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Performance of a State-of-Art Packaged Heat Pump for Residential Space Conditioning and Hot Water

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Abstract

Residential space conditioning and hot water loads in the US are typically decoupled and served by separate equipment. A residential home in four and above IECC climate zones can have two to three systems, typically a gas furnace for space heating, a gas-fired storage water heater, and an electric air conditioning (A/C) unit for space cooling. The decoupled nature of these individual systems prevents the units from working in a tandem nature, limiting the benefit of shared system costs and internal heat recovery. In addition, gas-fired furnaces and water heaters operate at less than 100% thermal efficiencies, insufficient for most CO2e emission reduction schemes. This manuscript summarizes data and key findings from the laboratory evaluation of a hybrid heat pump, a packaged Thermal Heat Pump (THP), and an Electrical Air Conditioning (EAC) unit for the combined operation of space conditioning and water heating. The packaged heat pump prototype achieved an AFUE of 121.6% and SEER of 8.

Keywords: Gas Absorption Heat Pump; Electric Air Conditioning, Space Conditioning, Water Heating, Packaged Heat Pump

1. Introduction

The annual heating needs are more significant than the yearly cooling needs for residential buildings in most climate regions in the US [1]. Gas-fired appliances predominantly provide US residential space and water heating. As of the 2018 EIA report [2], more than 26 million homes in the cold climate regions of the US are heated by gas-fired furnaces, water heaters, and boilers. These gas-fired appliances have approached their thermodynamics limits in terms of efficiency (<100%), and any further improvements will have diminishing returns. The most recent residential energy consumption survey data highlighted that the total natural gas consumption for residential space and water heating is 38 billion therms, 68% of all residential heating in the US [2]. Natural gas is primarily consumed in cold and very-cold climate regions, with more than 5,400 heating degree days (HDD) and more than 9,000 HDD, respectively.

Electric Heat Pumps (EHP) have recently grown in demand and popularity due to policy changes and push for electrification and low-carbon initiatives. EHPs, primarily as air-source heat pumps (ASHP), are in demand in milder climates for new retrofit homes, where local electricity rates are competitive (namely in the south and the pacific northwest). The significant advantages of ASHPs are (a) the ability to reduce heating costs and (b) the integration of cooling and heating with a reversible heat pump. Improvements in ASHP technology and new constructions have driven the adoption of ASHP further. As of 2015, more ASHPs are being shipped than gas furnaces and electric resistance heating [3].

In cold climates, where gas heating is well established, ASHP adoption has been slow because: (a) highly efficient condensing gas-fired appliances are well established and can be cost-effective, (b) drop-off in

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Coefficient of Performance (COP) and capacity at cold ambient temperatures, and the requirement for supplementary resistance heating, and (c) the initial cost of owning a cold-climate ASHPs (ccASHPs).

Therefore, the current market and changes in regulatory requirements have created the pathway and need for gas heating equipment with COPs higher than 100%, namely Gas Heat Pumps (GHP). The air-source Gas-Fired Absorption Heat Pump (GAHP) uses natural gas as energy input to extract heat from the outdoors to a conditioned indoor environment. GAHP products and prototypes in the market use the vapor absorption cycles where the sorbent is in the liquid phase and work on the principle that by combining refrigerant/working fluid with a sorbent, significantly less input energy is required to raise refrigerant pressure. During these cycles, thermal energy input is necessary to "desorb" the refrigerant/working fluid from the sorbent to yield a high-pressure vapor. At the core of the GAHP is a group of heat exchangers, vessels, and a pump that comprise the 'thermal compressor." LiBr/H2O (chilling) and NH3/H2O (heating) are common absorptions working fluid pairs. GAHP for heating is a developing technology with a notable advantage in heating performance in cold climates that exceeds ASHPs performance. In a single-effect vapor absorption cycle, approximately 40% of the heat delivered to the building is from the ambient ("heat pumping effect" and the remaining is from internal heat recovery [4]. For this reason, GAHP's in heating mode is much less sensitive to cold conditions and commonly do not require supplementary heating.

In the majority of residential applications in the US, space conditioning and hot water loads are typically decoupled and served by different types of equipment. A residential home in four and above IECC climate zones can have two to three types of equipment, typically a gas furnace for space heating, a gas-fired storage tank for water heating, and an electric air conditioning (A/C) unit for space cooling. The decoupled nature of these individual systems prevents the units from working in a tandem nature, limiting the benefit of shared system costs and internal heat recovery. Previous studies have described developing and demonstrating gas-fired "combi" systems for space and water heating, achieving a projected 140% AFUE while reducing water heating energy consumption and emissions by up to 50% [5–7]. One approach to address these challenges is an optimized and cost-effective packaged heat pump unit for combined space conditioning and water heating. The packaged unit offers advantages by coupling the space conditioning and water heating with a single unit to reduce the overall equipment and installation cost and operate at efficiencies of more than one.

