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Residential and Commercial Capacity Absorption Heat Pumps for Space and Domestic Water Heating Applications

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Abstract

Current gas-fired residential and light commercial (<42 kW) space and domestic water heating technologies have been approaching their fundamental limit of efficiency over the past few decades. As a result, the differentiating factors between gas-fired heating appliances has become less and less. In addition, gas appliances have been losing ground to electric heat pumps in mild climates because they offer higher primary energy efficiencies. A gas fired absorption heat pump has the ability to achieve higher coefficients of performance (COP) and provide a new upper level of performance. The current study investigates the performance of three different nominal heating capacity (3, 23.5, and 41 kW) gas-fired absorption systems designed for space and domestic water heating. Each heat pump is investigated over a range of ambient and hot water supply temperatures. Experimental results for each system are presented. The investigation shows that the energy and economic savings of these units when compared with conventional heating systems is significant.

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Keywords: Absorption, Heat pump, Space heating, Water heating, Ammonia-water

1. Introduction

Gas fired water heaters, furnaces and boilers have been approaching their efficiency limit over the past 30 years. In 2015 in the United States, 4.4 million residential gas storage water heaters, 2.8 million warm air gas furnaces and 0.098 million commercial gas storage water heaters were sold [1]. The development of a cost

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effective high efficiency gas replacement is needed but has not been realized to date. The gas absorption heat pump (GAHP) has the potential to be a replacement for standard gas appliances but current commercially available systems are cost prohibitive. This is in part because they use the complex Generator-Absorber heat eXchange (GAX) absorption cycle. In cold and very cold climates electric heat pumps (EHP) are not an option for space heating because their performance drops significantly at low ambient temperatures and results in these systems switching to a back-up heating method. This is typically electric resistance heaters which have a primary fuel coefficient of performance (COP) well below that of both heat pumps at nominal heating conditions (8.3°C ambient) and gas-fired heating systems.

Three gas-fired single-effect (SE) ammonia-water absorption heat pumps of varying heating capacities (3, 23.5 and 41 kW) are the focus of this paper. Figure 1 presents the SE cycle used for all three systems. In the desorber the heat of combustion is used to generate refrigerant from concentrated (ammonia rich) solution. The vapor refrigerant flows through the rectifier where water is selectively condensed before flowing through the condenser. Heat is exchanged with coupling water during the condensation of the refrigerant in the condenser. The liquid refrigerant then flows through the refrigerant heat exchanger where it is subcooled before flowing through an expansion device from the high to low pressure side of the system. Heat is exchanged with the ambient during the evaporation of the refrigerant in the evaporator. Low pressure refrigerant then flow through the refrigerant heat exchanger where it recuperates heat from the high pressure refrigerant. Refrigerant then flows to the absorber where it mixes with the dilute (ammonia weak) solution and is absorbed by the solution in an exothermic process. The heat of absorption is exchanged with the coupling water. The concentrated solution exiting the absorber is pumped to the high pressure side of the system where it recuperates heat in the rectifier and solution heat exchanger before entering the desorber. Dilute solution exiting the desorber exchanges heat with the concentrated solution and the flows through an expansion device from the high to low pressure side of the system. The solution then mixes with the refrigerant from the refrigerant loop. The combusted gas products exiting the desorber exchange heat with the coupling water in addition to the heat exchanged by the absorber and condenser.

The fundamental design of all three systems is the same and components are simply scaled to meet the required heating capacity. The SE cycle was selected because it is less complex than the GAX cycle, is still able to maintain relatively high heating COP values for a large range of operating conditions, and is easier to modulate to match the heating demand. Experimental performance of each system is presented and compared with results of thermodynamic system models. The potential for these systems to be cost-effective is then discussed.

