2016

Regional Climate Zone Modeling of a Commercial Absorption Heat Pump Hot Water Heater - Part 1: Southern and South Central Climate Zones

Patrick Geoghegan
Oak Ridge National Laboratory, geogheganpj@ornl.gov

Bo Shen
Oak Ridge National Laboratory, shenb@ornl.gov

Christopher Keinath
Stone Mountain Technologies, Inc., ckeinath@stonemtntechnologies.com

Michael Garrabrant
Stone Mountain Technologies, Inc., mgarrabrant@stonemtntechnologies.com

Regional Climate Zone Modeling of a Commercial Absorption Heat Pump Hot Water Heater – Part 1: Southern and South Central Climate Zones

Patrick Geoghegan[1*], Bo Shen[1], Chris Keinath[2], Michael Garrabrant[2]

[1]Oak Ridge National Laboratory
1 Bethel Valley Road
Oak Ridge, TN 37831
geogheganpj@ornl.gov

[2]Stone Mountain Technologies, Inc.
609 Wesinpar Road
Johnson City, TN 37604

*Corresponding Author

ABSTRACT

Commercial hot water heating accounts for approximately 0.78 Quads of primary energy use with 0.44 Quads of this amount from natural gas fired heaters. An ammonia-water based commercial absorption system, if fully deployed, could achieve a high level of savings, much higher than would be possible by conversion to the high efficiency non-heat-pump gas fired alternatives. In comparison with air source electric heat pumps, the absorption system is able to maintain higher coefficients of performance in colder climates. The ammonia-water system also has the advantage of zero Ozone Depletion Potential and low Global Warming Potential. A thermodynamic model of a single effect ammonia-water absorption system for commercial space and water heating was developed, and its performance was investigated for a range of ambient and return water temperatures. This allowed for the development of a performance map which was then used in a building energy modeling software. Modeling of two commercial water heating systems was performed; one using an absorption heat pump and another using a condensing gas storage system. The energy and financial savings were investigated for a range of locations and climate zones in the southern and south central United States. A follow up paper will analyze northern and north/central regions. Results showed that the system using an absorption heat pump offers significant savings.

This manuscript has been authored by UT-Battelle, LLC under Contract No. DE-AC05-00OR22725 with the U.S. Department of Energy. The United States Government retains and the publisher, by accepting the article for publication, acknowledges that the United States Government retains a non-exclusive, paid-up, irrevocable, world-wide license to publish or reproduce the published form of this manuscript, or allow others to do so, for United States Government purposes. The Department of Energy will provide public access to these results of federally sponsored research in accordance with the DOE Public Access Plan(http://energy.gov/downloads/doe-public-access-plan)

1. INTRODUCTION

The Gas Absorption Heat Pump (GAHP) has the ability to provide hot water at Coefficients of Performance (COP) greater than that of conventional boilers and furnaces. They achieve this by combing the heat of the high temperature source of the combusted gas and the low temperature source of the surrounding ambient. This allows the heat pump to achieve COP values greater than 1 while furnaces and boilers are limited to COP values less than 1. Operation of a GAHP is similar to that of a vapor compression system except that a thermal compressor is used in place of the mechanical compressor. The thermal compressor consists of a series of heat and mass exchangers (absorber, desorber, solution heat exchanger) and a low flow, higher pressure difference pump. As a result of their higher COP values, the GAHP uses less fuel and has the potential to significantly reduce annual operating cost.

However, Gas Absorption Heat Pumps have made little impact in the commercial hot water market. This is because GAHP systems that are commercially available have a higher cost premium in comparison to standard and high efficiency hot water tanks making the payback period for this technology unfavorable. Customers, contractors and service personnel are also more familiar with conventional technologies and this acts a barrier for adoption. With an increased awareness of energy use, customers are growing more aware of their energy footprint and GAHPs are now being investigated as a gas source replacement for the conventional systems. In order to elucidate on these points, this paper sets out to evaluate the performance of a GAHP across the southern and south central climate zones of the United States, availing of the ambient air temperature data, mains water temperature data and the hot water heater tank models available in EnergyPlus (www.energyplus.net). A follow-up paper will analyze GAHP performance in northern and north/central regions to accommodate ambient temperatures below 5.5 °C in the EnergyPlus Heat Pump model and incorporate a realistic defrost control strategy for GAHPs. Together, they will provide a clearer picture of where GAHPs could be introduced in to the US market to provide greatest energy and financial savings.