This manuscript summarizes data and key findings from the lab evaluation of a hybrid heat pump, a packaged Thermal Heat Pump (THP), and an Electrical Air Conditioning (EAC) unit for the combined operation of space conditioning and water heating.

2. Concept

This effort is to take a pre-commercial GAHP product that only provides whole-house heating and domestic hot water (DHW) and add a cost-effective cooling function to provide space conditioning and water heating in a single packaged unit. Figure 1 illustrates the 4-pipe configuration of the hybrid GAHP–EAC (system for space conditioning and water heating. The GAHP–EAC systems consist of (1) an air handling unit (AHU), (2) a DHW storage tank, and (3) an outdoor unit. The outdoor unit is a four-pipe hydronic system that provides heating and cooling with two independent systems, a 3:1 modulating GAHP and a standard two-stage EAC system through a refrigerant/water heat exchanger. The outdoor unit heating and cooling streams are connected to an indoor hydronic coil AHU with internal pumps and three-way valves to independently manage space conditioning and water heating processes. The AHU system provides heating or cooling based on two-stage 24 VAC thermostat calls and water heating via a separate recirculating loop connected to the indirect storage water heater controlled by an aquastat.

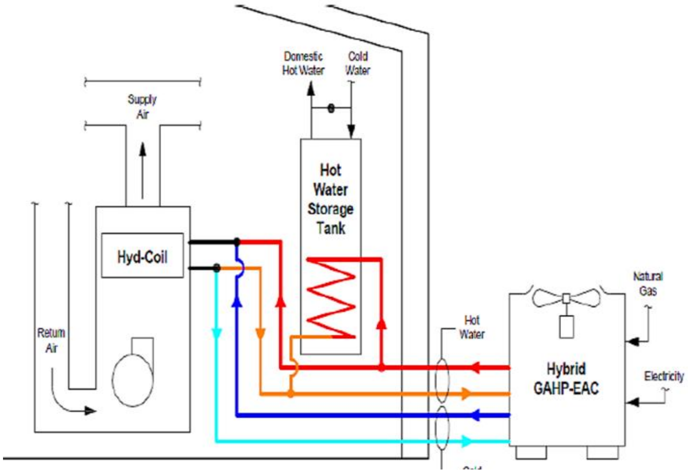

Figure 1: The hybrid GAHP-EAC concept for combined space conditioning and water heating

The GAHP is based on an economical single-effect, ammonia/water absorption cycle. With previously estimated Annual Fuel Utilization Efficiency (AFUE) of >140% [7,8]. Acknowledging that a reversible GAHP product is neither novel nor efficient compared to modern electrically-driven Air-Conditioning (A/C) technologies, this effort sought to add a vapor compression module using R410a as the refrigerant to provide cooling. Table 1 highlights the hybrid GAHP-EAC system specifications. The prototype system is illustrated in Figure 2.

Table 1: The hybrid GAHP-EAC specifications.

Outdoor Unit			
Thermal	**Heating**	Maximum capacity at 47°F	40 kBtu/hr (3.3 RT)
		Modulation	3:1
		Nominal water flow rate	4 GPM (15.1 lpm)
		Estimated AFUE	140%
	Cooling	High-stage capacity at 85°F	19 kBtu/hr (1.6 RT)
		Low-stage capacity at 85°F	15 kBtu/hr (1.25 RT)
		Nominal water flow rate	4.5 GPM (17 lpm)
		Compressor EER	10.10 (Btu/Wh)
	Water heating	Storage	80 gal
		Tank Temperature	120°F
Air Handling Unit			
Airflow		Maximum	700 scfm
		Minimum	200 scfm

The hybrid GAHP-EAC system relies upon the logic built into the system that responds to a 2-stage 24VAC thermostat and aquastat demand calls. This logic monitors the outdoor ambient temperature (OAT) and indirect storage water heater temperature, amongst other variables, to appropriately respond to space conditioning and water heating demands.

The hybrid GAHP-EAC system provides space and water heating with the GAHP. A configurable outdoor reset curve for space heating is based on the OAT and supply water temperature to modulate the GAHP between the maximum and minimum heating capacity as a function of OAT. During a heating call, hot glycol/water Heat Transfer Fluid (HTF) from the GAHP is delivered to the AHU at a constant water flow rate, with the AHU increasing the airflow as a function of the time of actuation at thermostat stages. The GAHP operation is centered around maximizing run-on-time and efficiency. In DHW mode, after receiving a call from the aquastat, the GAHP will turn on and supply water at 120-125°F until the aquastat is satisfied. The control will prioritize water heating if both space and water heating are being called.

The hybrid GAHP-EAC can simultaneously provide space cooling from the EAC and water heating from the GAHP. Since the EAC is a two-stage compressor air conditioner, it provides space cooling based on 1st and 2nd cooling stage calls from the thermostat. The second stage cooling call will be triggered if the thermostat is 3.0°F above the set point. The AHU will start at medium speed and increase the airflow as a function of time.