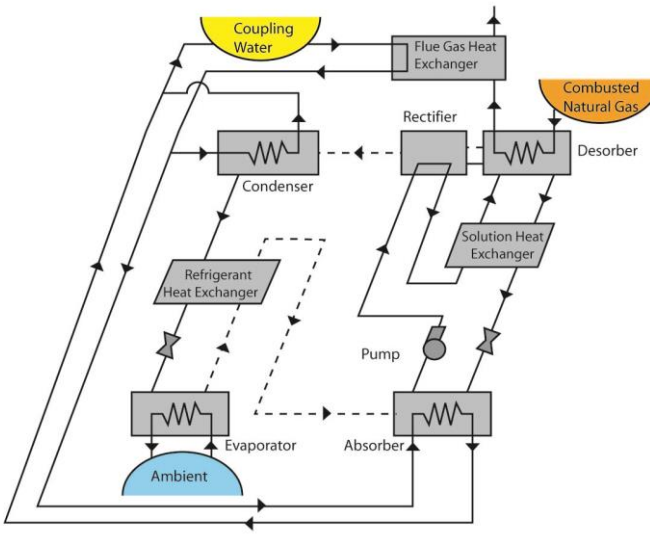


Fig 1. Gas-fired single-effect absorption heat pump

2. Modeling and Analysis

Thermodynamic cycle models developed as part of previous studies [2, 3, 4] were used to evaluate the experimental performance of each heat pump system. The development and optimization of each single-effect ammonia-water heat pump was

performed using the Engineering Equation Solver (EES) modelling platform [5] and the model development was performed in steps similar to those outlined by Herold et al. [6]. Mass, species and energy conservations were applied to each component. Three independent properties were used to fix each state point of the ammonia-water loop because it is a binary mixture. Liquid saturation was assumed at the absorber solution outlet, condenser refrigerant outlet, desorber solution outlet and for the rectifier reflux (q = 0). Vapor saturation was assumed at the desorber vapor outlet and rectifier vapor outlet (q = 1). Closest approach temperature (CAT) assumptions were initially made for the absorber, condenser, desorber and evaporator. Heat exchanger effectiveness assumptions were initially made for the solution and refrigerant heat exchangers (0.97 and 0.93, respectively). Vapor exiting the rectifier was initially assumed to have a purity of 0.9985. These assumptions allowed for the calculation of overall heat conductance UA values for each component. The CAT, effectiveness and refrigerant purity assumptions were replace with assumed UA values after each baseline system was established. Other key remaining assumptions included a fixed refrigerant glide in the evaporator, an assumed pressure loss between the evaporator inlet and the absorber outlet, and solution flow rates that were a function of pressure difference between the high and low side. The fixed refrigerant glide is a representative result of an expansion device that is designed to provide a fixed glide over a range of operating conditions. The variable solution flow rate is representative of the pumping requirement of absorption systems. The amount of solution returning to the pump depends on operating conditions because dilute solution flow is restricted by a fixed orifice. The cycle models with fixed UA values were investigated over a range of hydronic and ambient temperatures. The performance results from each model are used to provide a point of comparison between target performance values and measured experimental results.

Figure 2 presents two plots. The first is a plot of modelling results for the 23.5 kW residential space and water heating SE absorption heat pump. As a note, all three of the cycle models produced similar results. The plot presents the $COP_{GAS,HHV}$ of the heat pump versus the hydronic return temperature for a range of ambient temperatures. The hydronic return temperature is the temperature of the hydronic fluid entering the heat pump. The plot shows that performance decreases as the ambient temperature decreases and hydronic return temperature increases. The ability of the heat pump to extract heat from the ambient is reduced as the ambient temperature is reduced. The increased hydronic temperature results in less favourable operating conditions for the absorber and condenser. This has several negative impacts including reduced solution concentration and an increased dilute solution flow rate. The second plot presents a performance comparison between the 23.5 kW SE cycle model and a commercially available 18 kW GAX system [7]. The plot presents $COP_{GAS,HHV}$ as a function of ambient temperature for two hydronic supply temperatures (35 and 55°C). As a note, the Gas Utilization Efficiency (GUE) values reported for the 18 kW GAX system [7] were converted to $COP_{GAS,HHV}$ by multiplying by 0.89 to account for the change from lower to higher heating value. The plot shows that the theoretical performance of the SE model is greater than that of the actual GAX system. Modelling performance being greater than that of actual system performance is typically not surprising but considering the theoretical performance of a GAX cycle is much greater than that of a SE cycle, it would be expected that GAX system perform at or above that of the SE cycle for similar operating condition. It is also interesting to note that the GAX system experiences a drop-off in performance compared to the SE model as the ambient temperature is reduced. Some reduction in performance is to be expected at reduced ambient temperatures because the cycle model does not account for heat loss to the ambient which is expected to increase with decreased ambient temperature. The degree to which GAX system performance reduces may be accredited to the fact that as GAX cycles deviate from their design condition, they typically revert to performance of a similarly sized SE system. This is because the amount of heat recuperation in the Generator-Absorber heat exchanger is reduced.