2. MODELING

This work compares a standard hot water heating configuration for a full service restaurant with a GAHP alternative layout. The former consists of two 100 gallon (0.3785 m^3) tanks operating in series (Figure 1). The first is a high efficiency 58.3 kW (199 kBTU/hr) unit followed in series by a standard efficiency 58.3 kW (199 kBTU/hr) unit. The set point of each tank is set at 60°C (140°F). In practice, the second tank is a topping off tank to cater for peak demand. During an average water draw day, it should see little use. A high efficiency unit could be used in place of the standard efficiency unit but the benefit of the condensing heat exchanger would be lost because of the high inlet water temperature into this second tank. The restaurant has a recirculation loop that returns unused water back to second tank, in which no heat losses are assumed.

The GAHP configuration is shown in Figure 2. The heat pump itself resides outside the building and heat exchanges with a coil that circulates water from the first tank. This tank is effectively a 100 gallon (0.3785 m^3) storage tank with water from the coil entering in the middle of the tank and water to the coil exiting at the bottom. The heat pump switches off when the temperature sensor reaches 60°C (140°F).

The second tank in series is again a standard efficiency 100 gallon (0.3785 m^3) tank. The GAHP configuration feeds the restaurant recirculation loop. In both systems, the first tank is modeled as stratified whereas the second that is considered a well-mixed tank, due in part to the mixing brought on by the circulation loop.