Figure 2: The “Hybrid” GAHP-EAC unit. (Left) The outdoor unit in the environmental chamber; (Middle) The AHU unit system; (Right) The 80-gallon storage water heater.

3. Experimental Procedures

The hybrid GAHP-EAC system's performance for residential space conditioning and water heating applications was evaluated in a laboratory environment. The seasonal performance and capacities for space heating and cooling were characterized using standardized steady-state testing based on the North American rating method (ANSI Z21.40.4) and (AHRI 210/240), respectively. In addition to standardized testing, extended 24-hour Simulated Use Tests (SUT) on GTI Energy's Virtual Test Home (VTH) were conducted for real-world operation.

Both standardized steady-state test methods comprised a series of steady-state rating points under static conditions at full and partial loadings, including considering defrost modes and cycling degradation. Data from test rating points were used to calculate the seasonal performance metrics for the GAHP and EAC. The test conditions for the steady-steady testing can be referenced in the following standard's manual for ANSI Z21.40.4 [9] and AHRI 210/240 [10].

The hybrid GAHP-EAC system for space conditioning and water heating performance was evaluated in a simulated environment over 24 hours. In SUT testing, one space heating and cooling profile was implemented to present the heating and cooling season. One daily water heating profile was superimposed onto the space heating or cooling profile to simulate DHW usage for these two operating seasons. The SUTs space conditioning and water heating load profiles were based on yearly EnergyPlus modeling of a 3,016 sq-ft Vintage IECC 2012 home in Chicago, IL, as illustrated in Figure 3.

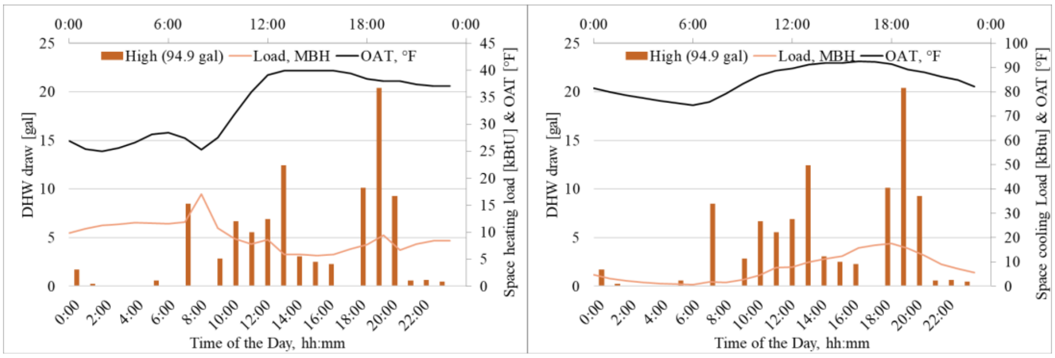

Figure 3: The load profiles for space conditioning and water heating. (Left) Heating season, (Right) Cooling season.

The SUTs implemented thermostat DHW simulators to evaluate the hybrid GAHP-EAC system performance and logic features to as-installed operating conditions. The thermostat simulator for simulated-use space conditioning evaluation is an algorithm based on a whole-home lumped heat capacity approach to simulate space conditioning part-load operation with HVAC equipment. This approach has been well documented in literatures [11,12].

A purpose-built test stand with instrumentation and the environmental chamber was used to evaluate the hybrid GAHP-EAC system, as illustrated in Figure 4. A National Instrument was used to record all data continuously throughout the data collection period with a maximum data sample interval of 5 seconds. The instrumentation list with make model and accuracy are summarized in Table 2.

Table 2: Instrumentation list and accuracy.

Measurement/ Component	Instrument Used	Instrument Accuracy
Natural Gas Pressure	Dwyer ISDP-008	±0.5% of reading
Natural Gas/Flue Temperatures	Omega T-type thermocouples	±1.1°C of reading
Natural Gas Flow	AM-250 and 500 pulses/cF	500 pulses/cubic foot
Flue Gas Composition	Bacharach PCA3 Analyzer	O_2: ±0.3% CO: Greater of ±5% of reading or ±10ppm NO/NO_2: Greater of ±5% of reading or ±5ppm
Air Temperature	Omega T-type thermocouples	± 0.75%
Air Relative Humidity (AHU)	Dwyer RHP-2D11	±2.0%RH (10– 90 % R.H.)
Air Relative Humidity (Outdoor)	Vaisala HMT120-130	±1.5%RH (0-90 % RH) @>0°C ±3.0%RH (0-90 % RH) @<0°C
Air Flow Rate	FE-1500-1-A-0-16x08-R-0-FX-1	± 2% of reading ± 0.12% of Full-scale
Air Flow Differential Pressure	Setra 2641-R25WD-11-C	± 0.25% of full scale (0" to 0.25" wc)
Pressure Rise	Setra 2641-2R5WD-11-C	± 0.25% of full scale (0" to 2.5" wc)
Ambient Pressure	Traceable Excursion-Trac Barometer	± 406 Pa
Glycol/Water Temperature	Omega RTDs PR-10L-3-100-1/8-9	Class B
Glycol/Water Flowrate	Dwyer MFS2-3	± 1% of reading
AHU Power (120Vac)	CCS WattNode Pulse RWNB-3Y-208-P and ACTL-0750-020	± 1% (5 to 100% rated current)
Outdoor Unit Power (240Vac)	CCS WattNode Pulse RWNB-3Y-208-P and ACTL-0750-020	± 1% (5 to 100% rated current)