Steady state testing was conducted with each system to investigate prototype performance for a range of ambient and hydronic return temperatures. Heating load and the gas based Coefficient of Performance (COP) were the main criteria of interest. Equation 1 was used to calculate the heating load of each prototype. The hydronic temperature was measured at the inlet and outlet of the system to determine the temperature gain in

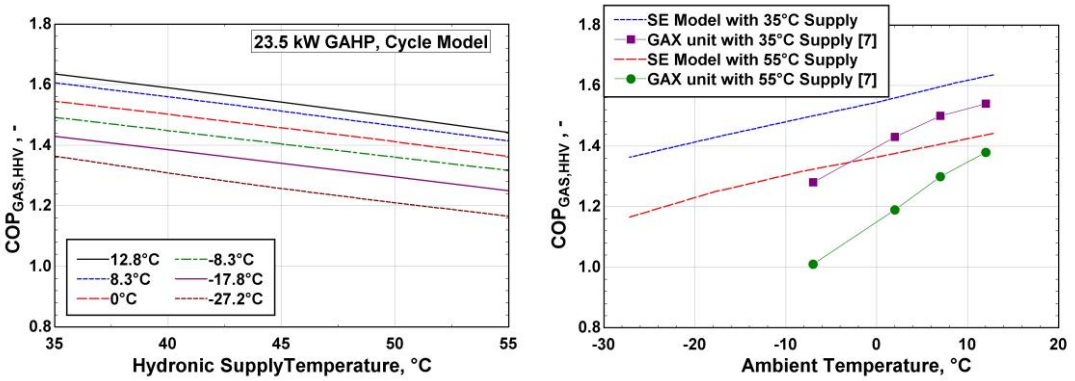


Fig 2. 23.5 kW SE GAHP cycle modelling results and comparison with 18 kW GAX GAHP [7]

the hydronic loop across the system.

$$Q_{Hydronic} = \dot{m} \times cp \times (T_{Hydronic\ Out} - T_{Hydronic\ In}) \quad (1)$$

Equation 2 was used to calculate the gas based COP value for each system. The numerator is the total heat supplied to the hydronically coupled water loop. For the 23.5 and 41 kW systems this includes heat additions from the absorber, condenser and flue gas heat exchanger. For the 3 kW system this included heat supplied by the absorber and condenser. The 3 kW also contains a flue gas heat exchanger but it is integrated into the water storage tank and is not accounted for in the temperature rise experienced by the hydronic loop. The denominator is the total gas input to the system on a higher heating value (HHV) basis.

$$COP_{GAS,HHV} = \frac{Qhydronic}{Q_{Gas}} \quad (2)$$

It should be noted that the HHV is used in the calculation of all COP values reported in this paper. These values are typically 10-11% lower than those reported using the Lower Heating Value (LHV) of natural gas. Electrical power is not included in this COP calculation. System performance was investigated over a range of ambient and water return temperatures.

3. 23.5 kW GAHP Residential Space and Water Heater

A 23.5 kW nominal capacity GAHP was developed to provide space and water heating for residential applications. The system size was selected based on typical heating loads of homes in colder climates in the U.S. Figure 3 presents images of one of two second generation prototypes fabricated. The image on the left is an external view of the unit which has dimensions of 1.19 × 0.97 × 1.12 meters. The evaporator is the largest component of the system and occupies 64% of the total footprint of the unit. The unit is intended for installation adjacent to a home and because of its outdoor installation it provides heated water for space heating (forced air or radiant) and domestic hot water. The image on the right is the main ammonia-water SE absorption heat pump sealed system which is located at one end of the prototype and occupies 36% of the footprint of the unit.

Performance of two first generation prototypes was presented by Garrabrant et al. [8]. Both units performed within 6% of design COP and heating load at design hydronic return and ambient conditions of 37.7



Fig 3. Images of the exterior of the 23.5 kW prototype (left) and the absorption sealed system (right)

and 8.3°C, respectively. The units were designed to maintain a set hydronic supply or delivery temperature when a call for heat is present. The systems were demonstrated to be capable of 4:1 modulation and performance data showed that lower firing rates had no significant impact on system performance. The performance data of the prototypes was compared to that of commercially available GAX GAHP systems and cold climate electric heat pumps. The SE prototypes were found to have similar performance of the more complex GAX systems and outperformed the cold climate electric heat pumps below a temperature of 0°C.

