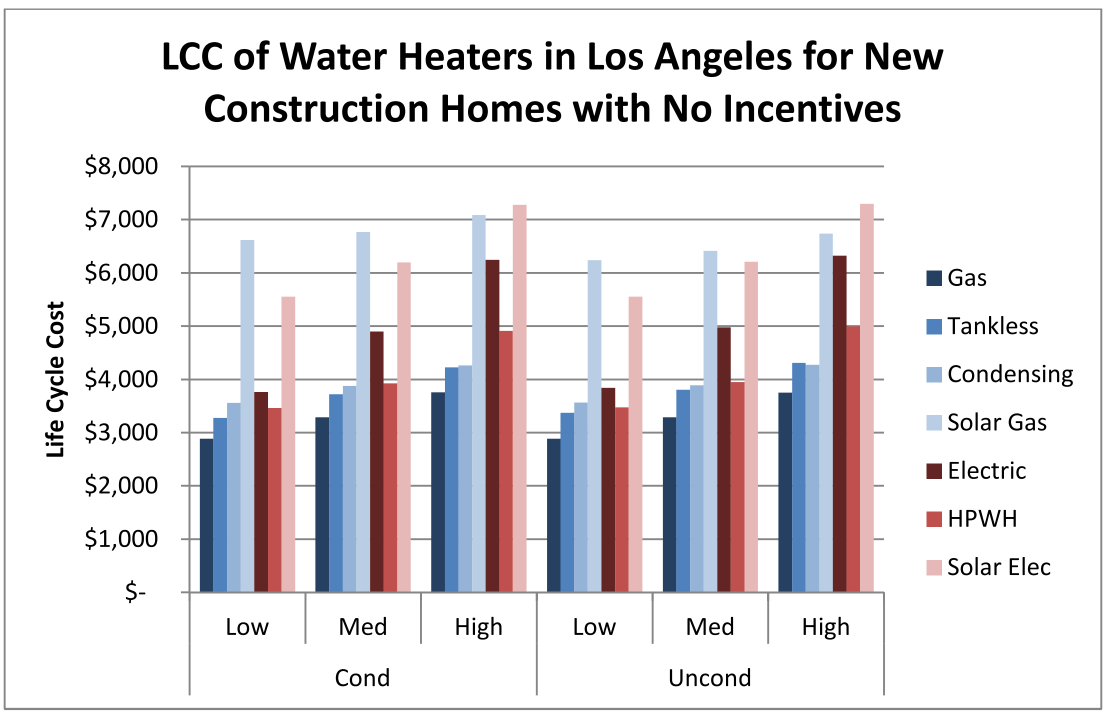
Gas
Tankless
Condensing
Solar Gas
Electric
HPWH
Solar Elec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$4,000
$3,000
$2,000
$1,000
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Med
High
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Med
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Cond
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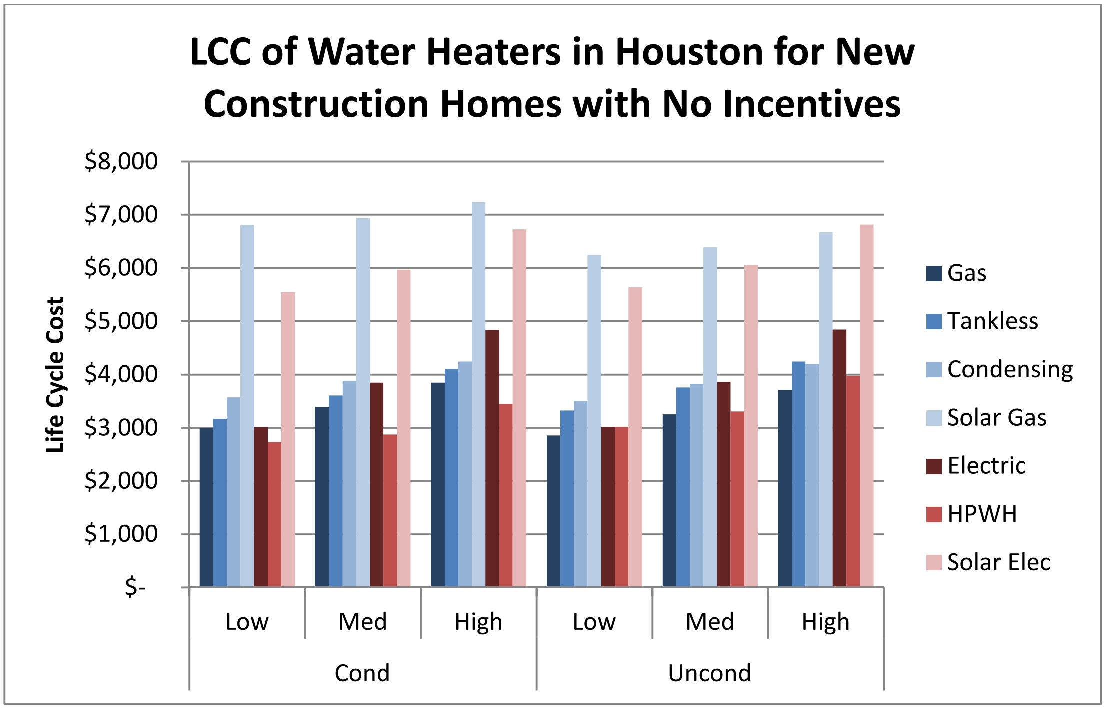
Gas
Tankless
Condensing
Solar Gas
Electric
HPWH
Solar Elec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Life Cycle Cost
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$4,000
$3,000
$2,000
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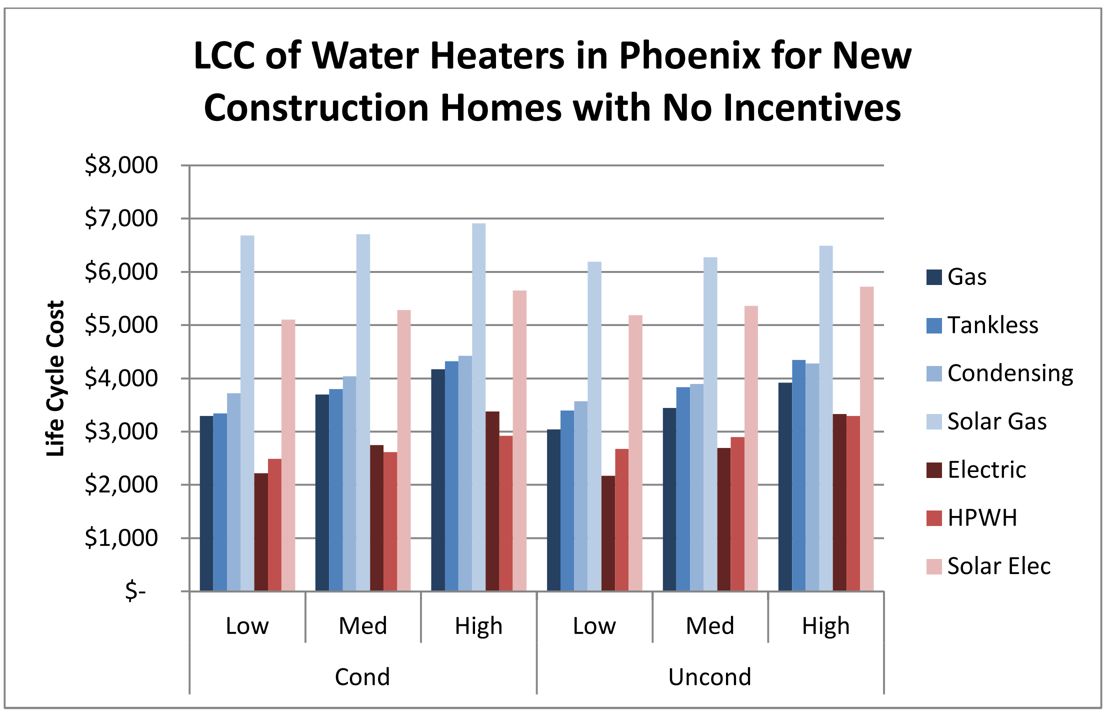
HPWH
Solar Elec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$4,000
$3,000
$2,000
$1,000
$-
Low
Med
High
Low
Med
High
Cond
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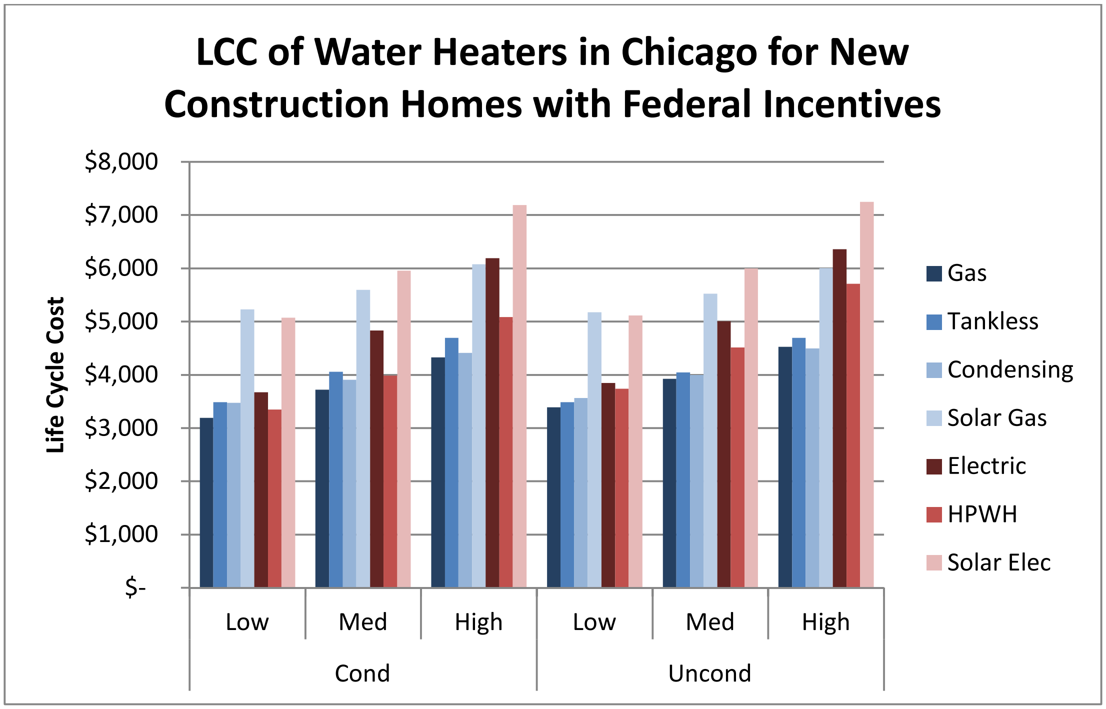