The electrical input at the System Boundary is the total power measurement of the indoor and outdoor units as:

$$E_{Sys} = E_{AHU} + E_{OU} \tag{1}$$

Where:

E_{Sys} = accumulated electrical power at the System Boundary (kWh)
E_{AHU} = accumulated electrical power at the AHU (kWh)
E_{OU} = accumulated electrical power at the outdoor unit (kWh)

The following basic equation is used to calculate the gas consumption by the system:

$$Q_f = \sum HHV \cdot \dot{V}_f \cdot \Delta t \cdot \rho_{fs}/\rho_{fa} \tag{2}$$

Where:
Q_f = accumulated HHV fuel energy input (Btu)
HHV = fuel higher heating value (Btu/ft^3)
$\dot{V}_f$ = flow rate of fuel (lbm/h)
Δt = testing period (h)
ρ_{fs} = fuel standard density (lbm/ft^3)
ρ_{fa} = fuel actual density (lbm/ft^3)

Thermal output at AHU is determined by measuring the airflow rate, supply and return temperature, and relative humidity. The enthalpy difference was computed using the measured temperatures and relative humidity in Engineering Equation Solver. The heating and cooling output was calculated with the following equation for heating:

$$Q_{SH} = \dot{V_a} \cdot \rho_{as} \cdot -\Delta h \cdot t \cdot \sqrt{\rho_{as}/\rho_{aa}} \tag{3}$$

Where:
Q = accumulated heating/cooling output of the AHU, (Btu)
$\dot{V_a}$ = AHU airflow, (acfm)
Δh = enthalpy difference between leaving and entering air temperatures at the AHU, (ºF)
ρ_{as} = dry-air density based on the fluid temperature at the flow meter (lbm/cF)
ρ_{ss} = dry-air density at standard at 14.69psia and 70°F, (lbm/cF)
t = data collection time step, (min)

The DHW output was measured and calculated with the direct measurement of the water flow rate, city water, and DHW supply water temperatures at the indirect storage tank as:

$$Q_{DHW} = \dot{V_w} \cdot \rho_w \cdot c_{p_w} \cdot \Delta T_{DHW} \cdot t \tag{4}$$

Where:
Q_{DHW} = accumulated DHW output, (Btu)
$\dot{V_w}$ = city water flow rate, (gpm)
ΔT_{DHW} = temperature difference between city water and DHW temperatures at the indirect storage water heater, (ºF)
ρ_w = water density based on the fluid temperature at the flow meter (lbm/gal)
c_{p_w} = specific heat based on average entering and leaving water temperatures across the indirect storage water heater (BTU/lbm -°F)
t = data collection time step, (min)

The coefficient of performance (COP) at the GAHP is calculated by the accumulated heat produced by the GAHP over the gas consumption as:

$$COP_{gas} = \frac{\sum_{t_o}^{t}\left(\dot{Q_{SH}} \cdot \Delta t\right) + \sum_{t_o}^{t}\left(\dot{Q_{DHW}} \cdot \Delta t\right) +}{\sum_{t_o}^{t}\left(\dot{Q_g} \cdot \Delta t\right)} \tag{5}$$

Where:
COP_{gas} = coefficient of performance at the GAHP, gas only
Δt = data collection time-step
t_o = start of steady-state measurement
t = data collection time step, (min)

The COP for the system is calculated by the accumulated heat produced by the GAHP over the gas and electric consumption as:

$$COP = \frac{\sum_{t_o}^{t}\left(\dot{Q_{SH}} \cdot \Delta t\right) + \sum_{t_o}^{t}\left(\dot{Q_{DHW}} \cdot \Delta t\right) +}{\sum_{t_o}^{t}\left(\dot{Q_g} \cdot \Delta t\right) + E_{Sys}} \tag{6}$$