Several system and component level limitations were identified during the experimental evaluation of the first generation prototypes. Adjustments were made to the design of the second generation prototype to address these limitations. One such adjustment was made to the rectifier component and resulted in higher ammonia purities exiting the rectifier and flowing through the refrigerant loop. Refrigerant purity is critical in maximizing system performance as it directly impacts the glide required to reach evaporator outlet qualities above 0.95. Larger glides require lower inlet temperatures, for a giving ambient, which in turn require lower operating pressures. Lower operating pressures limit the solution concentration exiting the absorber and the ammonia available in the desorber.

3.1 Performance testing

The performance of the second generation 23.5 kW prototypes was investigated over a range of hot water return and ambient temperatures. Depending on the application (space and/water heating), the hydronic water return temperature is expected to range from 15 to 60°C and the ambient could range from -29 to 40°C. It is therefore important to investigate performance over a range of operating conditions within these temperature constraints.

Figure 4 presents four plots. The top two plots show $COP_{GAS,HHV}$ as a function of hydronic return (inlet) temperature for ambient temperatures of 8.2 and 0°C. The bottom two plots show heat duty as a function of hydronic return temperature for ambient temperatures of 8.2 and 0°C. Experimental data is plotted against the optimized SE cycle model in all four plots. The plots show that both units perform near the predicted SE model performance for the range of ambient and water temperatures investigated. At the design ambient and return

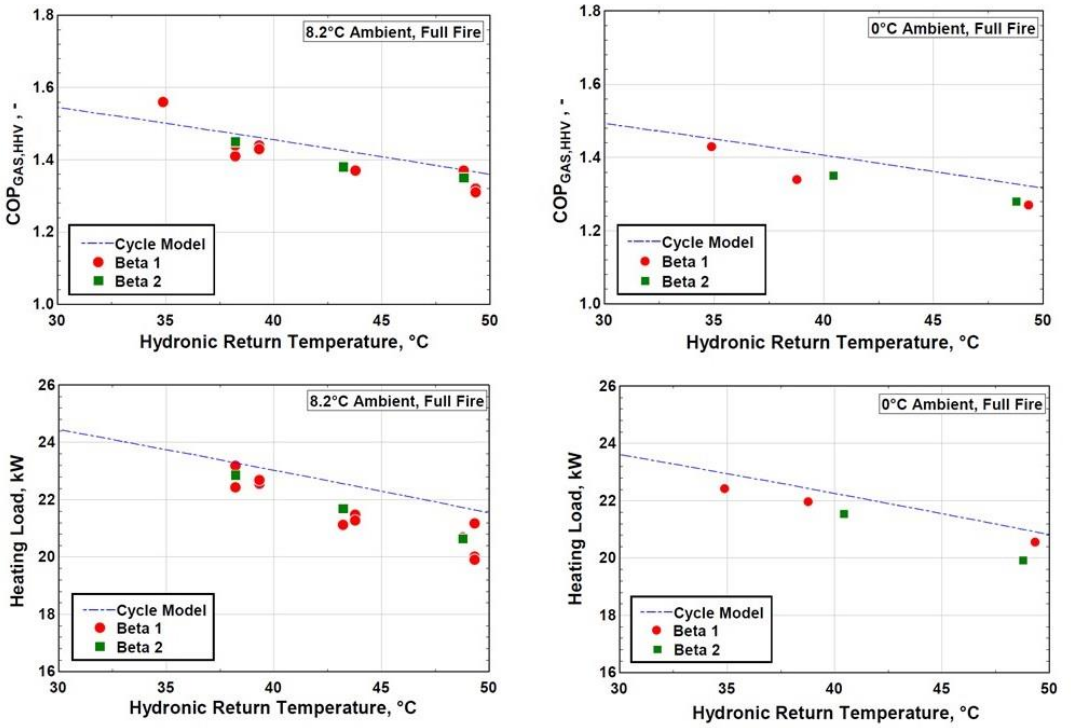


Fig 4. Performance results of the second gen. prototypes at 8.2 and 0°C

water temperatures of 8.3 and 37.8°C, respectively, both prototypes operated within 4% of design (design $COP_{GAS,HHV}$ of 1.45). This was an improvement to the first generation prototypes [8] and is a result of design changes that led to improved rectification. The discrepancy between prototype and cycle model performance can be attributed to factors experienced by the actual systems that are not accounted for in the cycle model. This includes heat loss from the system to the ambient. Any heat that is lost to the ambient cannot be recuperated or used to provide heat to the hydronic loop. This loss is more significant at lower ambient temperatures where the temperature difference between the system and the ambient is larger.

