AC-02-10-3

Prototype Commercial Hot Water Gas Heat Pump (CHWGHP)—Design and Performance

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ABSTRACT

About 1.06 quads of energy are consumed in the U.S. alone in the production of commercial hot water. The primary energy efficiency is about 90% for gas heaters and 32% for electric. In both cases, high-quality, highly exergetic energy is directly converted one-for-one into very low quality energy, yielding very low second law efficiencies.

A gas-fired hot water absorption heat pump has been developed, incorporating generator-absorber heat exchange (GAX) to deliver hot water at temperatures above 60°C (140°F). With a gas COP of 1.54, one-third of the gas used for commercial water heating can be conserved. The heat pump also provides "free" cooling with evaporator temperatures as low as –10°C. The projected payback to the typical commercial customer is about two years.

This paper summarizes the prototype heat pump design and testing, verifying component performance, heat pump capacity, temperature lift, water delivery temperature, and COP.

INTRODUCTION

A significant amount of energy is consumed in the production of commercial hot water. About 1.06 quads of energy are consumed in the U. S. alone, approximately evenly divided between gas and electric. Gas water heaters have reasonable first law efficiency in the 80% to 95% range. However, their second law efficiency is abysmally low—fuel combustion at up to 1400°C transfers heat directly to water below 65°C. Electrically heated hot water is even worse, by a factor of three. In essence, extremely high quality, highly exergetic energy is directly converted one-for-one into very low-quality energy, yielding very low second law efficiencies.

The water heater efficiency can be improved by employing heat pump technology, whereby the high-quality energy performs more useful work in the process of converting it to low-quality heat. This is a capsule description of what a heat pump does. There are several electric-powered commercial hot water heater heat pumps already in the market. However, electrical input introduces a three-to-one energy penalty due to the generation and transmission efficiencies. The electric heat pump must have a heating COP of at least three just to reach prime energy breakeven with gas water heaters. Due to the high-temperature lift required for commercial hot water, their electric COP is in the range of 3 to 3.5. This translates to a primary-energy heating COP only in the range of 1.0 to 1.2—slightly better than gas heaters. Typical rating conditions are 20°C (72°F) wet bulb at the cold end and 60°C (140°F) hot water out. This performance is sufficient to reach energy breakeven with direct gas firing, but may not be cost-effective, since electric energy on average costs two to three times as much as gas for an equivalent amount of hot water production. There are no real energy savings to justify the higher capital cost. In addition, when the cold-end temperature is lowered to provide useful cooling, the COP and capacity decline. All told, it is quite understandable that electric heat pumping of commercial hot water has remained a niche product.

In order to conserve a significant fraction of the 1.06 quads of energy currently consumed producing commercial hot water, it is clear what must be done. A hot water heat pump must be developed that is heat-powered, not electric-powered. It can be either direct gas-fired or waste heat-powered from a prime mover. Ideally, it would also provide high enough temperature lifts that the cold end co-product is always useful, either for air conditioning or refrigeration.

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The above attributes are precisely what is achieved by the gas-fired hot water heat pump described below. The absorption heat pump incorporates generator-absorber heat exchange (GAX) and provides high-temperature lifts at high efficiency. With a gas heating COP of 1.54, one-third of the gas used for commercial water heating is conserved. Additionally, the cold co-product conserves a comparable amount of energy normally used for space cooling.

A prototype has been developed and tested to verify component performance, heat pump capacity, temperature lift, water delivery temperature, and COP. This paper summarizes the prototype design and test results.

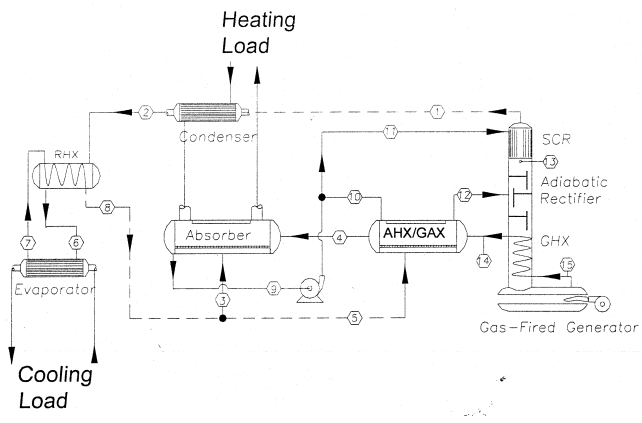

Figure 1 *Commercial hot water gas heat pump flowsheet.*

MULTIPURPOSE COMMERCIAL HOT WATER GAS HEAT PUMP (CHWGHP)

Flowsheet and Design Heat and Mass Balances

The CHWGHP is an ammonia-water absorption heat pump based on the GAX cycle. Figure 1 is the flowsheet of the CHWGHP, which identifies different components and thermodynamic statepoints. The components of the cycle are identified as follows: Condenser (COND), Evaporator (EVAP), Refrigerant Heat eXchanger (RHX), Generator (GEN), Adiabatic Rectifier (REC), Solution Cooled Rectifier (SCR), Water Cooled Absorber (WCA), Absorber Heat eXchanger (AHX), Generator Absorber heat eXchanger (GAX), and Generator Heat eXchanger (GHX).