Where:
COP = coefficient of performance at the GAHP

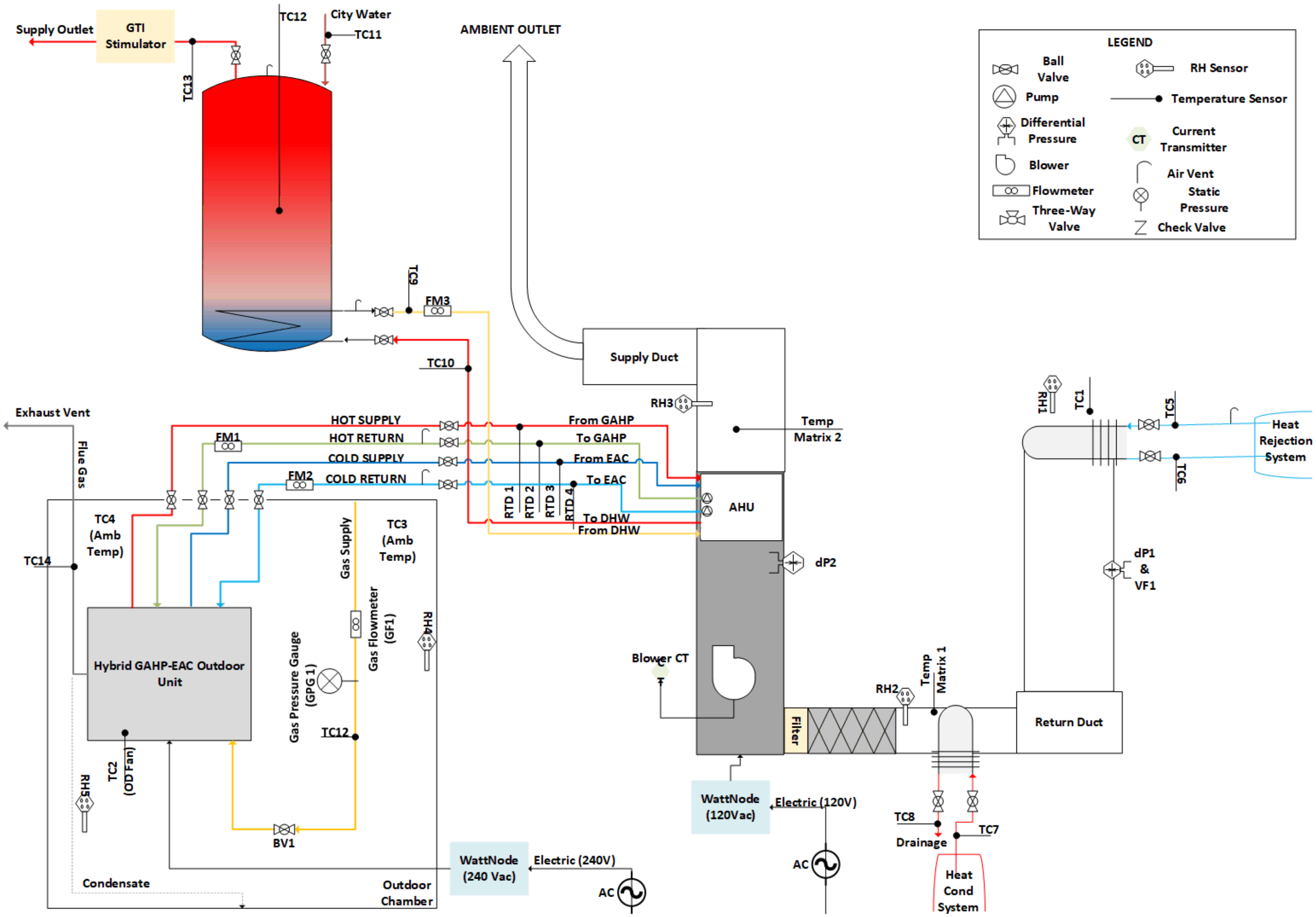

Figure 4: The purpose-built test stand and instrumentation.

4. Results and Discussion

The results of the steady-state heating and cooling test are summarized in Figure 5 and Figure 6, respectively. Using the data from Figure 5, the seasonal AFUE for US average climate bin conditions per ANSI Z21.40.4 was 121.6% for the US average climate bins based on a cycling degradation coefficient (Cd) of 0.25 as per the standard of ANSI Z21.40.4 based on air-water evaluation at a water return temperature of 95°F.

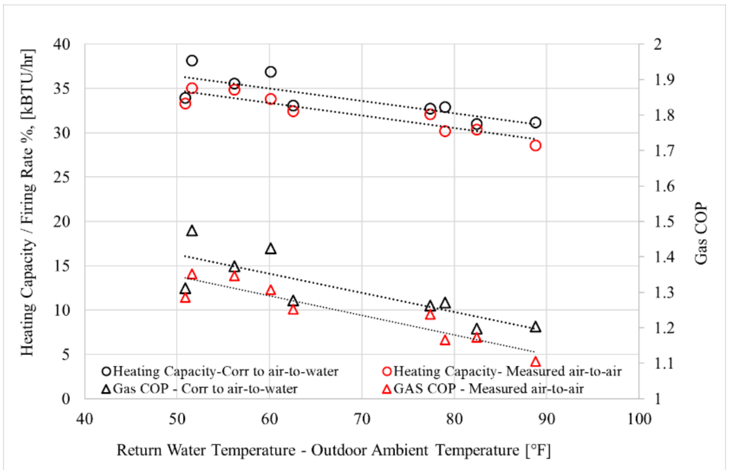

Figure 5: The ANSIZ21.40.4 steady-state testing results normalized at different modulation rates for air-to-water and air-to-air testing.