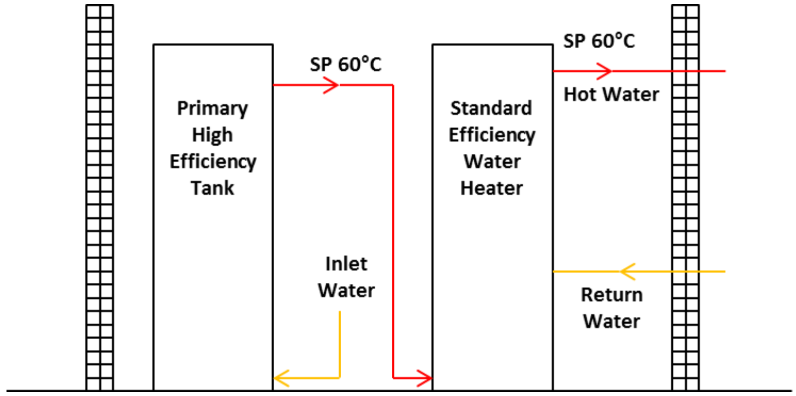

Figure 1: Standard configuration

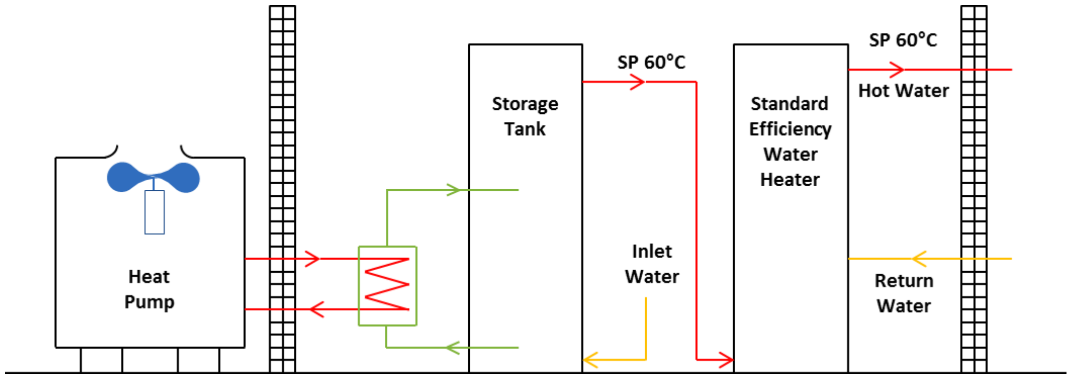

Figure 2: Gas Absorption Heat Pump (GAHP) configuration

2.1 Absorption system modeling

Development and optimization of a detailed single-effect ammonia-water heat pump was performed using the Engineering Equation Solver (EES) modeling platform (Klein 2015). Mass, species and energy conservation equations were used to analyze each component in the system. Three independent properties were required to establish state points at the inlet and outlet of each component because the working fluid is a binary mixture. Several assumptions were required to determine the concentration of the concentrated, dilute and refrigerant fluid streams. The refrigerant vapor exiting the rectifier was assumed to be a saturated vapor (quality of 1), the dilute solution exiting the desorber was assumed to be a saturated liquid (quality of 0) and the concentrated solution concentration was determined from a species balance. Heat transfer resistances were taken into account with the specification of overall heat conductance UAs for each heat exchanger. An initial set of state points were selected at an ambient and hydronic return temperatures of 8.3°C (47°F) and 37.8°C (100°F), respectively, for the design heat load of 41 kW (140 kBtu hr^{-1}). It should be noted that the hydronic return is the circulating water entering or returning to the heat pump unit. The system was then optimized to maximize coefficient of performance for the design conditions. A parametric analysis of key cycle inputs and heat exchanger sizes was completed, with the results analyzed against their impact on performance and estimated cost/reliability. The optimized baseline system was selected by 'locking' the key inputs and UA values, and was determined to have a Net Heating COP of 1.46. The Net Heating COP was calculated using Equation 1.

$$COP_{Net} = \frac{Q_{Absorber} + Q_{Condenser} + Q_{CHX}}{Q_{Gas}} \tag{1}$$

In this equation the absorber, condenser and condensing flue gas heat exchanger (CHX) duties, and total natural gas input are the only outputs and input considered. A series of pressure losses are assumed between the evaporator inlet and solution pump inlet.

The optimized cycle was then investigated over a range of ambient and hydronic return temperatures to evaluate system performance over the range of expected operating conditions. For this analysis the desorber and condensing flue gas heat exchanger combustion efficiencies vary depending on system operating conditions. Figure 3 is a plot of the Net Heating COP as a function of ambient and hydronic return temperatures. The plots show the expected trend where performance increases with decreased hydronic return temperature and increased ambient temperature. The system responds positively to the decreased hydronic return temperatures because it allows the high side to operate at a lower pressure and the absorber to produce higher solution concentrations. These operational changes result in increased refrigerant generation and flow rates. The increased ambient temperature allows for a higher low side pressure. This results in higher solution concentrations and refrigerant flow rates. Figure 4 is a plot of the cycle heating load as a function of ambient and hydronic return temperatures. The plot shows trends similar to that of Figure 3 which is expected based on the cycle COP being a function of the heating load. The COP is further reduced by 2.5% to allow the parasitic power (fan, blower, etc.) to vary with heating load.

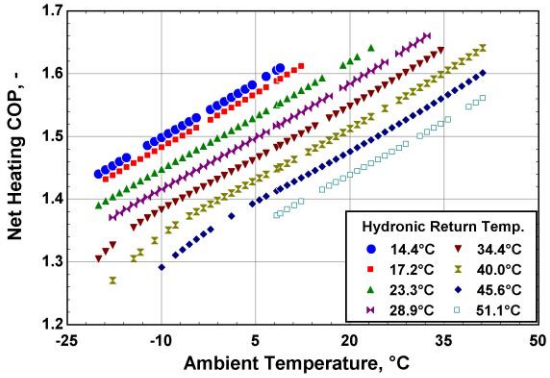

Figure 3: Net Heating COP versus Ambient and Hydronic Return Temperatures

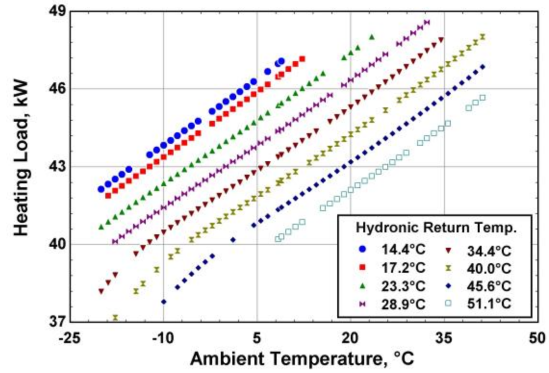

Figure 4: Heating duty versus Ambient and Hydronic Return Temperatures

To allow for use of this data in Energy Plus, a regression analysis was performed to develop a set of predictive equations based on the results presented in Figures 3 and 4. Equations 2 and 3 resulted from this analysis and the constants for each equation are presented in Table 1. As a note, the ambient and hydronic return temperature inputs for these equations are in Celsius and the output to Equation 3 is in kW.