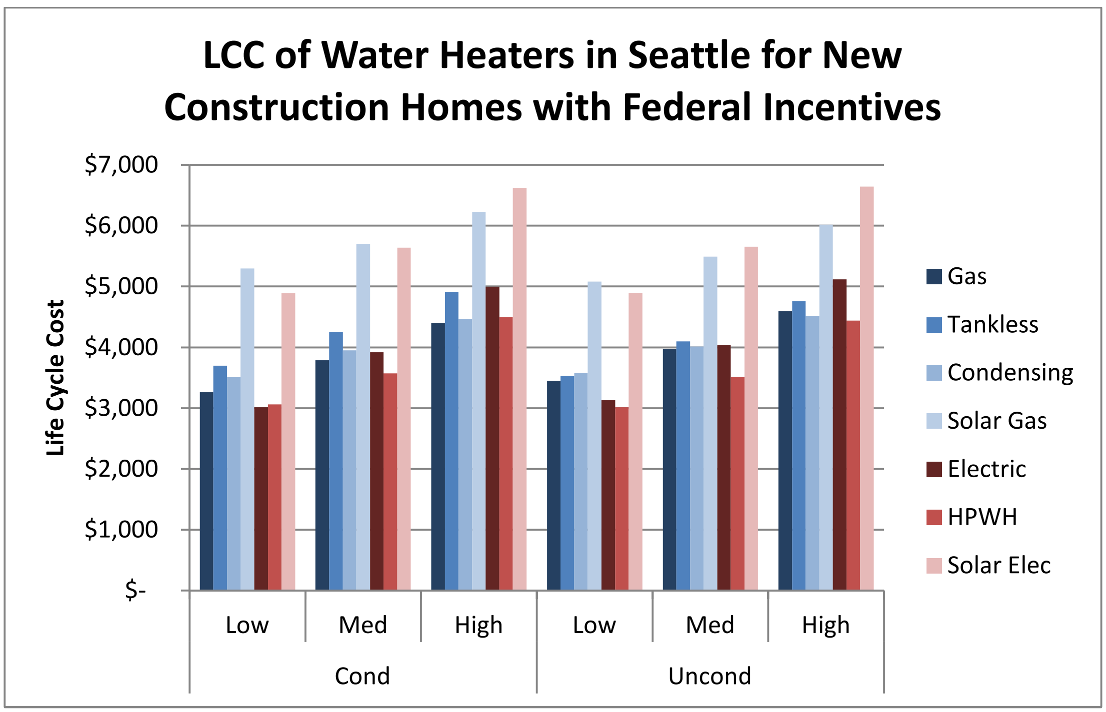
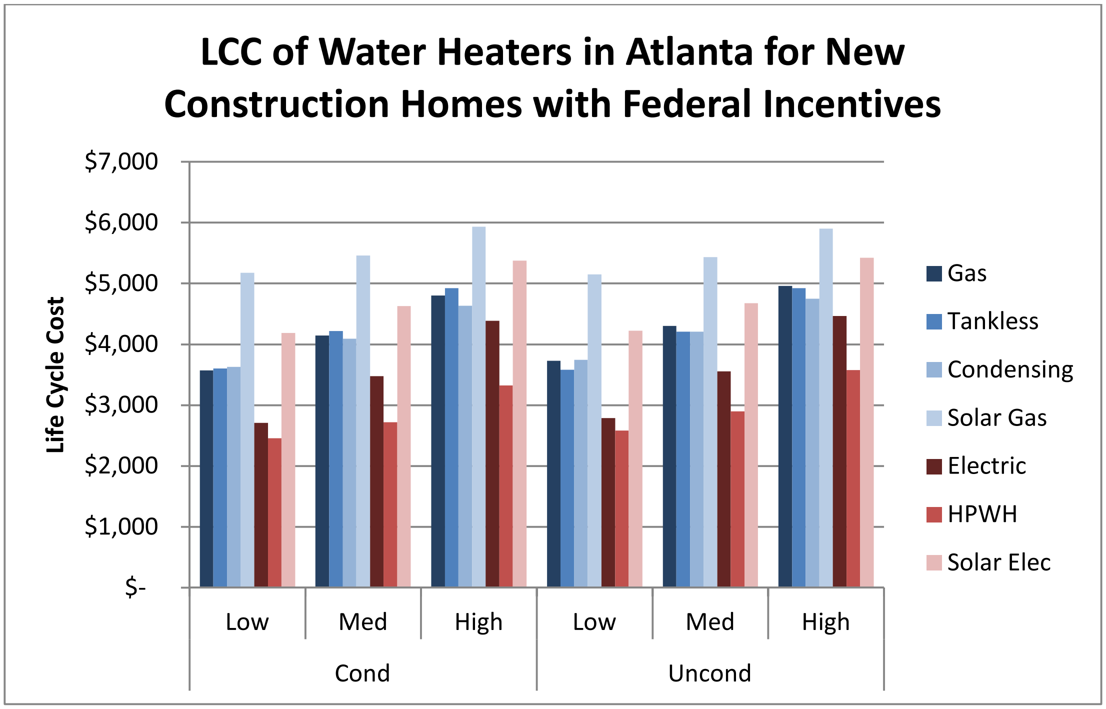
Condensing
Solar Gas
Electric
HPWH
Solar Elec

New Construction, Federal Incentives



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Life Cycle Cost
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$5,000
$4,000
$3,000
$2,000
$1,000
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Low
Med
High
Low
Med
High
Cond
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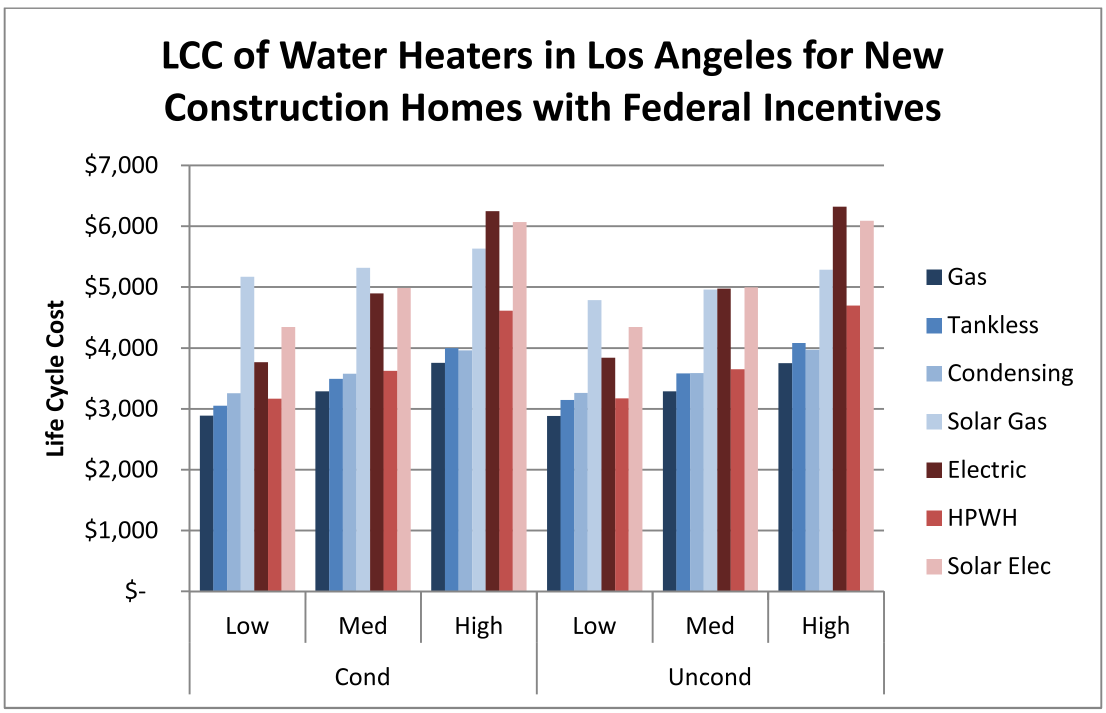
Gas
Tankless
Condensing
Solar Gas
Electric
HPWH
Solar Elec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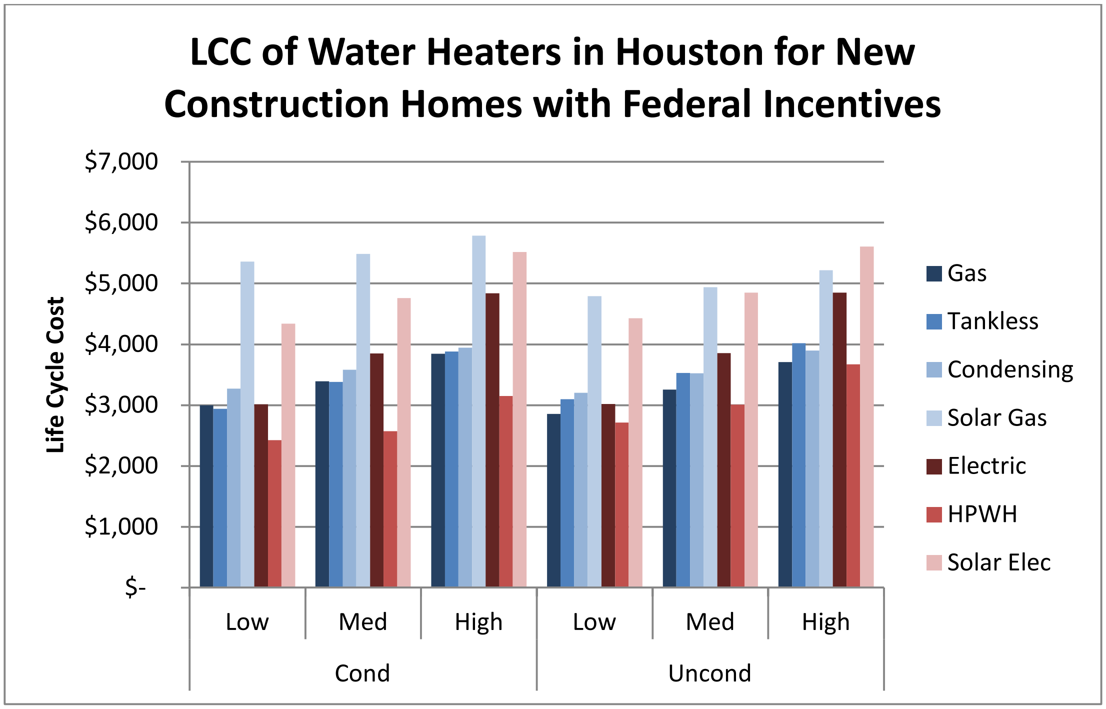
HPWH
Solar Elec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$1,000
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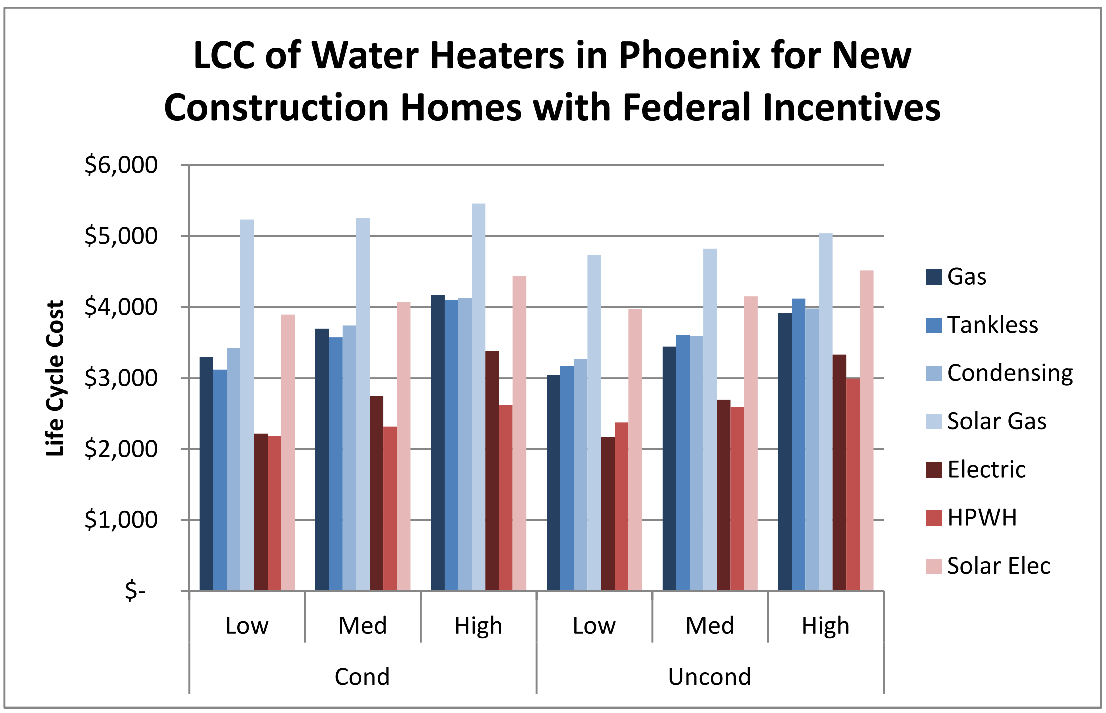
HPWH
Solar Elec


LCC of Water Heaters in Phoenix for New Construction Homes with Federal Incentives