At lower and intermediate firing rates, the air-air GAHP was maintained close to 95°F return hydronic temperature (RHT), similar to an air-water GAHP. Whereas, at level 2 firing rate, the performance and capacity of the air-air GAHP are 9 to 12% lower than an air-water GAHP. At a higher firing rate, the air-air GAHP RHT was 13 °F higher than the air-water GAHP.

Using the data from Figure 6, the Seasonal Energy Efficiency Rating (SEER) was 10.1 (EAC only) and 8.24 (EAC + AHU) with a cycling degradation coefficient (Cd) of 0.59. As per standard Cd of 0.2 as per AHRI 210/240, the SEER was 12.4 (EAC only) and 10.1 (EAC +AHU). The EAC's cycling degradation coefficient and electric consumption were considered high in this prototype due to (1) an additional refrigeration/water heat exchanger and (2) an additional pump needed for the water loop.

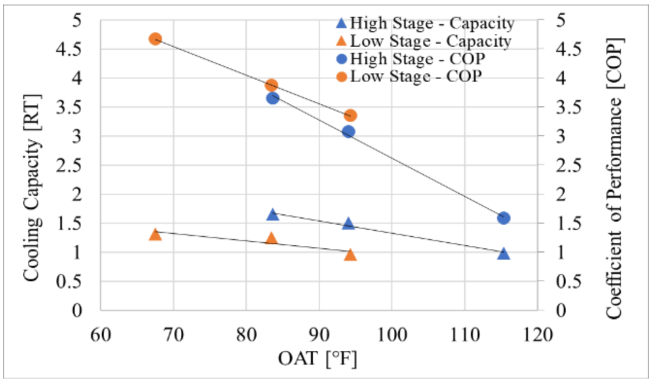

Figure 6: The AHRI 210/240 steady-state cooling test results.

Figure 7 summarizes and compares the supply air temperature of hybrid GAHP and EAC to other manufacturers' state-of-the-art technology. The GAHP supplied air at a higher temperature than ccASHPs in the market. At level 2 firing rate, the GAHP supplied air above 100°F. The EAC with hydronic loop provided supply air at ±2°F of manufacturer A's 16 SEER system.

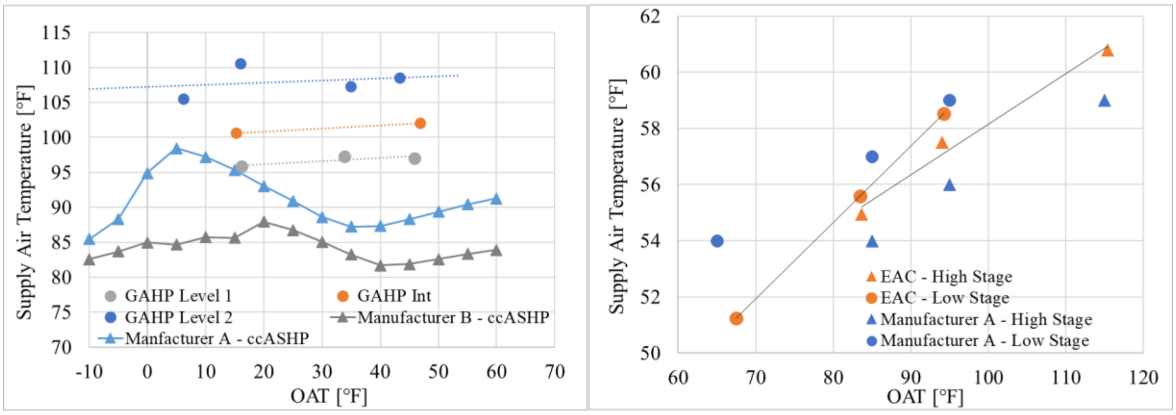

Figure 7: The supply temperature of the prototype in comparison to state-of-art system. (Left) Heating supply air temperature, (Right) Cooling supply air temperature.

The GHAP's flue gas composition and emissions were measured during steady-state tests. Table 3 summarizes the average measurements of flue gas composition on a dry basis. Compared to the 40 kBtu/hr gas furnaces, the hybrid's GAHP-EAC carbon monoxide (CO) in free air is 3 - 4 times lower. The NOx emission of 7.8 ng/J is roughly 45% lower than the requirement per Rule 1146 and SCAQMD requirements for residential and commercial space heating of 14 ng/J. Therefore, the hybrid GAHP-EAC can be classified as a low-NOx system.