3.2 AFUE Testing

Performance of one of the prototypes was investigated using the ANSI Z21.40.4 test method to estimate the Annualized Fuel Utilization Efficiency (AFUE) of the second generation gas absorption heat pump prototypes. It should be noted that testing was not performed in an ANSI certified test facility but that steps were taken to perform the testing as close to possible at the specified test conditions. The ANSI Z21.40.4 requires a series of steady state tests to be performed at full and part loading to calculate the AFUE. The AFUE is the ratio of useful heat delivered by a system over a complete heating season to the gas consumed to drive the unit. As a note, the AFUE does not include the electricity consumption in the calculation. Performance data used in the ANSI bin calculation is presented in Table 1. The estimated AFUE for Climate Region IV, which corresponds to an outdoor design temperature of -15°C and 5643 bin hours, was 141%. The estimated AFUE for Climate Region V, which corresponds to an outdoor design temperature of -23°C and 6956 bin hours, was 149%. It should be noted that typical high efficiency gas heating systems are limited to AFUE values between

Table 1: Performance data from AFUE testing

Rating Point	Ambient Temp, °C	Hydronic Supply, °C	Hydronic Return, °C	Firing Rate, kW	Heating load, kW	$COP_{GAS,HHV}$	Total Electric Usage, kW
1	8.2	38.4	35	4.8	7.5	1.56	0.29
2	1.6	38.2	35	4.9	7.2	1.46	0.29
3	-8.2	37.9	35	4.5	5.9	1.32	0.32
4	-8.3	40.9	35	9.0	12.4	1.37	0.36
5	1.6	45.1	35	15.6	22.4	1.43	0.49
6	-8.3	45.2	35	16.6	21.9	1.32	0.52
7	-13.9	44.8	35	16.3	21.1	1.30	0.53

90 and 98.5%. This highlights the fact that the GAHP offers a substantial improvement in heating efficiency when compared to conventional gas heating systems.

3.3 Modulation

The ability of the space and water heating SE absorption heat pumps to modulate is critical to their success. This is because space heating requirements can vary significantly depending on the time of year. If modulation of the GAHP is limited, the unit will be forced to cycle on and off to meet the heating requirements of a house during the warmer cooling months. Cycling will reduce overall system performance because the performance of the GAHP is limited during its start-up period. Ideally, the unit would modulate to meet the heating requirement for the building and provide heating continuously. It should be noted that the ability of a SE cycle to modulate while maintaining high levels of performance is greater than that of a GAX cycle. This is because the GAX cycle requires higher desorber temperatures that are harder to achieve at reduced firing rates. In these cases, performance of the GAX system will reduce to that of SE cycle with similarly size heat exchangers.

Modulation of 4:1 was achieved with the first generation 23.5 kW SE GAHP prototypes [8]. This was repeated with the second generation prototypes and is highlighted in the performance data presented in Table 1 at the ambient temperature of -8.2°C. The performance results in the table show that there is limited impact to $COP_{GAS,HHV}$ as a result of modulating the system down. Instead performance actually increases as the firing rate is reduced to 56% of design and performance at 100% fire is the same as at 28%.

4. 41 kW GAHP Commercial Water Heater

A 41 kW nominal capacity GAHP was developed to provide hot water for commercial water and space heating applications. The system capacity was sized based on typical heating requirements for commercial applications such as full service restaurants that use large amounts of hot water on a daily basis.

Geoghegan [9] presented the development of a first prototype in a study that included cycle modeling, breadboard testing, and prototype fabrication and testing. The first prototype has dimensions of 1.24 × 1.68 × 1.65 meters. The system performed within 10% of design COP_{Gas} and heating load at design hydronic return and ambient conditions of 37.7 and 8.3°C, respectively, and was demonstrated to be capable of 3:1 modulation. Identified system limitations included below design rectifier performance and a high pressure loss through the hydronic side of the system (140 kPa at 53 lpm). It was also determined that the evaporator coil was roughly 25% oversized. This did not have a significant impact on system performance but was the main driver of the overall system size as it occupied 60% of the total system volume.