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$2,000
$1,000
$-
Low
Med
High
Low
Med
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Cond
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Gas
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Condensing
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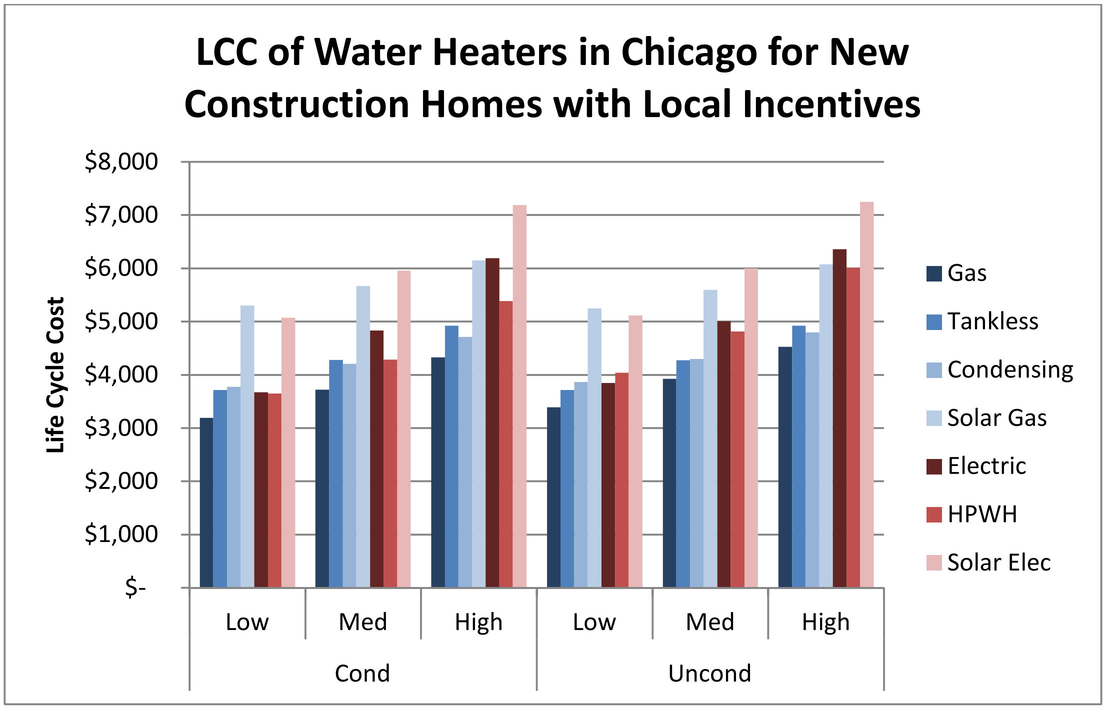
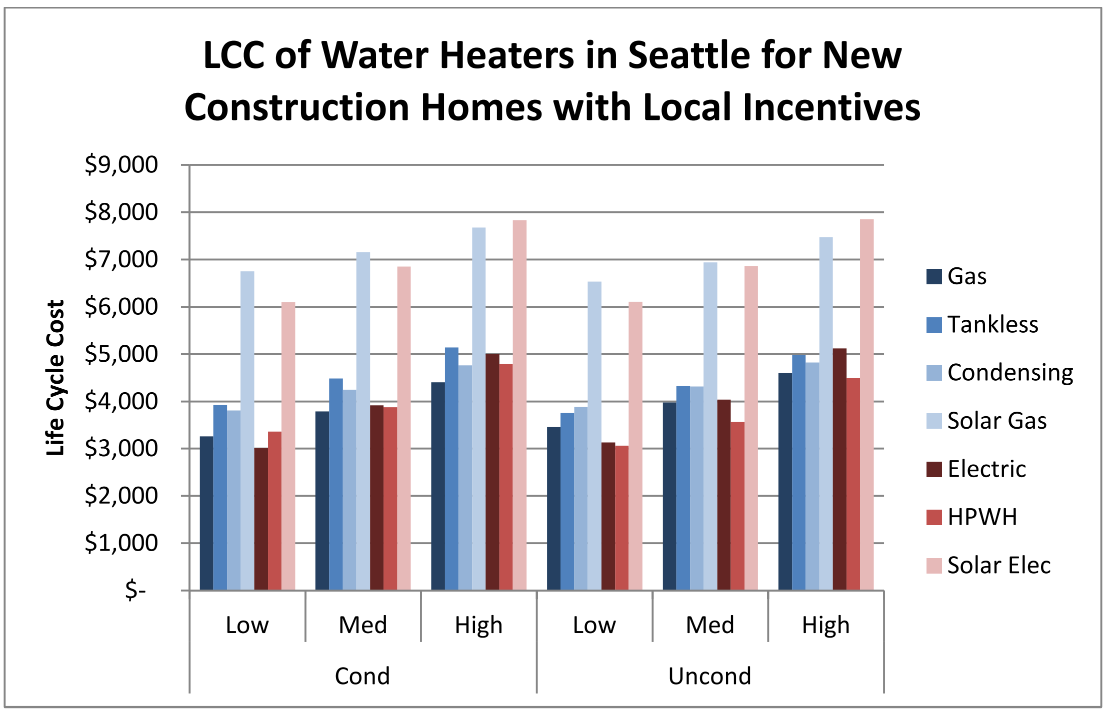
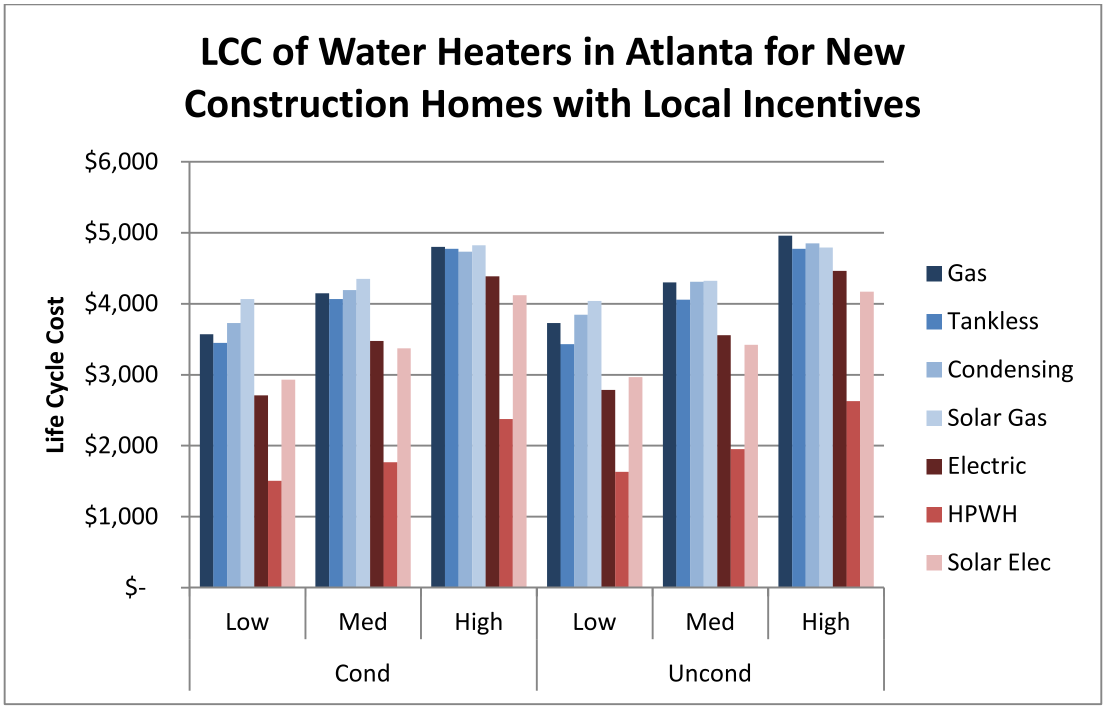
Electric
HPWH
Solar Elec

New Construction, Local Incentives



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Life Cycle Cost
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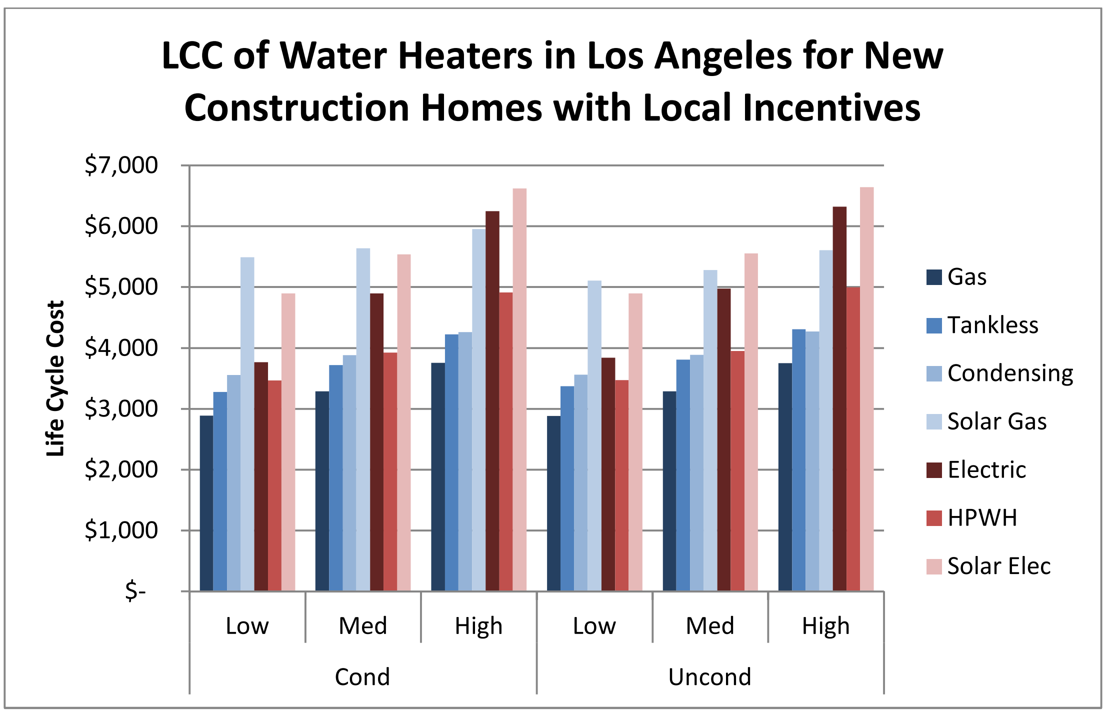
HPWH
Solar Elec


LCC of Water Heaters in Los Angeles for New Construction Homes with Local Incentives
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Cond
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Gas