Table 3: The steady-state emission measurement of the GAHP at different modulation levels

Modulation Level	O_2 %	CO_2 %	CO Air Free (ppm)	NO Air Free (ppm)	NO_2 Air Free (ppm)	NO_x (ng/J)
Level 1	4.6±0.3	9.2±0.3	11.5 ±12.7	10±6.3	6±6.3	7.8
Intermediate	5.2±0.3	8.9±0.3	32±12.8	8±6.4	7±6.4	7.3
Level 2	5.3±0.3	8.8±0.3	101±12.6	9.1±6.2	7±6.2	7.8

Figure 8 summarizes the hourly heating load, COP gas, OAT, and the indoor temperature during the simulated use testing over 24 hours. As outlined in Figure 8 (Left), the hybrid GAHP – EAC mainly operated in the intermediate firing rate region at a heating rate of 15 to 20 kBtu/hr. During the cyclic operation, the GAHP took about 25-30 minutes to reach steady-state operation, of which 6-10 minutes of initial load were used to heat the system's mass. Therefore, during the 24-hour duration, the GAHP only operated on the first stage of the heating call. The heating call's second stage was programmed to be triggered only when the indoor temperature is 3°F above the thermostat set point. The GAHP took an average of 60 minutes to 145 minutes to raise the home's temperature by 2°F from 69°F to 71°F, as shown in Figure 8 (Right).

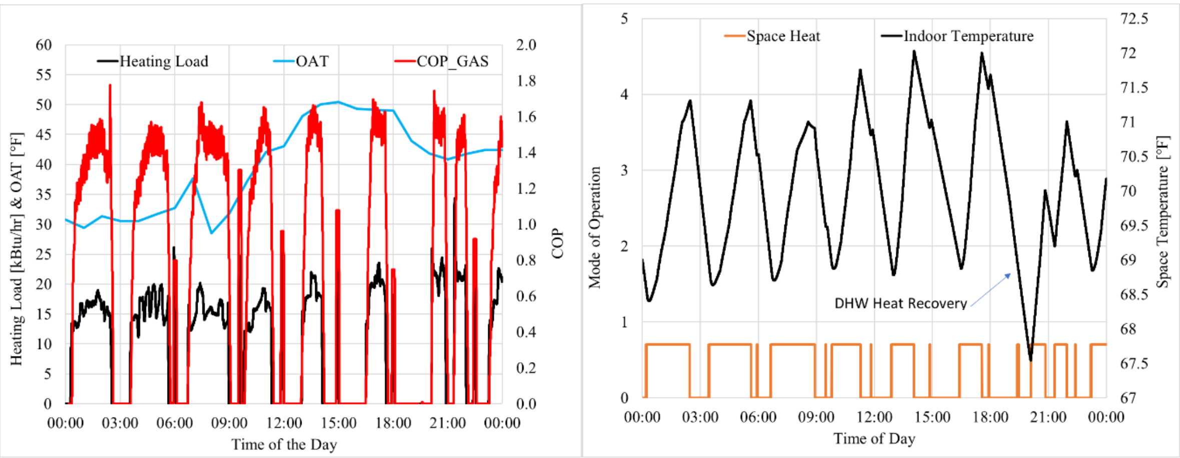

Figure 8: The 24-hour simulated use test results. (Left) The space heating load, COP gas, and OAT. (Right) The indoor temperature during space heating.

Table 4 summarizes the average space, water, and combi performance of GAHP when operated at an average OAT of 39.3 °F over 24 hours. The electric load made up 9% of the total energy used in a combi operation, and heat loss from the tank was 0.57 kBtu/hr (0.86 °F/hr), with a 24-hour standby electrical usage of 0.623 kWh. The combined system COP was 1.08. The significant electrical load due to the additional pumps and blower affects the system COP, as these are roughly 50% of the total electric load.

Table 4: The daily average of the SUT data for space heating and water heating

	Space Heating	**Water Heating**	**Combined**
Gas [kBtu]	177	44.84	223.1
Heating Supplied - @GAHP [kBtu]	231	53.19	284.19
Heating Supplied – @AHU [kBtu]	215	50.02	265.02
GAHP Electric Load [kWh]	2.24	0.586	2.98
AHU/Pump Electric Load [kWh]	2.79	0.371	3.63
COP_{gas}	1.21	1.11	1.19
COP_{total}	1.10	1.02	1.08
Standby-by Electric usage [kWh]			0.623
Water Drawn [Gal]	-	91.42 [37.2 kBtu]	-
Heat Loss [kBtu/hr]	-	0.57 [0.86°F/hr]	-
Tank Initial Temp [°F]	-	120.1	-
Tank Final Temp [°F]	-	118.7	-
Average City Water Temp [°F]	-	70.43	-
Average OAT [°F]	39.3		