Geoghegan et al. [4] conducted a modeling study to investigate the potential energy savings provided by a GAHP commercial water heater compared to a high-efficiency (condensing) gas storage water heater. The daily draw pattern investigated was specific to a full service restaurant and had a 24 hour draw total of 7950 liters. The study was limited to six cities in the South and South Central climate zones of the US. The average annual gas based COP was determined to be 1.35 and 0.89 for the GAHP and conventional high efficiency system, respectively. On average the GAHP offered a 35% energy savings compared to the high efficiency storage water heater.

A second prototype was developed with modifications to address performance limitations of the first prototype. Design modifications to six of the eight heat exchangers were performed and this resulted in size reductions in five of the eight components. Figure 5 is an image of the second prototype of this GAHP system. In addition to the performance modifications, the envelope of the packaged system was greatly reduced. This is mainly because the size of the evaporator coil was reduced by 25%. The second prototype has dimensions of 1.41 × 1.02 × 1.53 meters. It is 70% of the volume of the first prototype and occupies a footprint that is 23% smaller.

4.1 Performance Testing

Steady state testing was performed at ambient temperatures of the 8.2°C and 0°C for a range of hydronic inlet temperatures (37 to 49°C). Figure 6 presents $COP_{GAS,HHV}$ comparison plots for the cycle model and experimental data. The plots show that performance was slightly below cycle model targets (within 6%) for most of the data collected and is within 12% for all of the data collected. Performance was within 4% of design COP at design hydronic return and ambient conditions of 37.7 and 8.3°C, respectively. The plot also shows that the difference between the measured and modelled performance increases as the hydronic return temperature increases. This is a result of changes that are not accounted for in the modelling including increased heat loss due to higher operating temperatures.

Overall the prototype experienced performance gains that were expected based on the component modifications that were performed. The pressure loss through the hydronic side of the system was reduced to 50 kPa at 53 lpm by design modifications to the condensing flue gas heat exchanger and refrigerant condenser. Refrigerant purity limitations experienced with the earlier system was addressed with modifications to the rectifier component. This included the elimination of liquid carryover which was the main cause for the deviation between modelled and measured COP.



Fig 5. Image of second prototype 41 kW commercial water heating GAHP

Reducing the evaporator coil size did result in an increased pressure loss from the evaporator inlet to the absorber outlet (60 kPa) but greatly reduced the overall size of the unit and the projected cost of the coil. The higher than design pressure loss has been identified as the main factor that is now

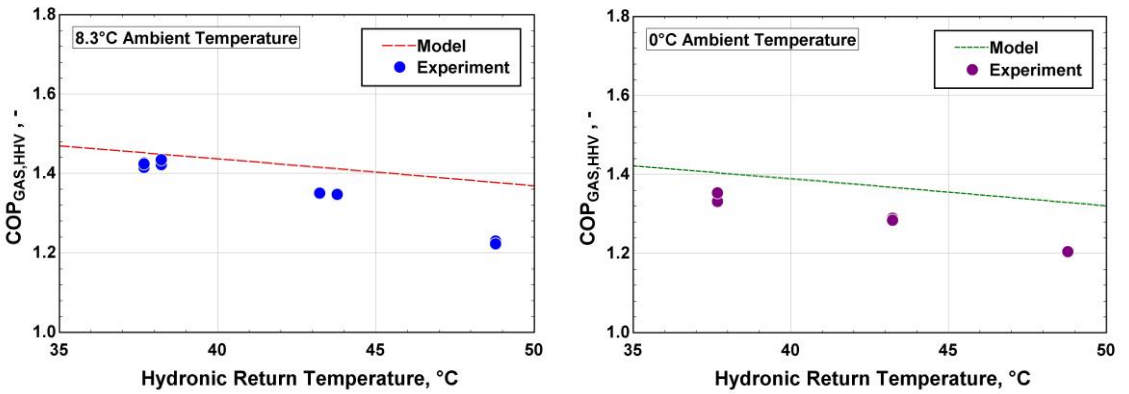


Fig 6. Experimental and modelling results for a range of hydronic return temperature at ambients of 8.3 and 0°C

limiting the system from meeting its design COP at the design hydronic return and ambient conditions of 37.7 and 8.3°C, respectively. This highlights a balancing act between meeting desired size constraints and performance targets.

5. 3 kW GAHP Residential Water Heater

A 3 kW nominal capacity GAHP was developed by Garrabrant et al. [2] to provide water heating for residential applications. The study of two absorption heat pump systems for residential water heating applications; a single-effect and GAX cycle were investigated. The systems were modelled to investigate performance over a range of potential water temperatures. Both systems showed significant performance improvements when compared to conventional water heating technologies. Component models were then used to design heat and mass exchangers that were investigated as part of a full system on a breadboard test facility. Testing at baseline design conditions in the single-effect cycle configuration resulted in a system *COP* within 3 percent of the model predictions. Significant limitations were noted in the GAX cycle configuration resulting in poor performance when compared to the system model.