$$COP_{NET} = a + b \times T_{hyd} + c \times T_{hyd}^2 + d \times T_{amb} + e \times T_{amb}^2 + g \times T_{hyd} \times T_{amb} \tag{2}$$

$$Q_{Heating} = a + b \times T_{hyd} + c \times T_{hyd}^2 + d \times T_{amb} + e \times T_{amb}^2 + g \times T_{hyd} \times T_{amb} \tag{3}$$

Predictive results from these equations were compared to the modeling results and the average errors for the COP and heating duty were determined to be 0.14 and 0.14%, respectively. These equations were used in the building-energy modeling software of Energy Plus to evaluate the performance of the system for a range of operating conditions and scenarios and allow for comparison with other heating systems.

Table 1: Constants for Net COP and heating load equations

	COP Equation Constants	Heat Load Equation Constants
a	1.6322	47.7539
b	-0.005	-0.146
c	-2.0545E-05	-0.0006
d	0.0057	0.167
e	-2.7973E-07	-9.3037E-06
g	1.1585E-06	3.7201E-05

2.2 Energy Plus modeling

EnergyPlus single-speed, air-source heat pump water heating coil, and the stratified tank model were used for the full service restaurant hot water heating simulation. The performance data, i.e. rated COP, water heating capacity, and normalized part load performance curves were inputted to the EnergyPlus IDF file. It was assumed there is no cyclic degradation of the absorption heat pump, as a future control will be implemented to minimize the dynamic loss.

The absorption HPWH was coupled with a stratified water tank. A skin loss coefficient per unit area to ambient temperature was defined to calculate heat loss from the hot water to the surrounding air, which is 1.7 W/m^2-K. The water tank was configured to have six nodes, i.e. six control volumes with different water temperatures, which are uniformly distributed from the top to the bottom of the tank. The return water to the HPWH was drawn from the bottom node and the heated water out of the HPWH flows to the middle node of the tank. The hot water is discharged to the second tank from the top node. The makeup temperature is from the city mains, which goes to the bottom of the tank. The sensor controlling the HPWH On/Off was placed at 1 meter up from the tank bottom. And the HPWH setting point is 60°C, which has a 2°C temperature dead band. No supplemental heaters were used in the tank. A 1-minute time step was set for the simulation. The HPWH model in Energy Plus currently has a cut-off operating switch at 5.5°C. This is because electric heat pumps perform poorly at low ambient and it is more efficient to switch to the available back up heater. This is not the case for GAHP systems but the cut-off could not be disabled in the modeling software. As a result, energy simulations were conducted six US cities in climate zones defined as hot-dry/mixed dry and hot-humid by Baechler & Love (2010) where the ambient temperature rarely goes below 5.5°C when a water draw is required. Future work will remove this cut-off switch in order to estimate the benefits of a GAHP system in more northern climates.

2.3 Water Draw Pattern

The same water draw pattern was used for all simulations in order to have a real comparison between the two water heating configurations across the southern United States (Figure 5). Fisher, D., & Pietrucha W. (2008) provide a hot water load profile for a full service restaurant with an average usage of 2100 gallons per day (7.95 m^3 per day). The data was averaged to a 15 minute period so as to function properly with the EnergyPlus time step of 1 minute. It should be noted that the purpose of this work is not to size the hot water equipment for specific tasks but to make a comparison between two possible configurations.