An approximately 13% performance degradation between the GAHP performance rating and an optimized GAHP system can be found. This degradation percentage is anticipated with other technologies such as furnaces and ASHPs. Figure 9 shows the resulting performance of a 96% AFUE furnace at the same testing conditions implemented for the SMTI 40k hybrid GAHP system at 36 °F ambient temperature day. This testing condition was used for comparison since it represents the majority of load hours of heating equipment in the US national average climate. The 96% AFUE furnace sized per Air Conditioning Contractors of America (ACCA) Manual S scored 89% daily gas efficiency. The difference between AFUE and installed performance was about 7%, similar to the difference observed between air-to-air configuration ANSI Z21.40.4 testing and SUT evaluation in the 40k hybrid GAHP system.

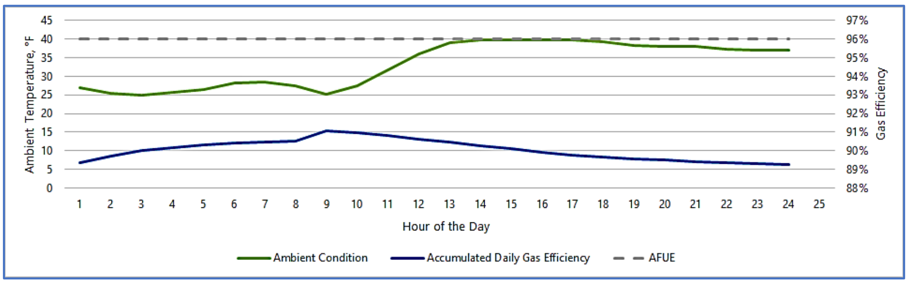

Figure 9: Daily efficiency of a 96 AFUE furnace at VTH 36 °F ambient test used for the SMTI 40k hybrid GAHP system.

Similarly, Table 5 summarizes the average space cooling with EAC and water heating with GAHP performance over 24 hours. The average COP of EAC for space cooling was 2.56 at average OAT of 92.2 ˚F. Similarly, the average system COP of water heating with GAHP was 1.3, and the heat loss from the tank was similar to the heating test at 0.62 kBtu/hr. Compared to the heating test, the 24-hour standby electrical load was 0.36 kWh; the lower standby electrical load in the cooling test was due to the EAC's extended long-time over the 24 hours.

Table 5: Summary of the SUT testing for space cooling with EAC and DHW with GAHP

	Space Cooling	**Water Heating**
Gas [kBtu]	-	32
Cooling Supplied - @EAC [kBtu]	205.1	-
Cooling Supplied – @AHU [kBtu]	191.3	-
Heating Supplied - @ DHW	-	48.1
GAHP/EAC Electric Load [kWh]	16.3	0.7
AHU/Pump Electric Load [kWh]	5.5	0.8
COP_{gas}		1.5
COP_{total}	2.56	1.3
Standby-by Electric usage [kWh]	0.36	
Water Drawn [Gal]	-	86.3
Heat Loss [kBtu/hr]	-	0.62 (0.94˚F/hr)
Tank Initial Temp [°F]	-	118.7
Tank Final Temp [°F]	-	117.9
Average City Water Temp [°F]	-	70.8
Average OAT [°F]	92.2	

5. Conclusion and Future Work

A series of laboratory evaluations were conducted to map the performance of the Hybrid GAHP-EAC system for residential HVAC and water heating applications. The seasonal heating and cooling performance of the Hybrid GAHP-EAC system were: (1) AFUE: 121.6%; (2) SEER: 8. In the SUT testing for space and water heating, the system performance degraded by 6- 11% in comparison to the steady-state testing. The average system COP, including the gas and the electrical load, was 1.10 (space heating), 1.02 (water heating), and 1.08 (combined) at an average OAT of 39.3 ˚F. The electric consumption was 9% of the energy consumption.

In the SUT testing for space cooling and water heating, the system performance degraded by less than 7% due to the long runtime of the EAC. The EAC operated at extended periods in stage 1, as the trigger from stage 1 to stage 2 was ±3 ˚F above the thermostat set point. The average COP was 2.56 for space cooling and 1.3 for water heating at an OAT of 92.2 ˚F.

Future work for space heating are:

- Improvement in controls for the time taken for space heating to maintain comfort. The 3 ˚F trigger for the next stage seems high to keep the required comfort in the home.

- The electric load should be considered in the design phase. The electric load makes up 9% of the energy usage. As observed in the SUT testing, the combined COP of the system was 1.08 at 39.3 ˚F.

Future work for space cooling are:

- Like space heating, a 3 ˚F trigger for the next stage seems high to maintain the required comfort in the home and reduce the EAC's performance. Therefore, the motivation for the next step should be a combination of time-based and OAT reset.

Acknowledgments

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