Garrabrant et al. [10] presented the performance testing of three prototype direct gas-fired ammonia-water absorption heat pump water heaters. Steady state performance was investigated for the range of temperatures experienced when heating water for residential use and was indicative of an Energy Factor (EF) of 1.2-1.3. The EF is a rating developed by the United States Department of Energy that allows for the comparison of energy efficiency between different types of residential water heating systems using a 24 hour use test. The authors discussed the importance of intelligent controls and the ability to operate over a significant range of high to low side pressure differences. They presented a comparison of primary energy consumption and operation cost between the gas-fired heat pump and commercially available units. This comparison showed that a gas fired heat pump would minimize primary energy use and operating costs compared to conventional water heat designs.

Glanville et al. [11] presented field test results of a second-generation direct gas-fired ammonia-water absorption heat pump water heater. The field test was conducted over a 10 month period and showed the system is able to operate over a large range of ambient and water temperatures. Laboratory and field test data

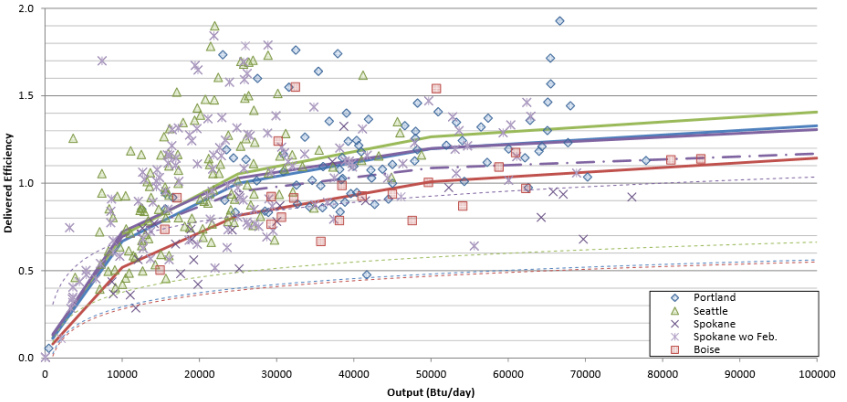


Fig. 7. Delivered GAHP water heat efficiency as a function of daily heating output [12]

indicated that a 1.3 Uniform Energy Factor (UEF) could be achieved. The UEF is an updated version of the EF rating system where the test method has been updated to address shortcomings of the original test method and rating system. This rating system went into effect in the U.S. in July of 2015. The installed GAHP showed considerable efficiency improvements and energy savings when compared to conventional gas water heating systems.

Glanville et al. [12] presented the field evaluation of four third-generation pre-commercial direct gas-fired ammonia-water absorption heat pump water heaters installed in high water use homes in the Pacific NW of the United States. Units were designed to allow for easy installation with connections similar to those of a conventional gas tank water heater. Performance of the units was monitored over a period of 9 months and showed the systems could offer energy savings of 50% or more compared to conventional gas tank water heaters. Figure 7 is a plot from this study that showed the delivered daily efficiency of the GAHP water heater as a function of daily heating output. The solid lines on the plots are the average GAHP delivered efficiency for each field test site. The plot indicates that a minimum amount of heating output/hot water usage is needed to obtain the energy savings benefits of the GAHP water heater.



Fig 8. Image of third generation GAHP water heater

5.1 Performance testing

Steady state performance testing was performed on several third generation GAHP residential water heaters. Figure 8 is an image of one of the GAHP heat pump water heaters investigated. The data was analyzed to determine $COP_{Gas,HHV}$ and the heating load. Performance data was recorded for a range of water inlet (29-57°C) and ambient (5-20°C) temperatures. Figure 9 is a plot of $COP_{Gas,HHV}$ and heating duty for the range of testing performed. As a note, the experimental heating load does not include the heat input from the condensing heat exchanger because it is integrated into the water storage tank.