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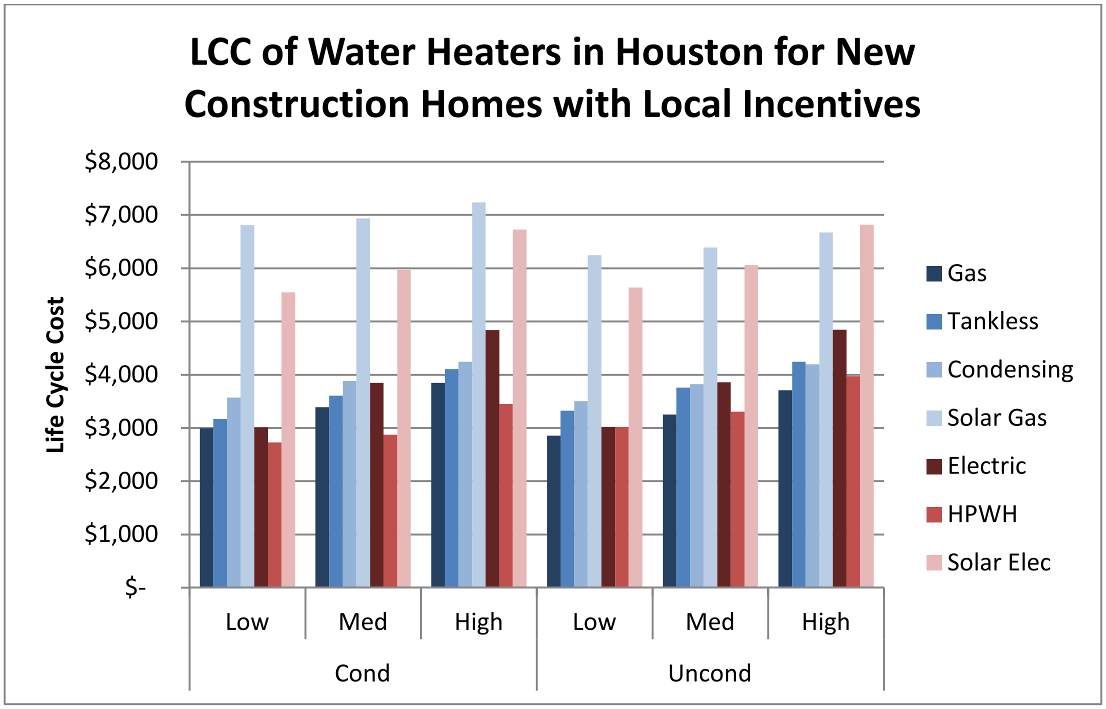
Condensing
Solar Gas
Electric
HPWH
Solar Elec



LCC of Water Heaters in Houston for New Construction Homes with Local Incentives
Life Cycle Cost
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$6,000
$5,000
$4,000
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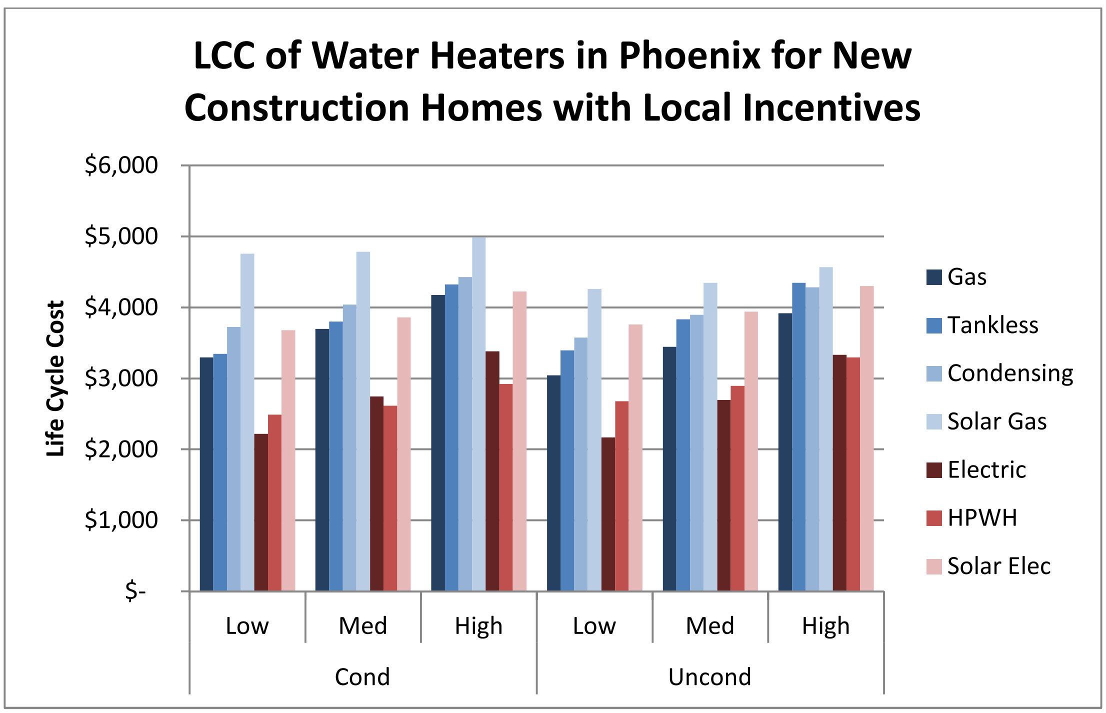

LCC of Water Heaters in Phoenix for New Construction Homes with Local Incentives
Life Cycle Cost
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Solar Gas
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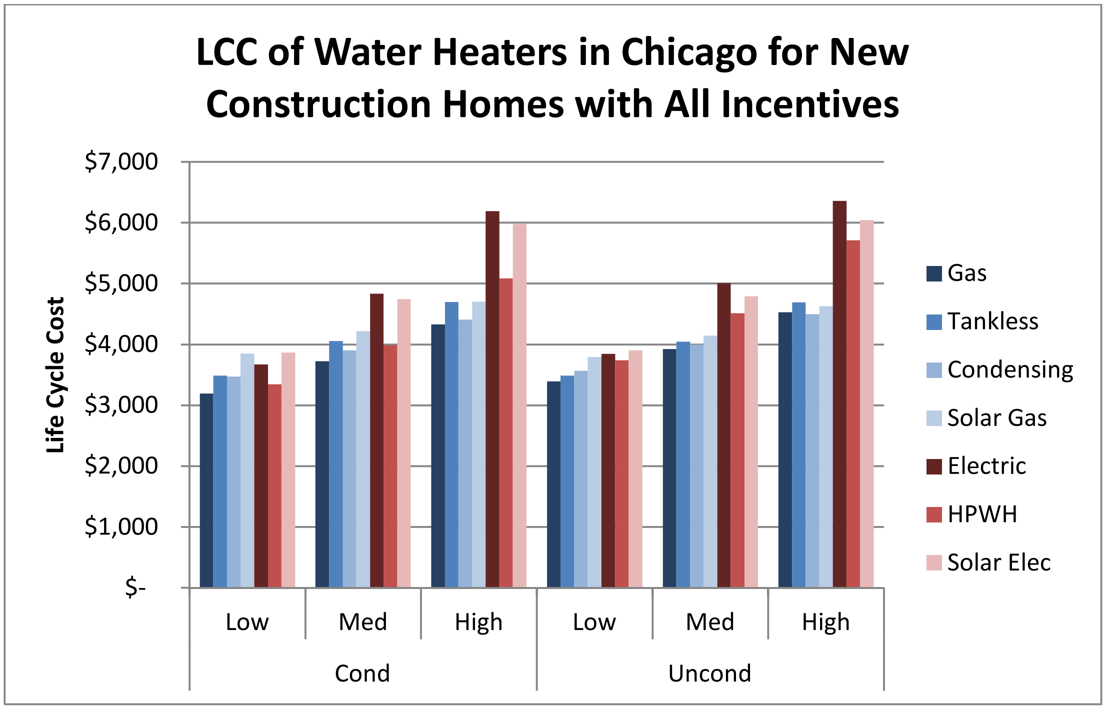
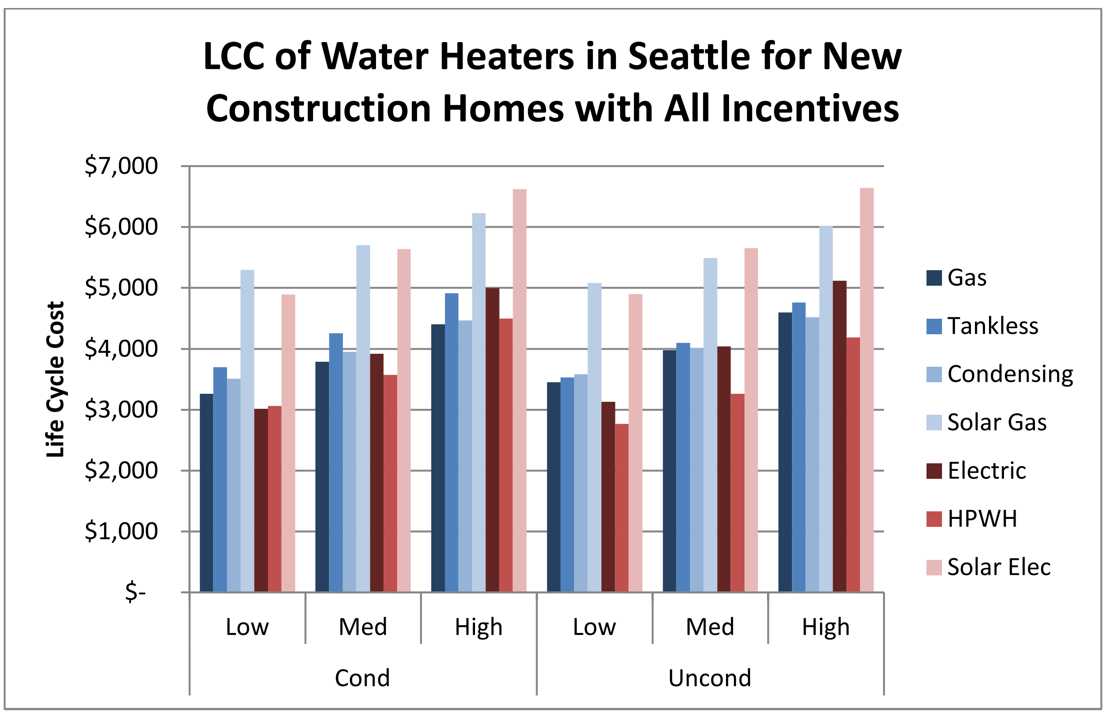
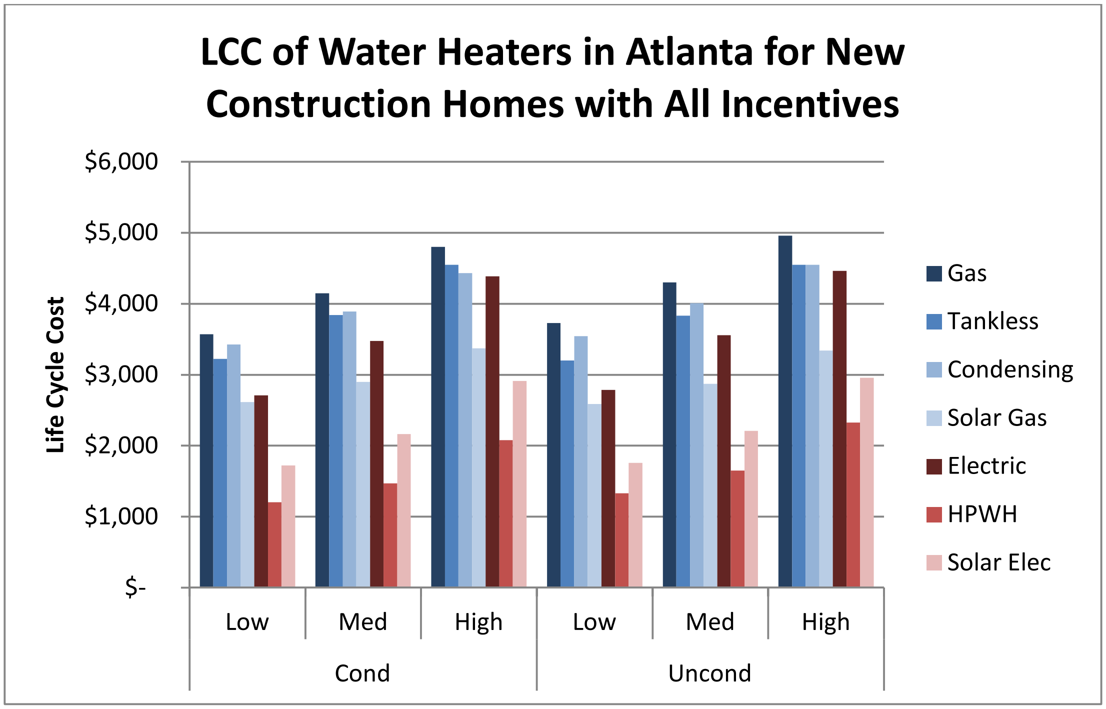
Solar Elec

New Construction, All Incentives






LCC of Water Heaters in Atlanta for New Construction Homes with All Incentives
Life Cycle Cost