A standard simulation package with built-in ammonia-water property routines (Klein and Alvarado 1996) was used in the analysis. The cycle modeling procedure and analytical techniques are similar to those presented by Herold et al. (1995). The results presented in Table 1 include realistic driving forces for heat and mass transfer and reflect the performance of actual components. About 95% of design performance has been achieved in a demonstration prototype of an advanced GAX cycle using those components (Anand and Erickson 2000).

Figure 2 is the corresponding thermodynamic representation of the cycle on a Gibbs diagram, showing temperatures, pressures, and concentrations at the statepoints. The statepoint conditions and thermodynamic properties of the CHWGHP

TABLE 1
Design Thermodynamic Properties

State Point	T [°C]	Conc. [0-1]	Flow Rate [kg/s]	Quality [0-1]	Enthalpy [kJ/kg]	Sp. Vol. [m^3/kg]
1	91.70	0.9826	0.0258	1.000	1447.2	0.0820
2	48.89	0.9826	0.0258	0.000	223.2	0.0018
3	36.89	0.9826	0.0147	0.970	1314.3	0.2700
4	90.86	0.2415	0.0467	0.000	230.3	0.0012
5	36.89	0.9826	0.0111	0.970	1314.3	0.2700
6	5.56	0.9826	0.0258	0.061	89.5	0.0016
7	11.87	0.9826	0.0258	0.919	1179.7	0.2333
8	36.89	0.9826	0.0258	0.970	1313.4	0.2696
9	54.44	0.4190	0.0614	0.000	9.8	0.0012
10	54.46	0.4190	0.0346	0.000	10.4	0.0012
11	54.46	0.4190	0.0267	0.000	10.4	0.0012
12	140.69	0.4190	0.0346	0.252	766.8	0.0249
13	107.65	0.4190	0.0267	0.000	255.4	0.0013
14	150.33	0.0100	0.0356	0.000	629.0	0.0011
15	208.34	0.0100	0.0356	0.000	886.0	0.0012

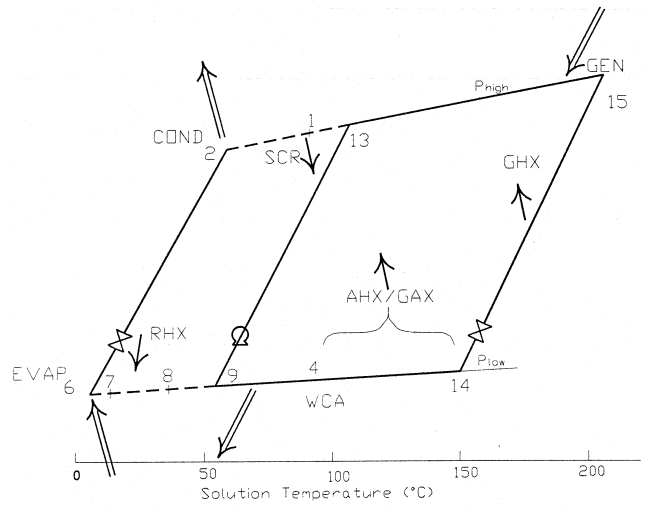

Figure 2 *Gibbs diagram for CHWGHP cycle.*

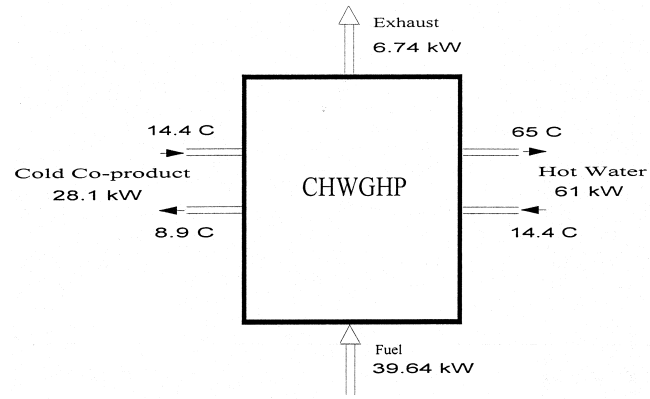

Figure 3 *Input/output representation of commercial hot water gas heat pump.*

designed for the application illustrated in Figure 2 are tabulated in Table 1.