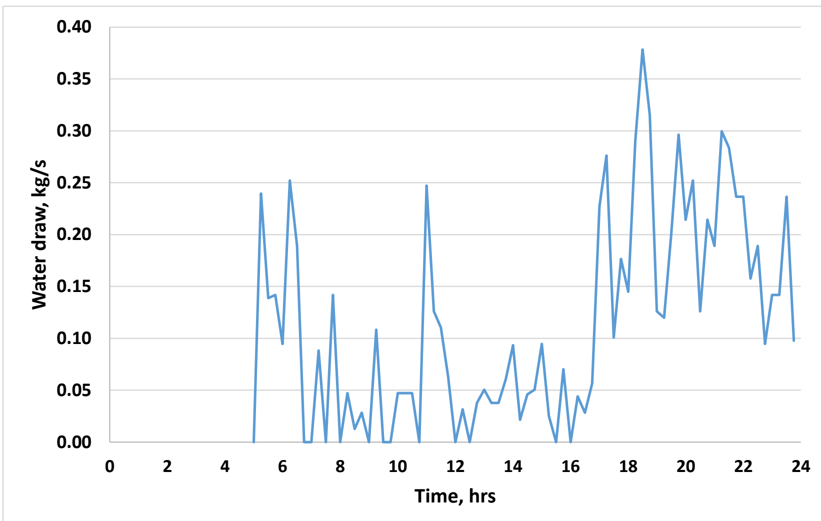

Figure 5: 15 minute period full service restaurant water draw pattern

3. RESULTS

EnergyPlus provided performance data for a full year of operation at the selected site location. Figure 6 displays the exit temperature from Tank 1 of the GAHP system for a nominal day in Houston. March 23rd was chosen because the daily average ambient temperature of 19.9 °C is close to that of the annual average. The inlet water temperature remains constant throughout the day at 7.9 °C whereas the ambient air temperature varies from 17.2 to 23.6 °C. The exit temperature from the GAHP storage tank remains close to the set point of 60 °C for much of the day. When the heavy water draw occurs the temperature drops to as low as 50.4 °C and the system relies on the second tank to reach the set point temperature. On this day, the GAHP delivered a COP, based on hot water heating to gas usage, of 1.48. The 58.3 kW high efficiency tank 1 of the standard installation ensures the set point is essentially maintained throughout the nominal day but this is achieved at a lower COP of 0.89.

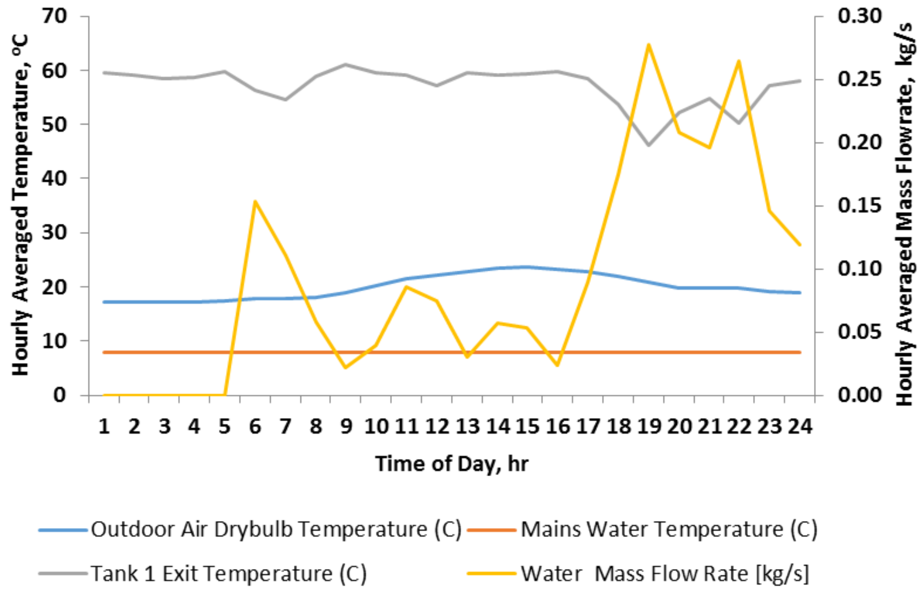

Figure 6: GAHP tank performance for a nominal day in Houston

The annual gas usage for the six cities, based on the high efficiency configuration, is shown in Figure 7. The mains water temperature differentiates the cities where Miami and Phoenix have on average the warmest inlet temperatures. Overall annual gas usage range between 450 MJ/year and 600 MJ/year. The ratio of Tank 1 usage (high efficiency) to Tank 2 usage (standard efficiency) imply the two tanks in series offer a reasonable configuration for the full service restaurant daily water draw. The second tank has the additional capacity to handle more peak demand. The equivalent annual gas usage for the GAHP system is shown in Figure 8. In this case, the values range between 275 MJ/year and 400 MJ/year. In comparison to the baseline above, the ratio of Tank 2 usage is to Tank 1 usage is higher so some care should be taken for peak water demand.