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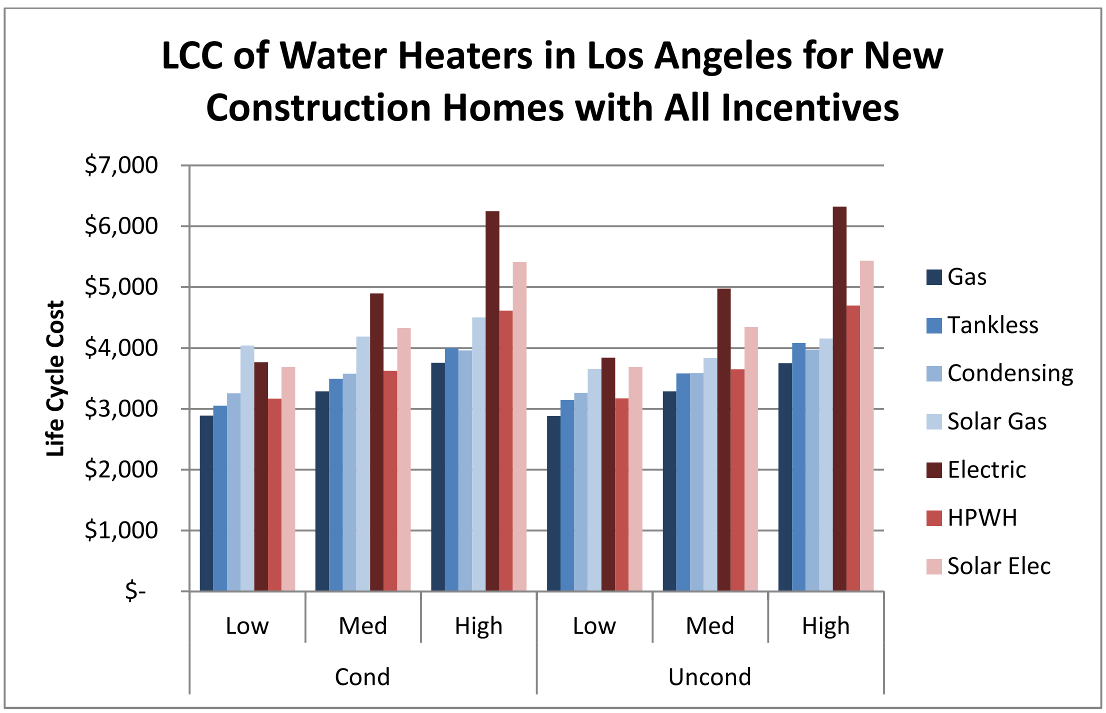
LCC of Water Heaters in Los Angeles for New Construction Homes with All Incentives
Life Cycle Cost
$7,000
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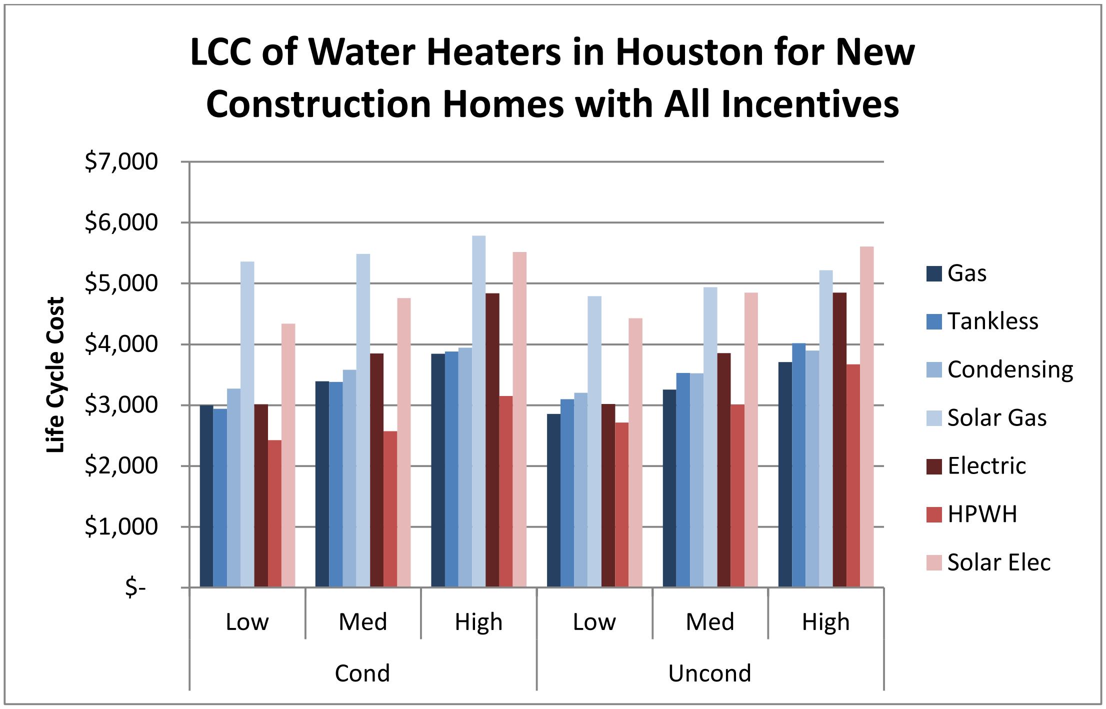
LCC of Water Heaters in Houston for New Construction Homes with All Incentives
Life Cycle Cost
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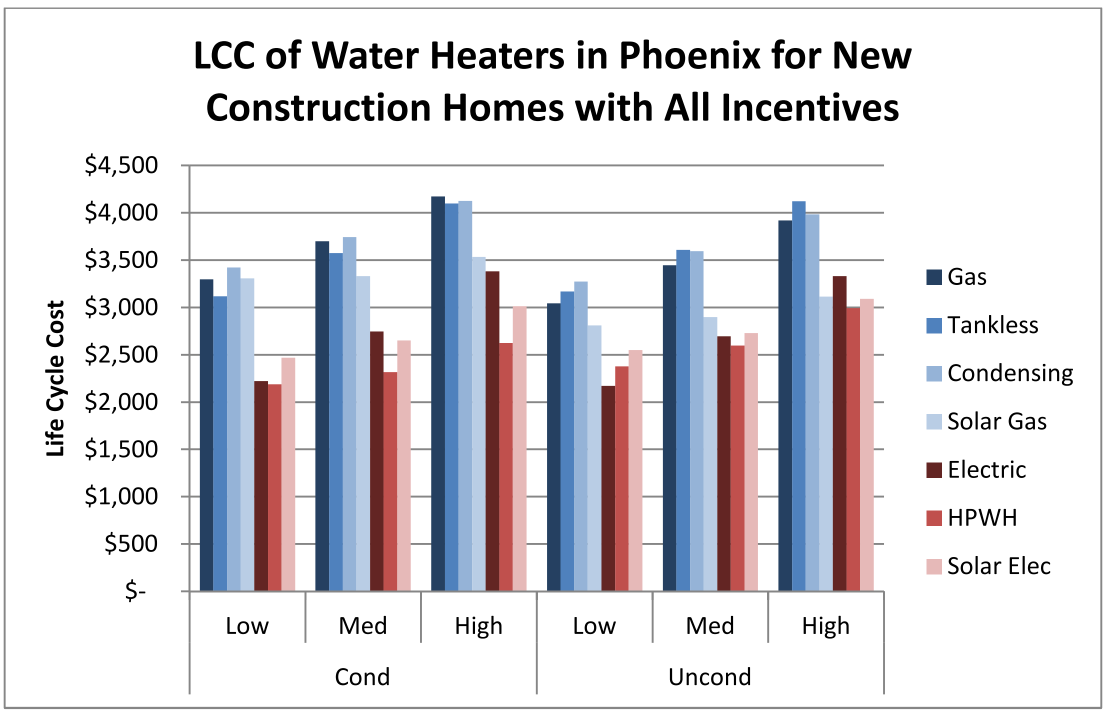
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Life Cycle Cost
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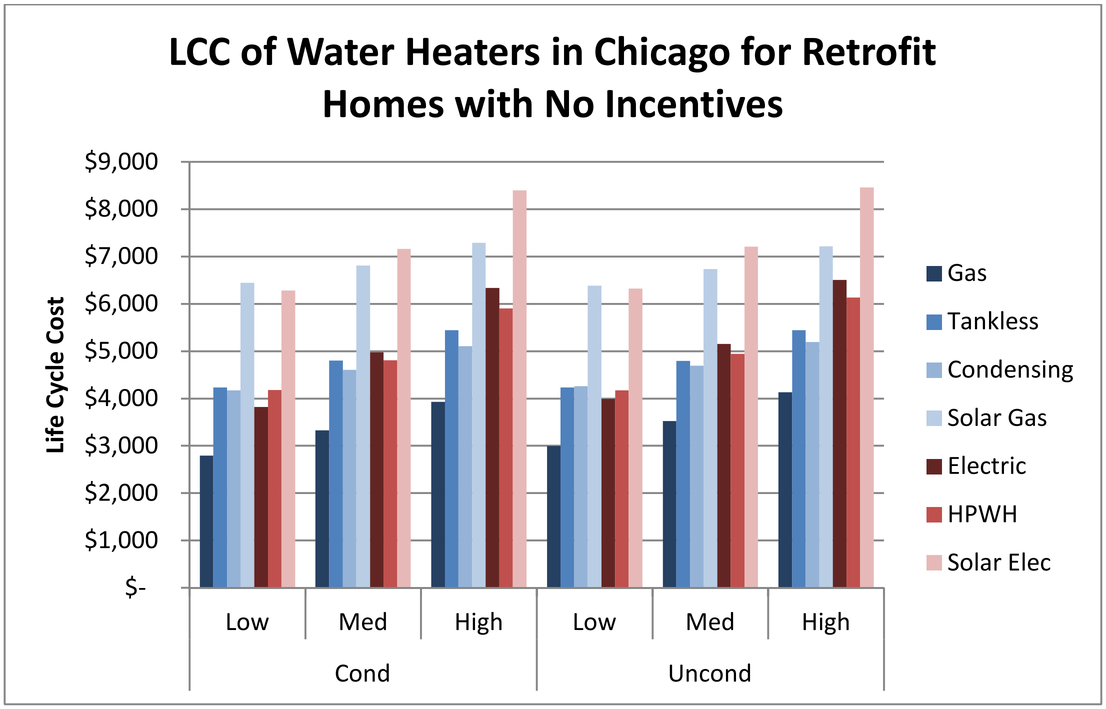
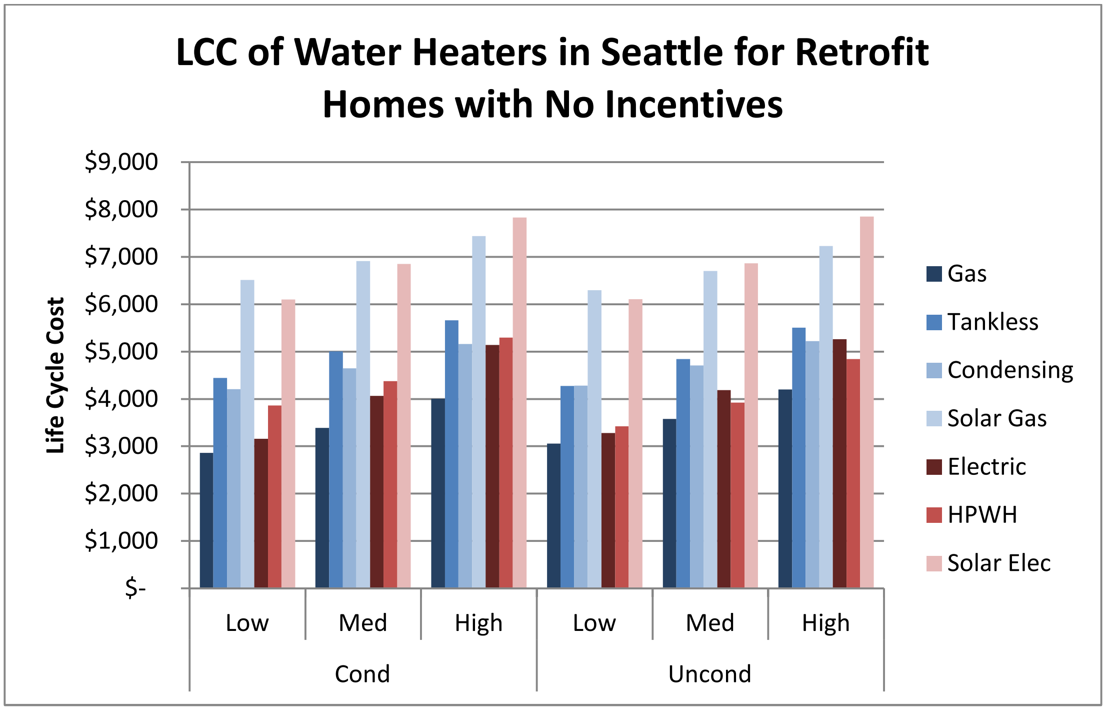
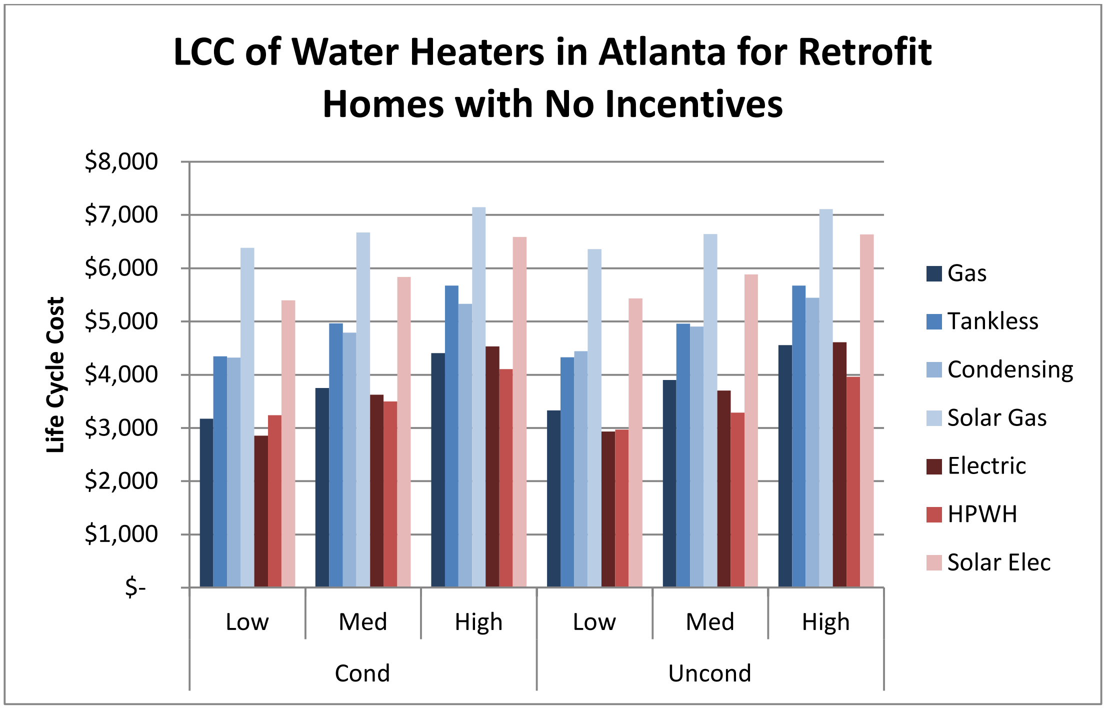
Retrofit, No Incentives






LCC of Water Heaters in Atlanta for Retrofit Homes with No Incentives
Life Cycle Cost
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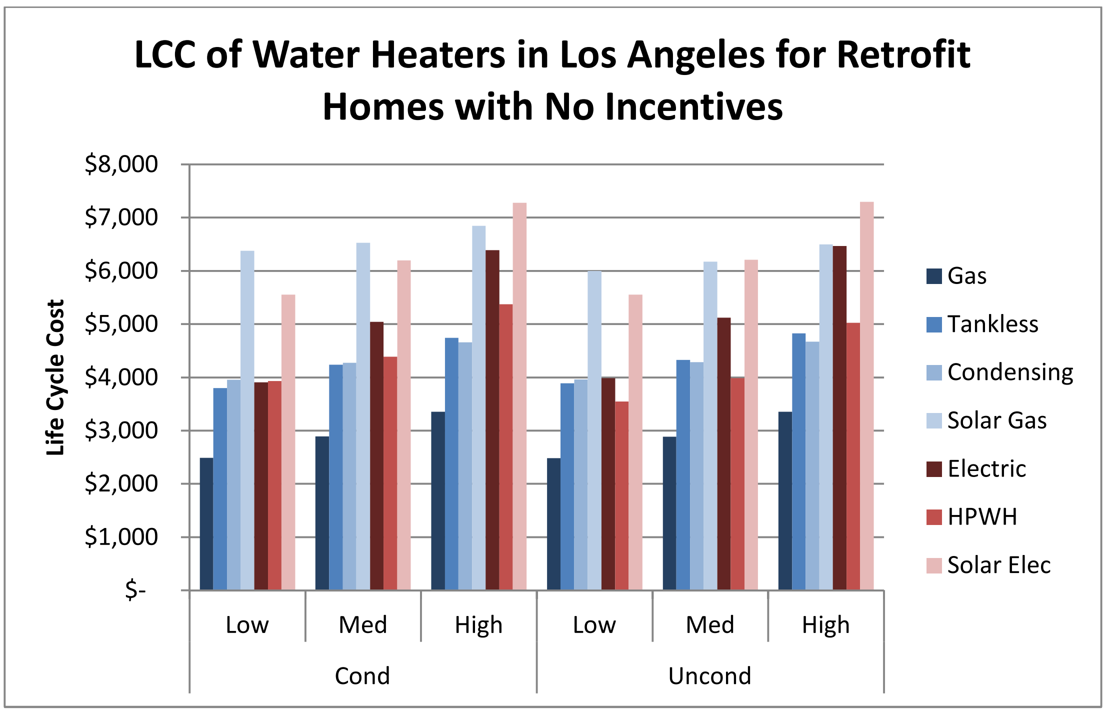