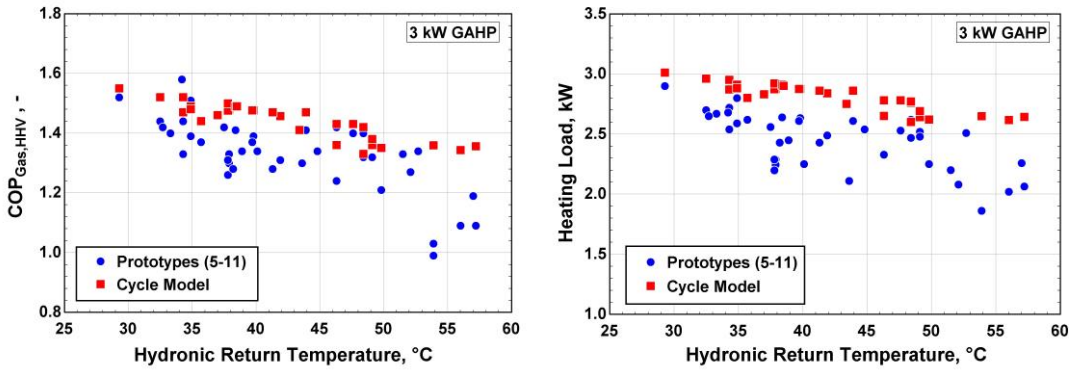


Fig 9. GAHP Water Heater Gas COP and Heating Duty (Experimental vs. Model)

The plots highlight the typical performance characteristics of a gas absorption heat pump. Performance of the GAHP decreases as the hydronic return temperature increases. This is because higher water temperatures in the condenser increase the high side pressure which reduces the amount of refrigerant generated and increases the dilute solution flow rate. The higher water temperatures in the absorber reduce the ammonia concentration in the concentrated solution.

The performance of the prototype GAHP units is generally within 15% of the model predictions. There is some variation in performance between the different units investigated.

6. Potential Energy and Economic Savings

All of the gas absorption heat pumps under development have the potential to significantly reduce energy usage for their respective applications (residential water heating, space heating and commercial water heating). The residential water heating system can offer an energy savings of greater than 50% [12] which translates to a potential operating cost savings of 50% for the end user. The residential space heating system achieved an AFUE of 141% while the highest efficiency conventional heating technologies are limited AFUE values of 90-98.5%. For a heating season in US Region 4, the GAHP is expected to use 28597 kWh of gas energy to provide 40622 kWh of heat. A conventional system with an AFUE of 90% would use 45136 kWh of gas energy. Using a GAHP instead of the conventional system would result in an annual energy savings of 16540 kWh and an annual operating cost savings of $ 994 (US) assuming $17.62 per 1000 ft^3 as of August 2016 [13]. The commercial water heating GAHP has the potential to reduce energy usage by 35% in a full service restaurant application [4] where water heating accounts for 16% of the total energy usage [14]. Annual gas energy savings ranged from 48600 to 55600 kWh in the study by Geoghegan et al [4] and this equates to an annual operating cost savings range of $2900 to $3400 (US). It is evident that all of these systems have the potential to greatly reduce energy usage and operating cost.

7. Conclusions

The development and experimental evaluation of three different capacity (3, 23.5 and 41 kW) gas absorption heat pumps was presented. A review of prior work showed that each system is at a different stage of development and testing. The 3 kW residential water heating system has progressed through three generations

of development, and has undergone extensive laboratory and field testing. Laboratory and in-situ data has indicated that the system can achieve a UEF value of between 1.2 and 1.3. Field testing has highlighted that the system can offer an energy savings above 50% when compared to conventional gas water storage systems. The 23.5 kW space heating system has progressed through two generations of development and has undergone extensive laboratory testing. This testing has included steady state testing, AFUE testing where the unit achieved a value of 141%, and active defrost testing. Testing has shown that the second generation unit is performing close to design for the range of conditions investigated. The 41 kW system has undergone the first stage of development and is currently undergoing extensive laboratory testing. Performance testing to this point is limited and the unit is performing within 10% of cycle model design.

The potential energy and operating cost savings of these units is significant with the 24.5 kW system offering an annual energy and operating cost savings of 16540 kWh and $ 994 (US) when compared to a high efficiency conventional residential space heating system in climate region IV in the United States. The 41 kW has the potential to offer an annual energy and operating cost savings of 55600 kWh and $3400 (US) compared to a high efficiency conventional commercial water heating system in use at a full service restaurant in the South and South Central regions of the United States. The benefits and savings of using gas absorption system pumps can be realized in many space and water heating applications and are not just limited to these examples.

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