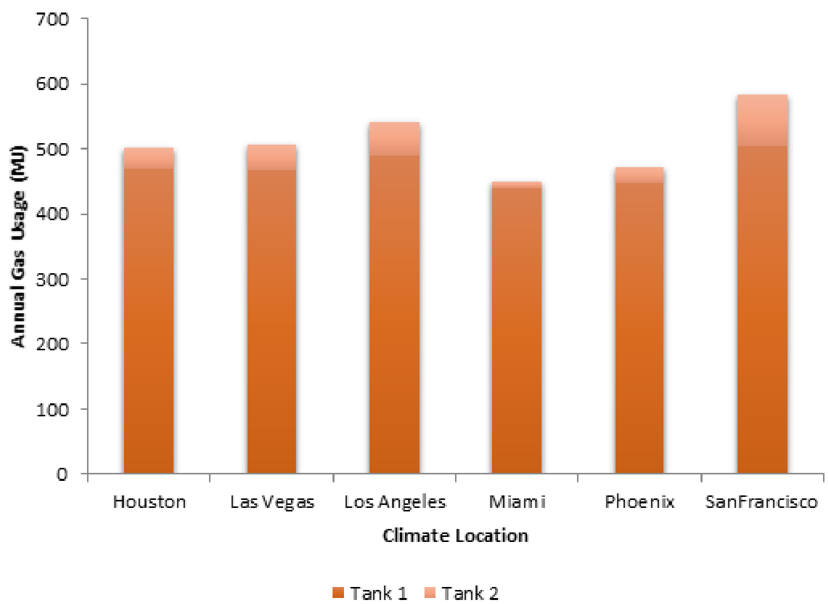

Figure 7: Annual gas usage (Mega-Joules) for the high efficiency configuration

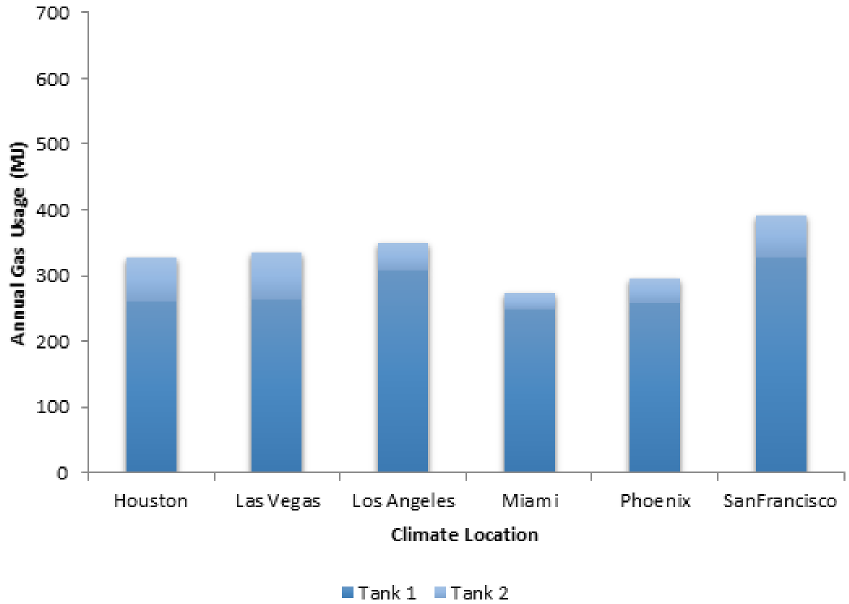

Figure 8: Annual gas usage (Mega-Joules) for the GAHP

The GAHP offers considerable gas savings which is presented in Table 2 below. The average annual gas savings for the cities investigated is 35%. These results are reiterated by a comparison of the COPs for both systems in terms of water heated to gas usage (Figure 9). As expected the GAHP system is able to maintain system level COP values above 1 while the standard high efficiency system is limited to COP values well below 1.