LCC of Water Heaters in Los Angeles for Retrofit Homes with No Incentives
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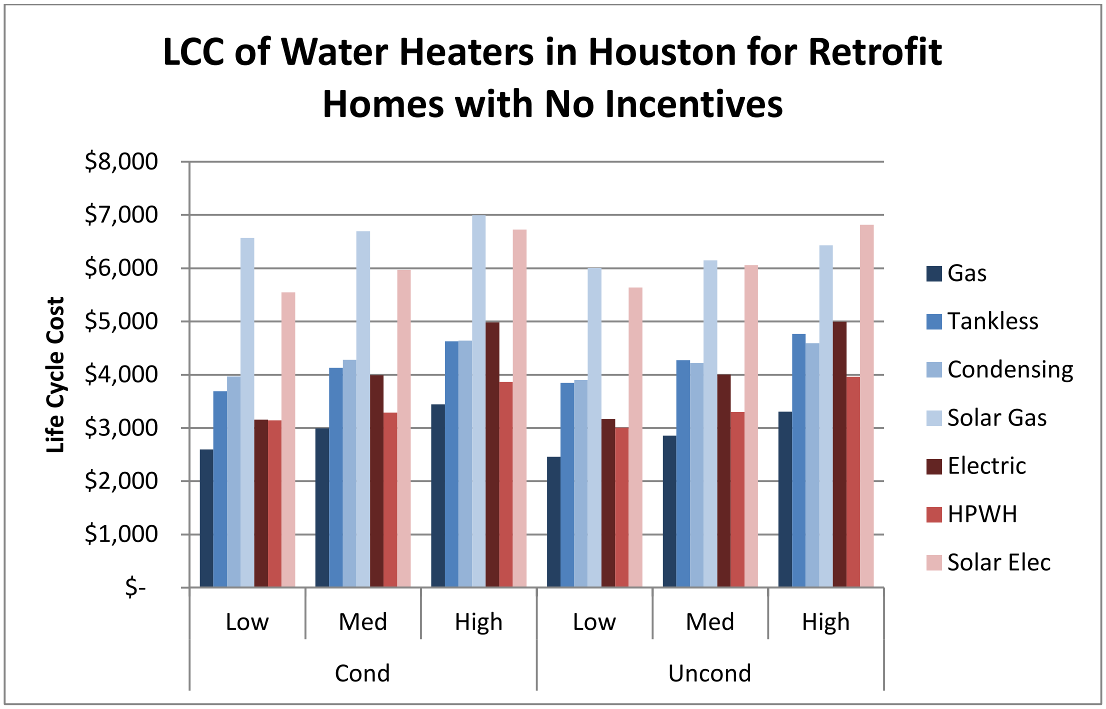
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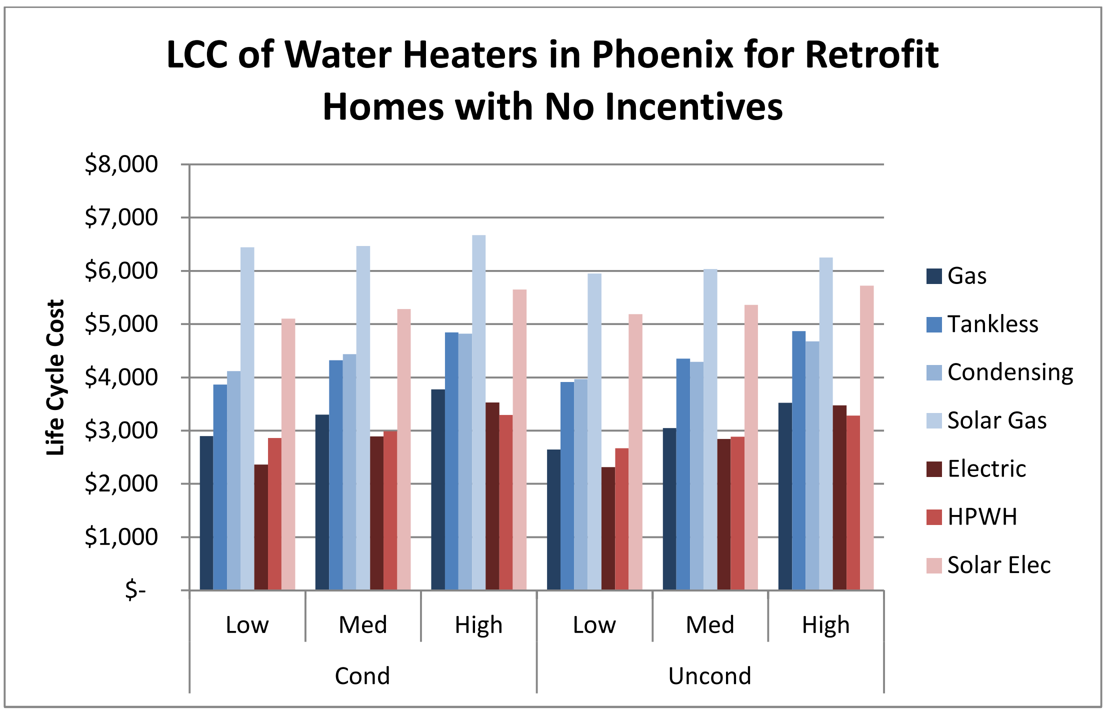
LCC of Water Heaters in Phoenix for Retrofit Homes with No Incentives
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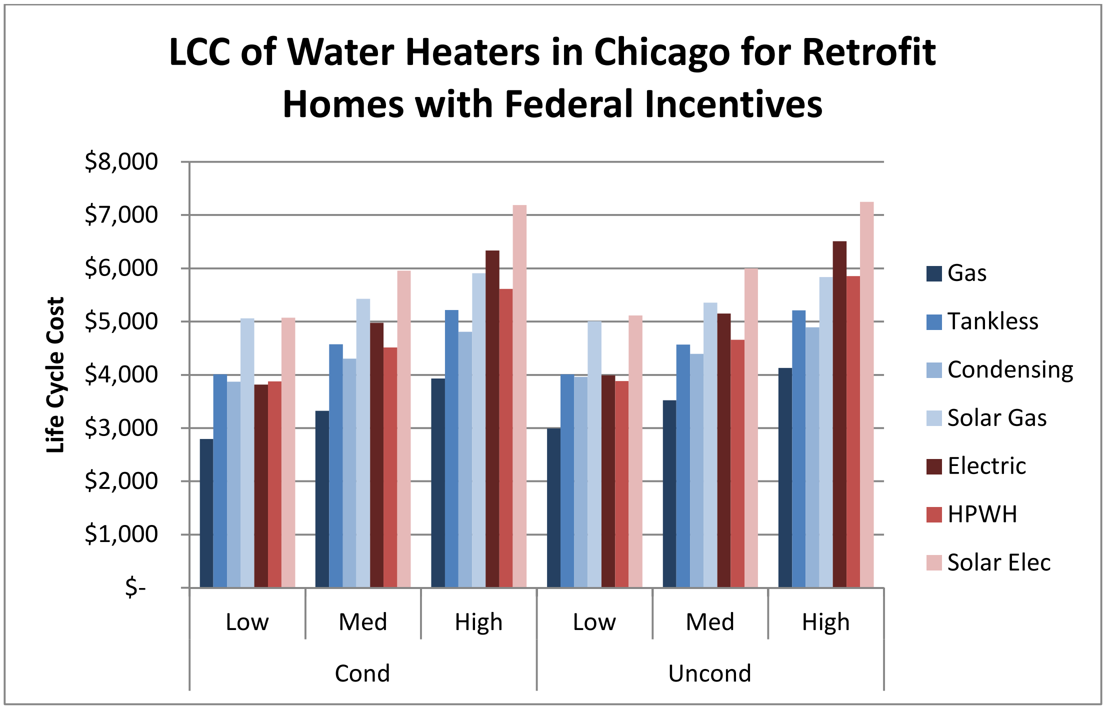
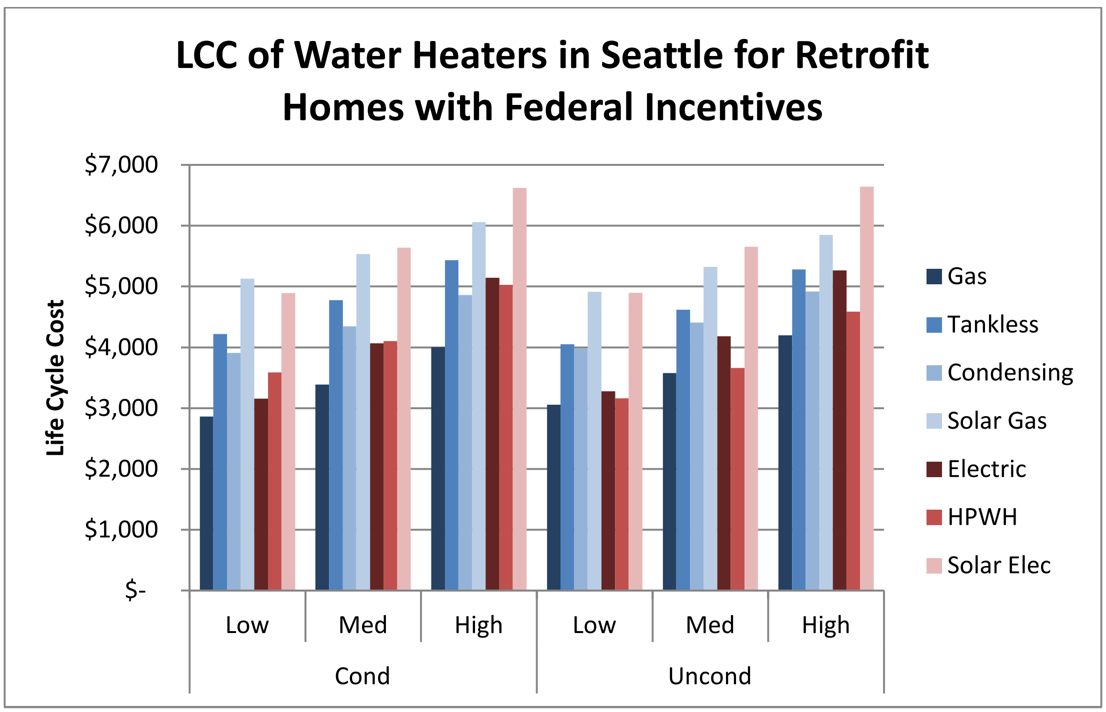
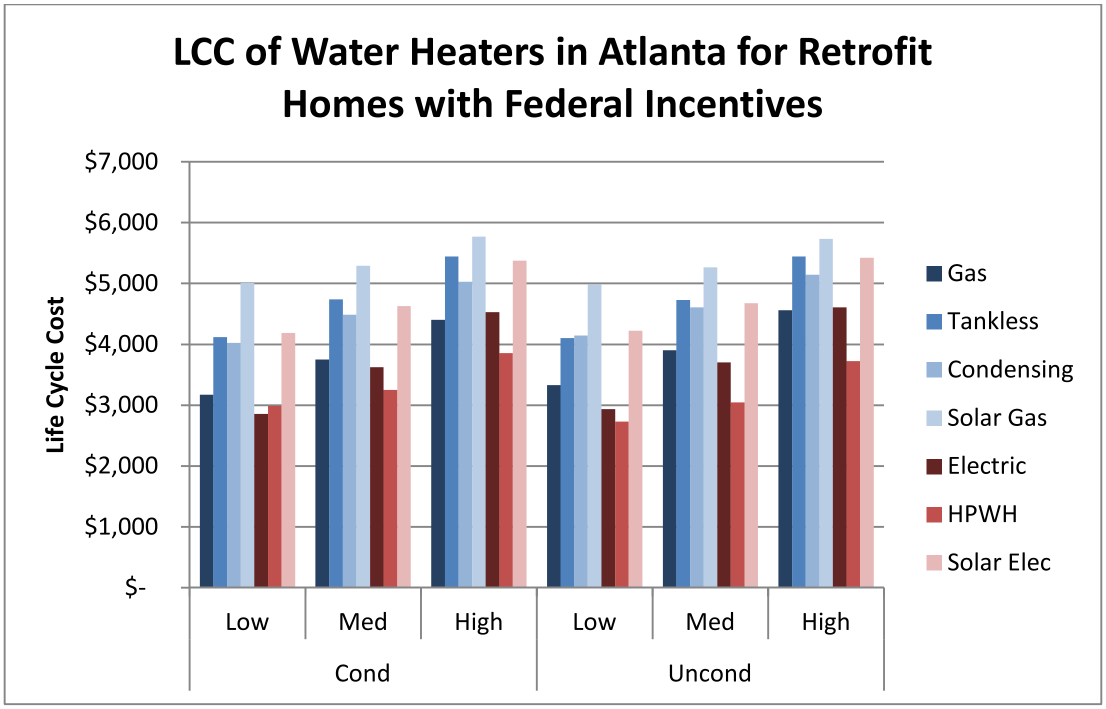
Retrofit, Federal Incentives






LCC of Water Heaters in Atlanta for Retrofit Homes with Federal Incentives
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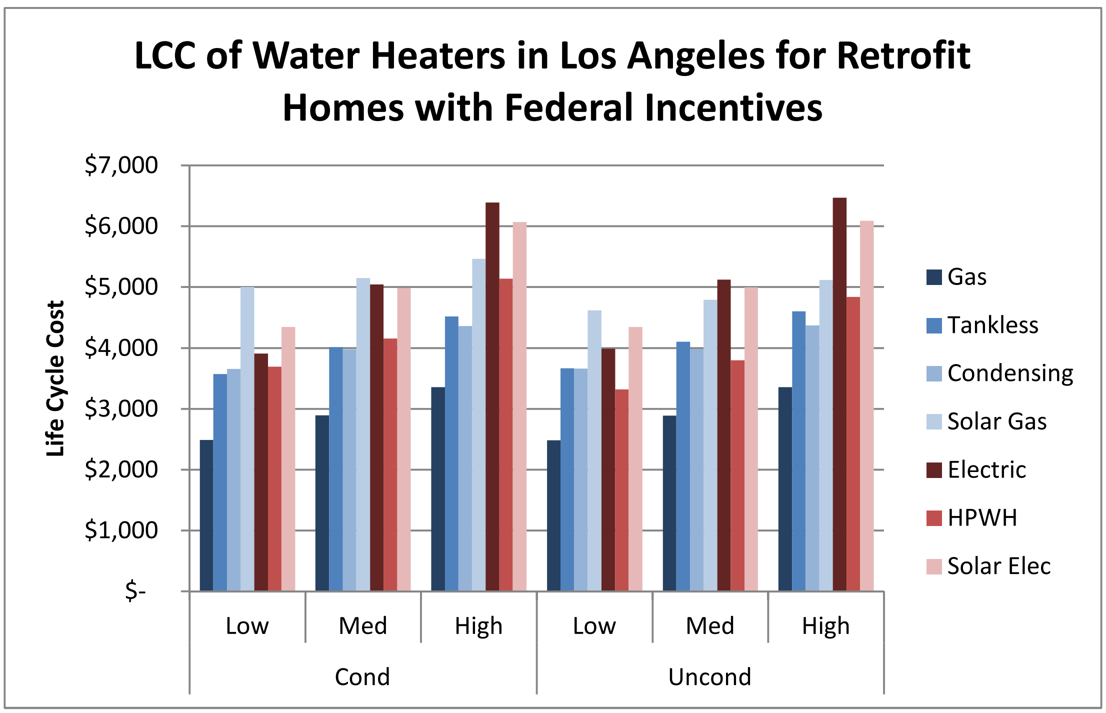
LCC of Water Heaters in Los Angeles for Retrofit Homes with Federal Incentives
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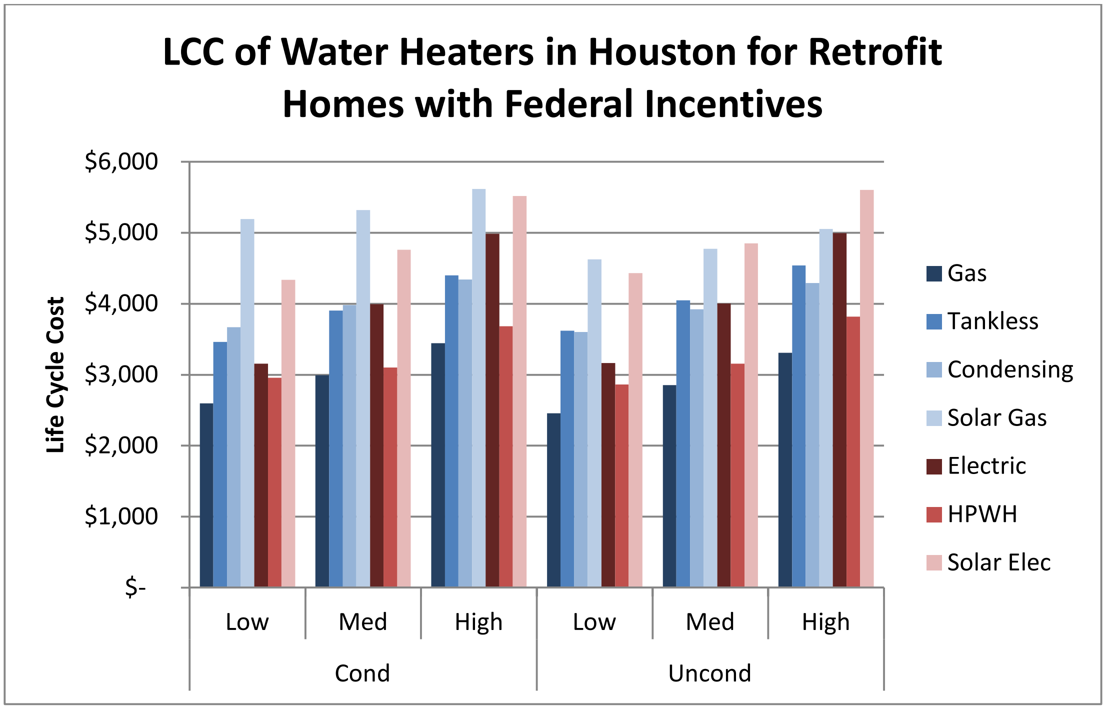
LCC of Water Heaters in Houston for Retrofit Homes with Federal Incentives
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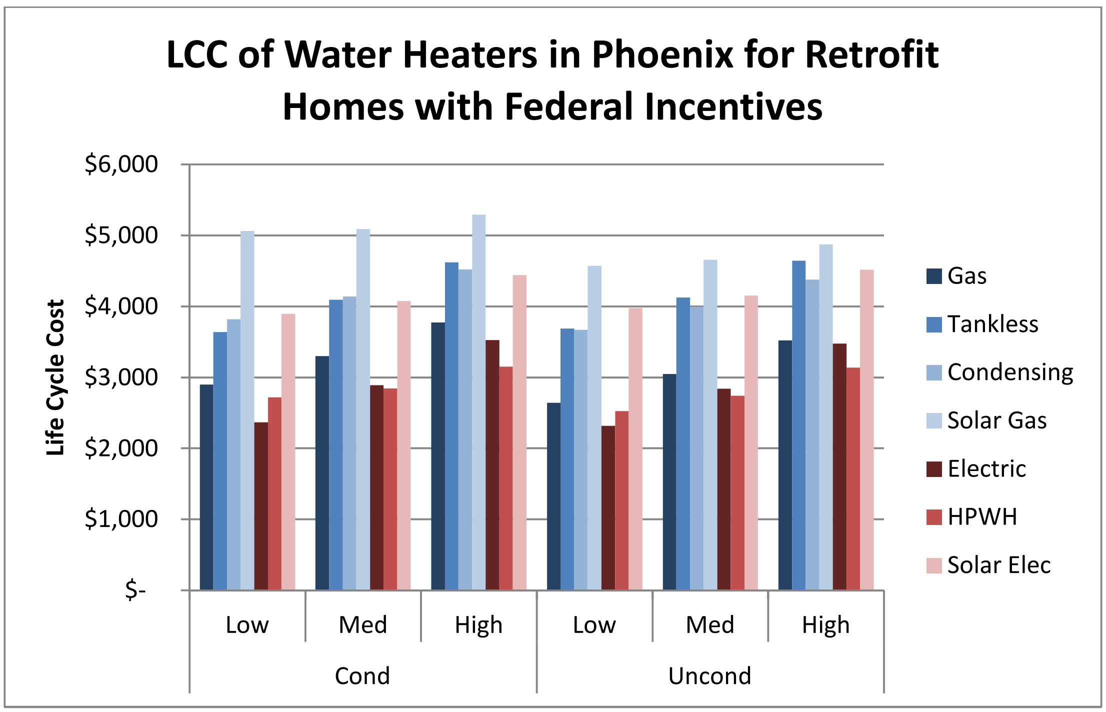
LCC of Water Heaters in Phoenix for Retrofit Homes with Federal Incentives
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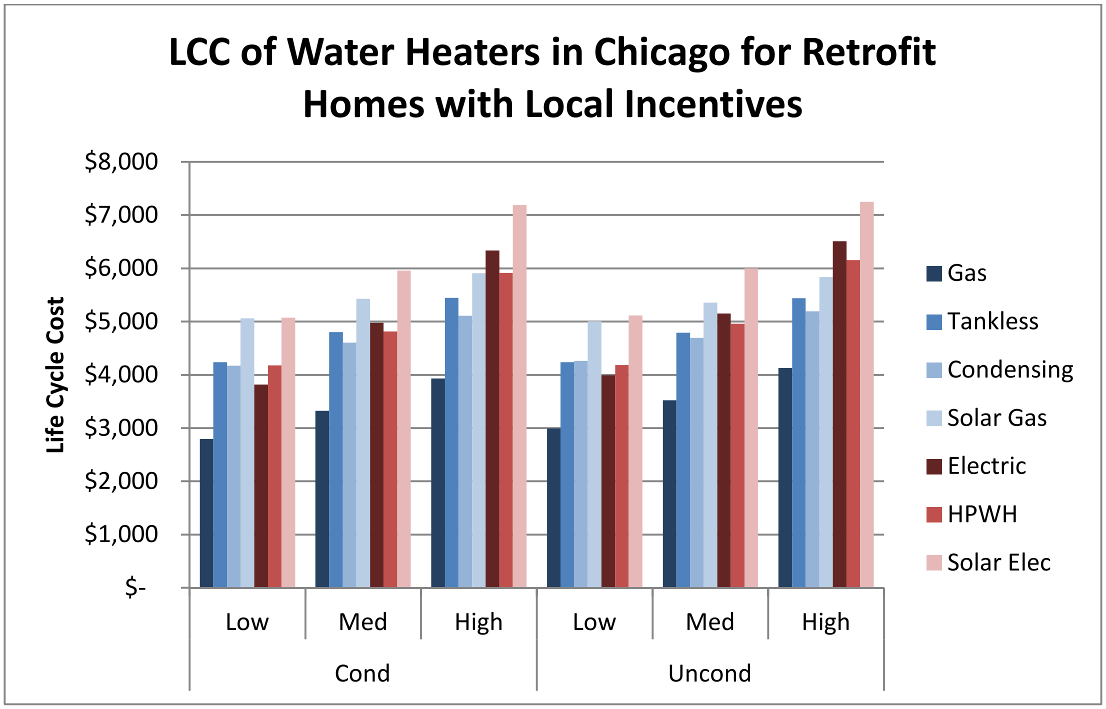
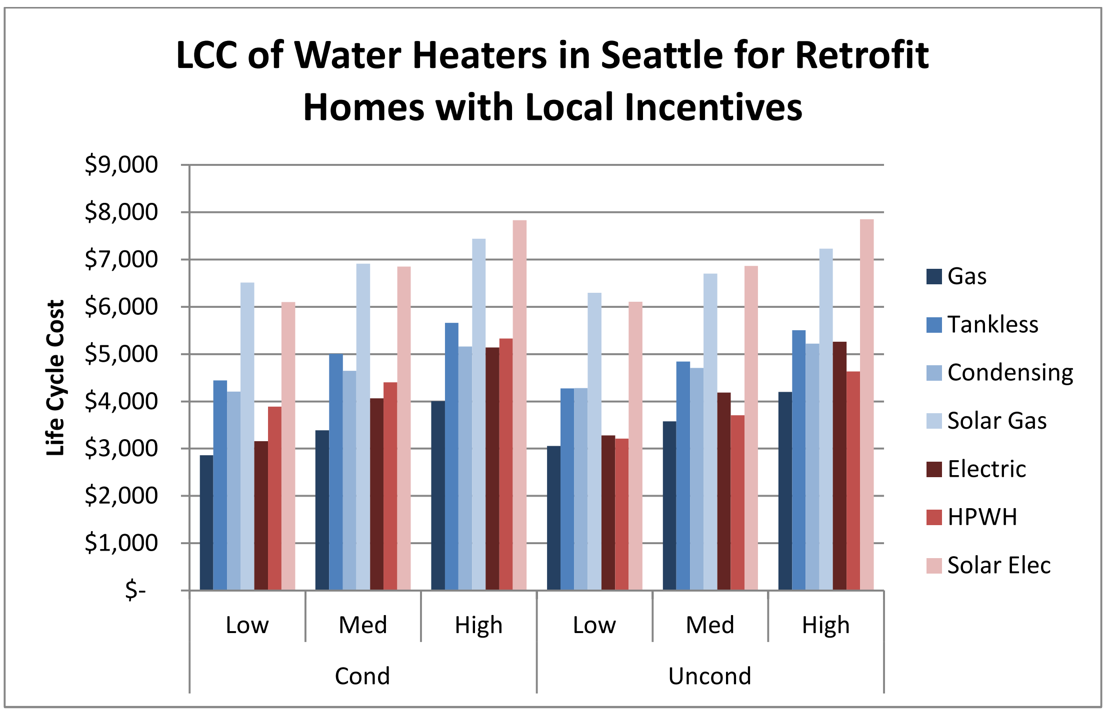
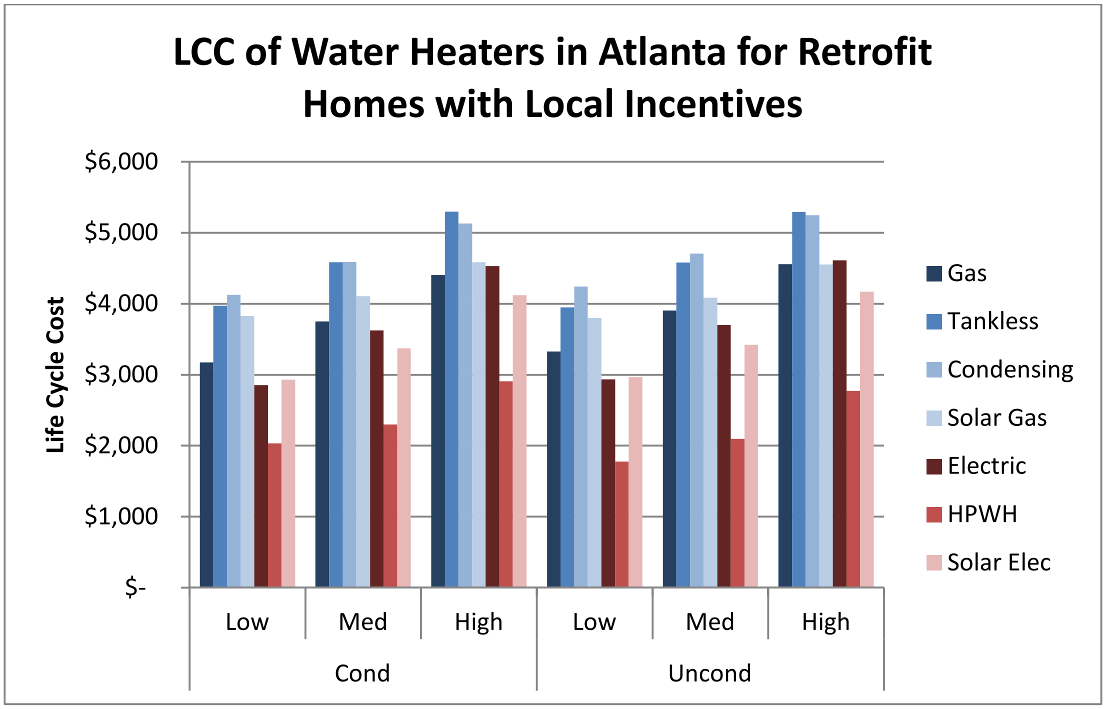
Retrofit, Local Incentives






LCC of Water Heaters in Atlanta for Retrofit Homes with Local Incentives
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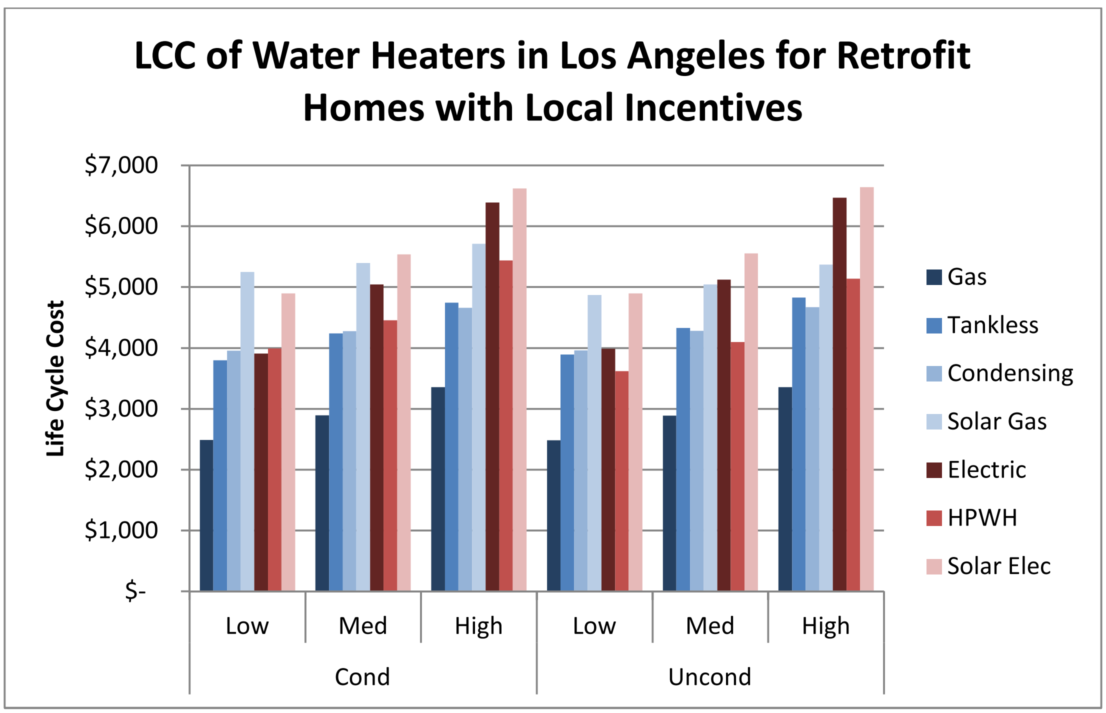
LCC of Water Heaters in Los Angeles for Retrofit Homes with Local Incentives
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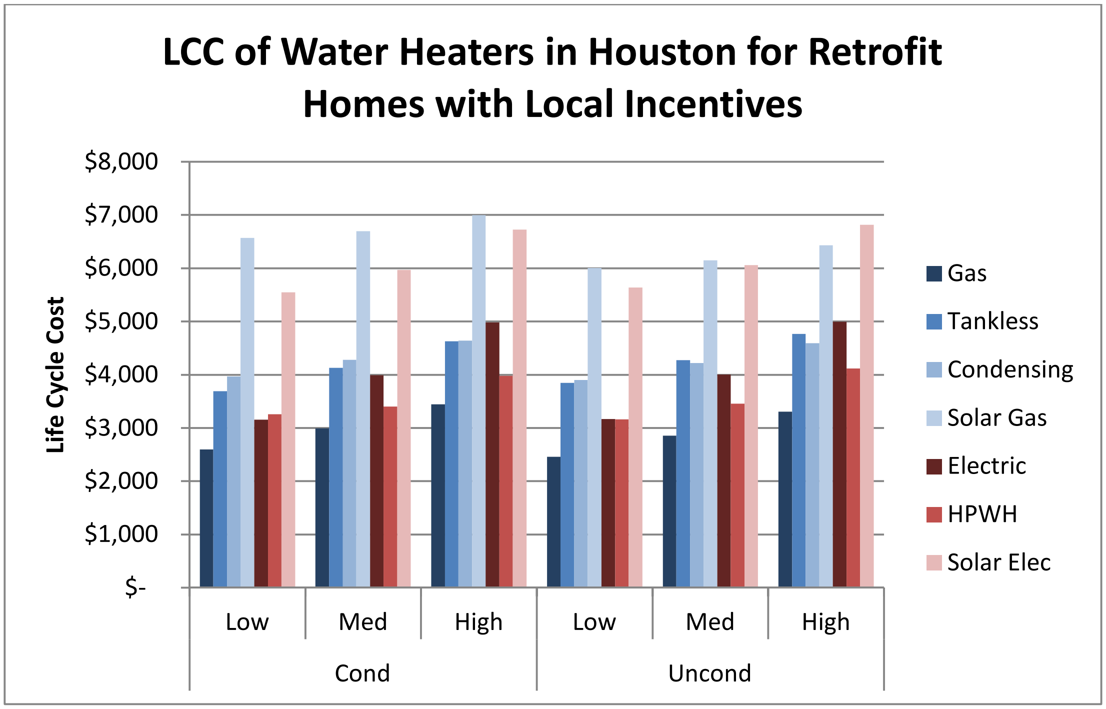
LCC of Water Heaters in Houston for Retrofit Homes with Local Incentives
Life Cycle Cost
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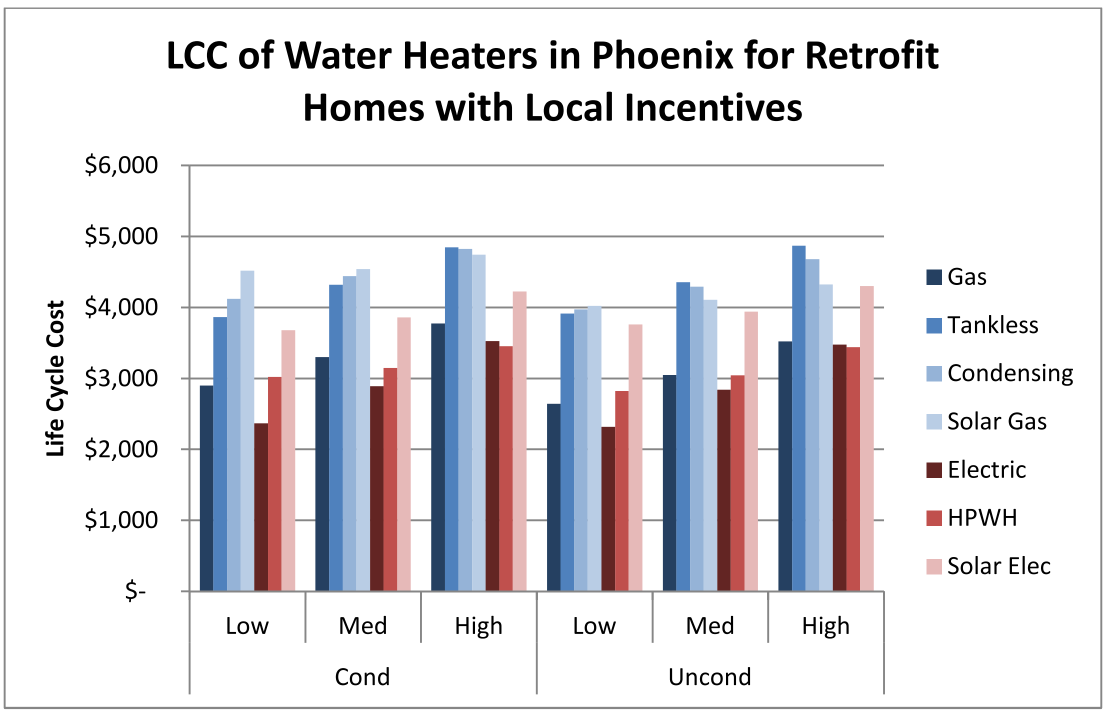