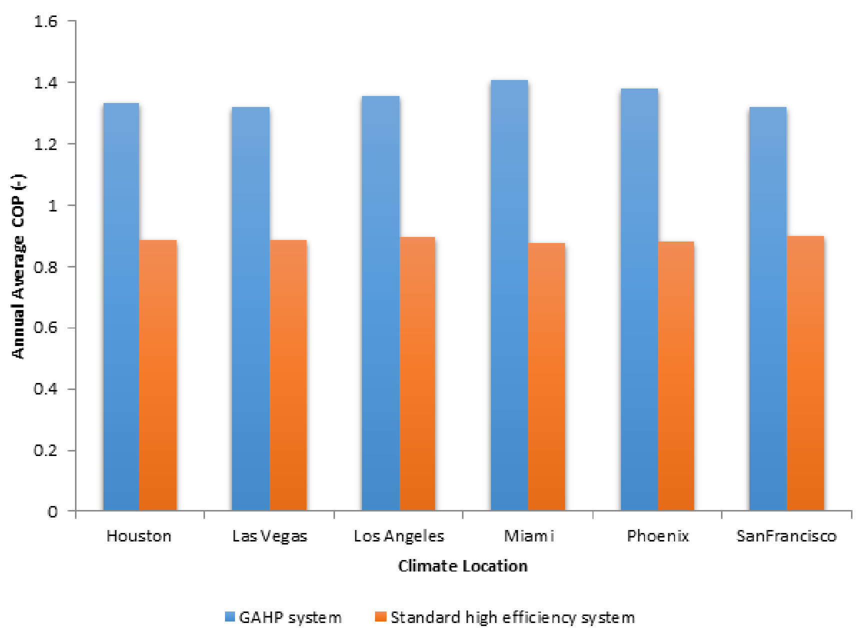

Figure 9: Comparison of Annual Average COP

Table 2: Annual Percentage Savings of the GAHP in comparison to the High Efficiency configuration.

Climate Zone Location	Annual Percentage Gas Savings (%)
Houston	34
Las Vegas	34
Los Angeles	35
Miami	39
Phoenix	37
San Francisco	33

6. CONCLUSIONS

A Gas Absorption Heat Pump (GAHP) hot water heating configuration for a full service restaurant was investigated in Energy Plus for the Southern and South Central Climate Zones. The performance of the GAHP system was compared to that of a high efficiency system utilizing a condensing gas water heater. Performance of the GAHP was very favorable in terms of annual gas energy savings for cities located in the hot-dry/mixed dry and hot-humid climate zones. Percentage of savings was between 33 and 39% with an average annual savings of 35%. In future work, the EnergyPlus Heat Pump model will be adapted for cold climate regions where ambient temperatures below 5.5 °C are to be expected but where the GAHP is also expected to offer significant energy savings.

NOMENCLATURE

a, b, c, d, g	variables	
COP	coefficient of performance	(W/W)
C	total cost	(US$)
EHP	Electric heat pump	
GAHP	Gas absorption heat pump	
HPWH	Heat pump water heater	
N	number	(–)
Q	Heat duty	(kW)
SP	set point	(°C)
T	Temperature	(°C)
UA	Overall heat conductance	(W/K)

Subscript

amb	ambient
CHX	Condensing Heat eXchanger
hyd	hydronic

REFERENCES

Klein, S. A. 2015, *Engineering Equation Solver*, F-Chart Software

EnergyPlus 8.1, “EnergyPlus Engineering Reference: TheReference to EnergyPlus Calculations”, 10/01/2013.

Fisher, D., & Pietrucha W. (2008). Case Studies in Restaurant Water Heating. ACEEE Summer Study on Energy Efficiency in Buildings (pages 9-83 to 9-93). Pacific Grove, CA: ACEEE.

Baechler, M.C., & Love, P.M. (2010). High Performance Home Technologies: Guide to Determining Climate Regions by County. Building America Best Practices Series, Volume 7.1. PNNL-17211