LCC of Water Heaters in Phoenix for Retrofit Homes with Local Incentives
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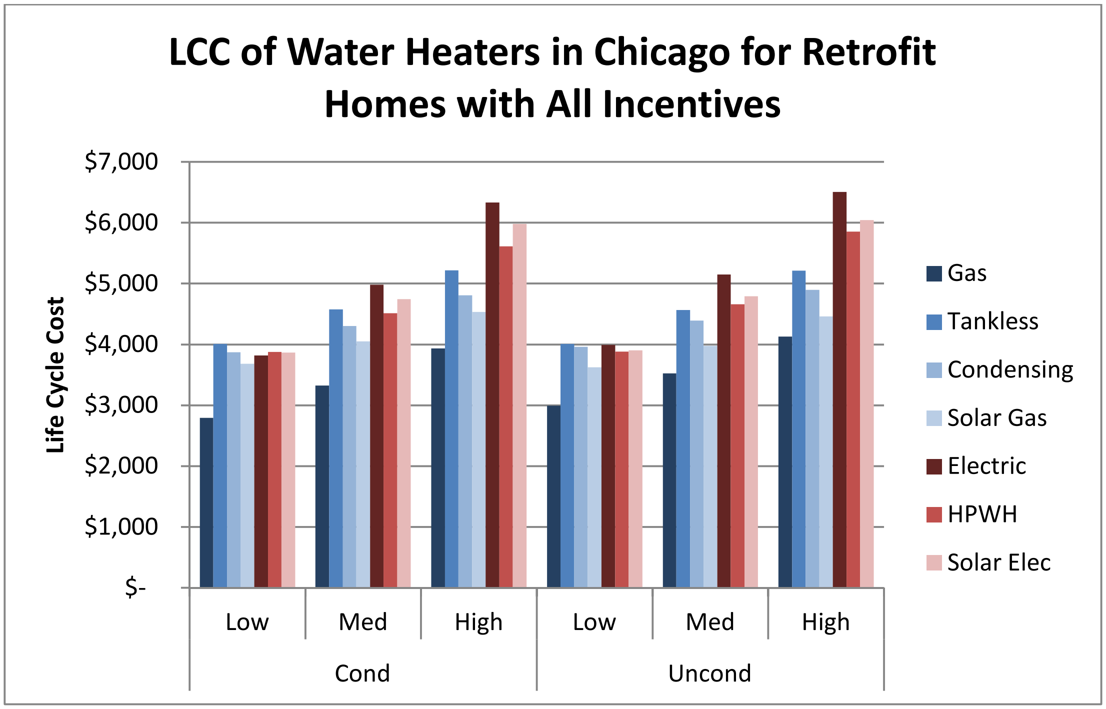
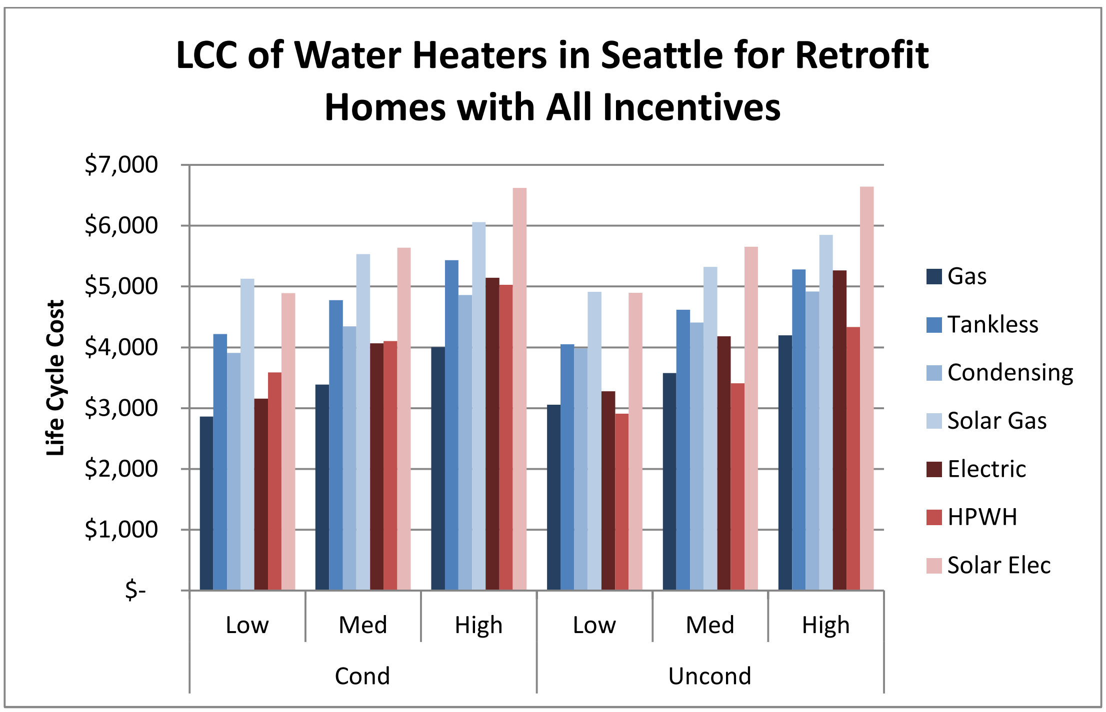
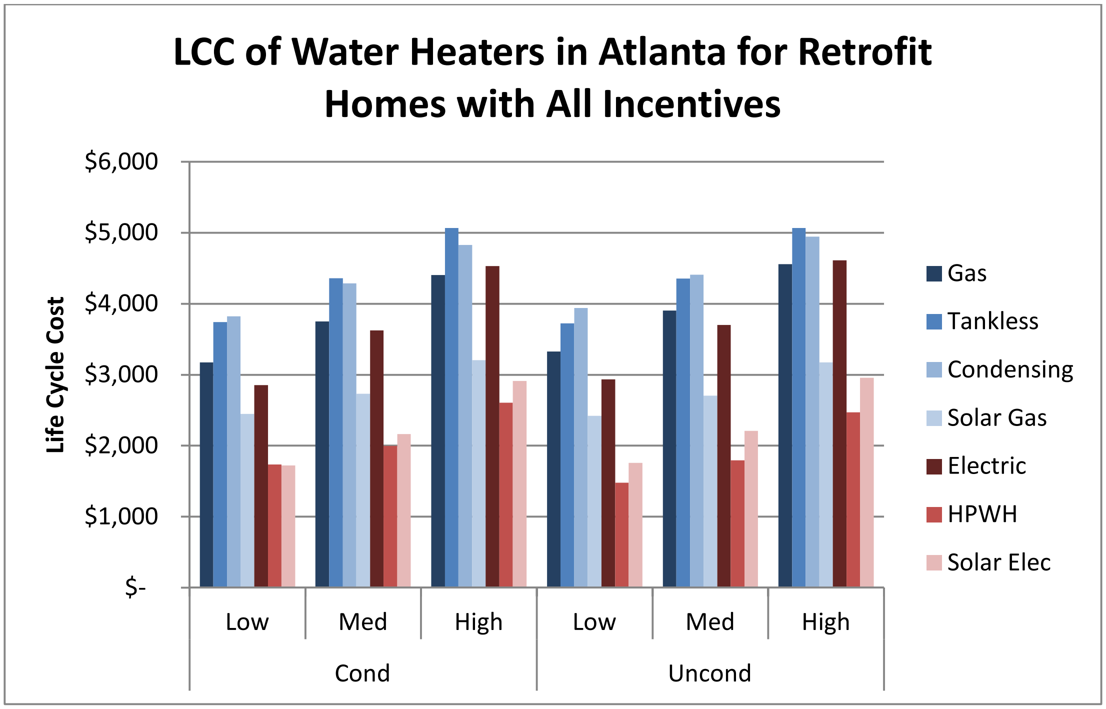
Retrofit, All Incentives






LCC of Water Heaters in Atlanta for Retrofit Homes with All Incentives
Life Cycle Cost
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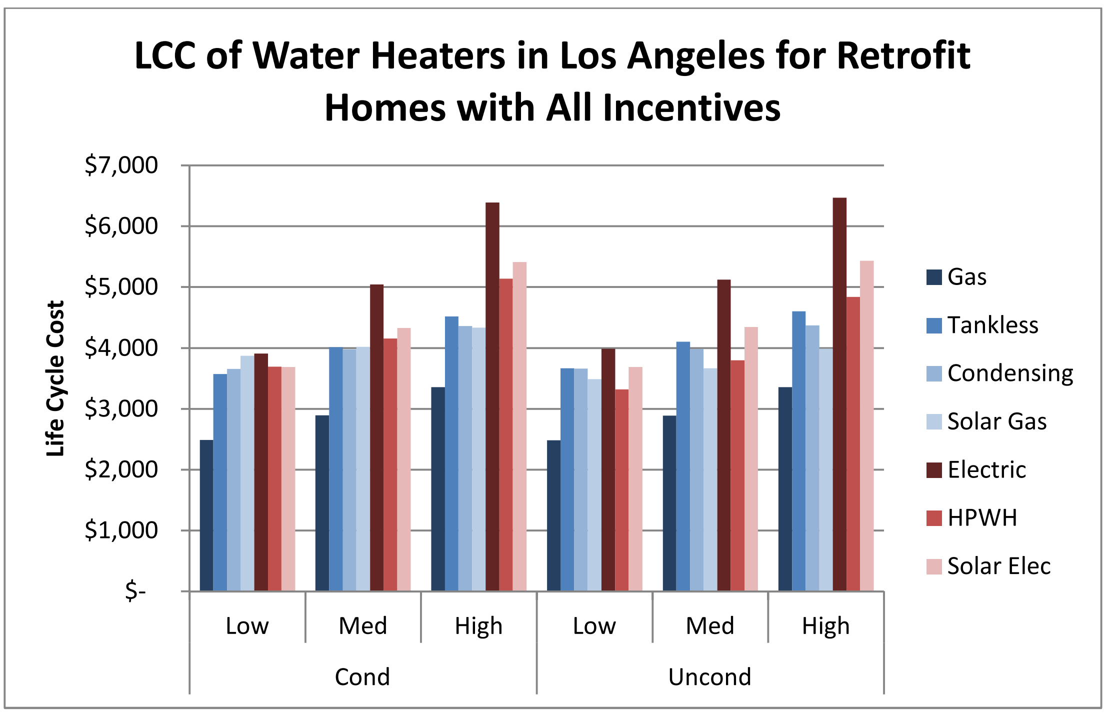
LCC of Water Heaters in Los Angeles for Retrofit Homes with All Incentives
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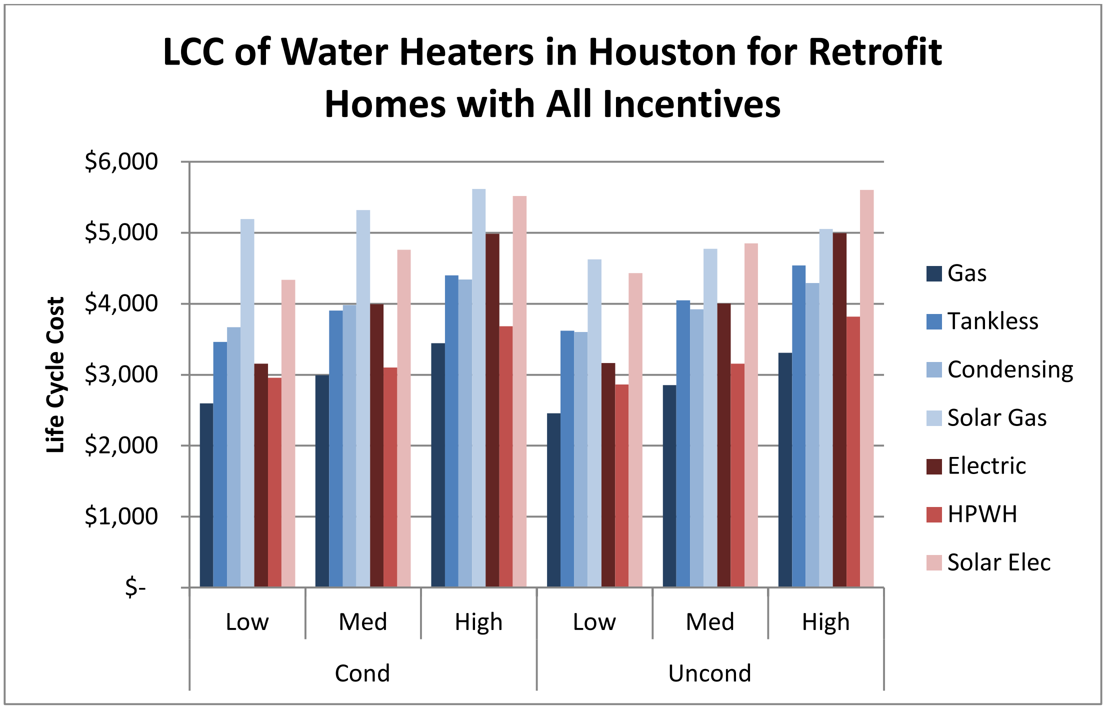
LCC of Water Heaters in Houston for Retrofit Homes with All Incentives
Life Cycle Cost
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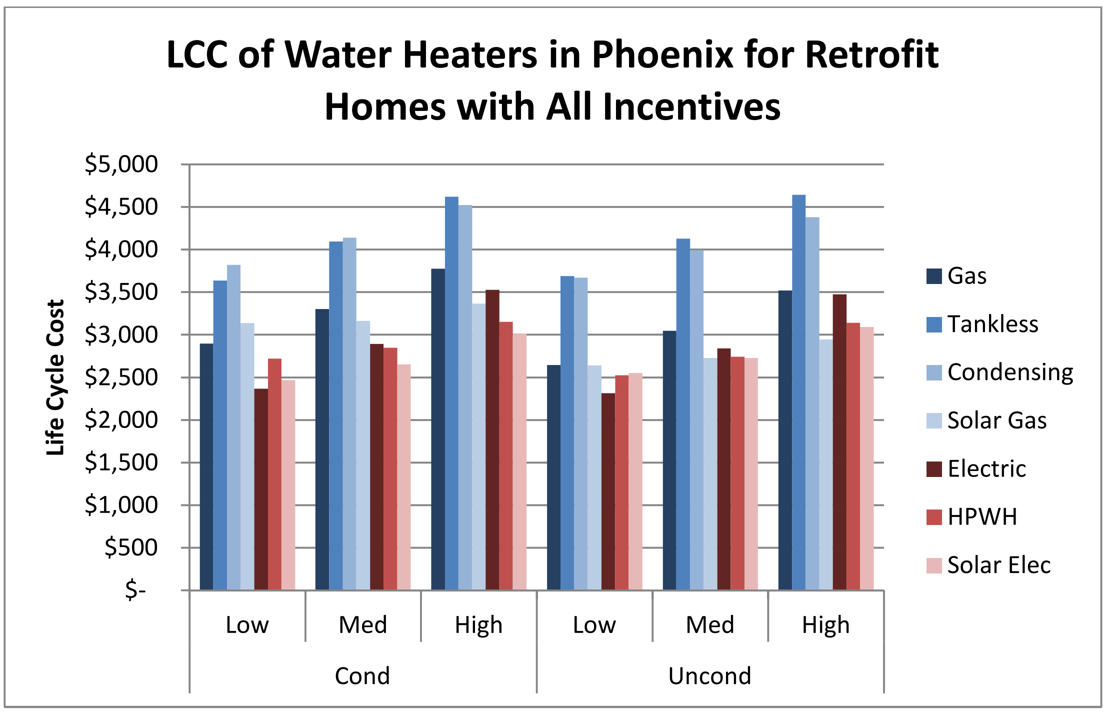
LCC of Water Heaters in Phoenix for Retrofit Homes with All Incentives
Life Cycle Cost
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