In Figure 2, the two pressure levels of the cycle are marked P_{high} and P_{low}. The constant concentration lines are straight diagonal lines on this diagram. Vapor exiting the generator/rectifier is condensed at P_{high} and T_{cond}. The liquid from the condenser is precooled in the RHX and evaporated at P_{low} and T_{evap}. The strong solution (strong absorbing capacity) leaves the generator at T_{gen} and P_{high} and is subcooled in the GHX. The solution then flows sequentially through the GAX/AHX and WCA and absorbs the vapor from the evaporator. The resulting weak solution exits the WCA at Tabs and is pumped to the high pressure, completing the cycle.

Figure 3 illustrates the overall operational effect and external heat balance of the multipurpose integrated CHWGHP. It is supplied with 39.64 kW of fuel. This prime energy input is used by the CHWGHP to heat pump 28.1 kW from the low-temperature coolant to the high-temperature hot water. With a typical burner efficiency of 83%, 6.74 kW of input fuel energy is lost in the exhaust. A net amount of 61 kW (28.1+0.83*39.64) of heat is thus supplied to the hot water, heating 0.29 kg/s (4.58 gpm) of water from 14.44°C (58°F) to 65°C (149°F). The gas heating COP of the CHWGHP is 1.54 (61/39.64). In the process, 1.21 kg/s (19.2 gpm) of coolant is chilled from 14.44°C (58°F) to 8.89°C (48°F) and can be used for space conditioning. No additional energy is required to produce the cooling.

Key Development Issues

High Glide Absorber. The CHWGHP uses several conventional components that are already mass-produced: ammonia evaporator, ammonia condenser, and receivers. The specialized components are the WCA, AHX/GAX, GEN, SCR, and RECT. New configurations for these components have been developed (Anand and Erickson 2000; Anand and Erickson 1999; Erickson et al. 1996), which enhance performance, reduce size, and reduce manufacturing cost. Traditional absorbers are not designed for high-temperature glide but for low cost, usually resulting in quite low glide. However, the recent worldwide activity on ammonia water GAX cycles has generated great interest in high glide absorbers, since they are vital to the GAX concept. Achieving high glide in the absorber turned out to be much more difficult than anticipated. A promising concept based on diabatic tray design has been developed by the authors (Anand and Erickson 2000). This technology was adapted for the new high glide water-cooled absorber.

Load Matching Capability. The generic problem always faced by integrated appliances is load matching. The integrated appliance is designed to deliver products at optimum conditions. However, the individual loads that the appliance is designed to supply (e.g., hot water and cooling) tend to occur at different times and in different quantities than the outputs delivered by the appliance.

For the integrated commercial hot water heater heat pump, a relatively straightforward solution has been developed to solve this problem. First, regarding load quantity mismatch, the air-conditioning load of most commercial establishments requiring hot water is typically substantially larger than the hot water load. Hence, the appliance is sized for the hot water load, and the air conditioning is merely a bonus. A normal air-conditioning unit will be required, preferably downsized to account for the contribution from the hot water heat pump. A hot water storage tank is incorporated to accommodate temporal variations in hot water load.

Service Water Issues. Two important issues, external to the heat pump per se, are the need to preserve sanitation of the service water and the deposition of scale from the heated water. Both issues are resolved by providing a closed hydronic heat transfer loop between the CHWGHP and the service water. The hydronic loop contains a sanitary liquid (typically water), a circulating pump, and a water-to-water heat exchanger, which can readily be de-scaled on the service water

side. The hydronic liquid removes heat from the condenser and absorber of the CHWGHP and transfers it to the service water.

A similar loop is normally employed on the cold end—either a brine loop for refrigeration or a glycol loop for air conditioning. In some situations, a thermosyphon heat transfer can be used to avoid another circulating pump.

As shown in Figure 1, the service water to be heated exchanges heat with an intermediate hydronic heat transfer fluid that flows in series through the condenser and WCA. This flow sequence maximizes both the COP and the final hot water temperature, while avoiding undue system complexity. The COP and delivery temperature are qualitatively and quantitatively evaluated by performing an enthalpy-temperature (Q-T) analysis for the condenser and WCA. This analysis calculates the enthalpy versus temperature of each fluid stream and then compares the several streams to ensure that the enthalpy transfer (i.e., heat transfer) at all points in the heat exchanger has a positive temperature driving force. Figure 4 presents the Q-T of the condenser and WCA for the design conditions presented in Table 1. In Figure 4, the dotted line represents the temperature profile of the intermediate heat transfer fluid. The transfer fluid flows in series through the condenser and WCA and picks up 61 kW of heat. This heat is then transferred to the service hot water, resulting in the water temperature increasing from 14.44°C to 65°C, as shown by the hot water profile. The condensation and absorption profiles represent the corresponding heat release curves in the condenser and WCA, which are arranged in counter-current heat transfer with the intermediate fluid. The condensation profile has a dramatic drop in condensation temperature at the hot end characteristic of the condensation process encountered in ammonia-water systems. The minimum approach temperature (pinch) is observed at the cold end of the absorber and is 13.8°C (24.9°F).

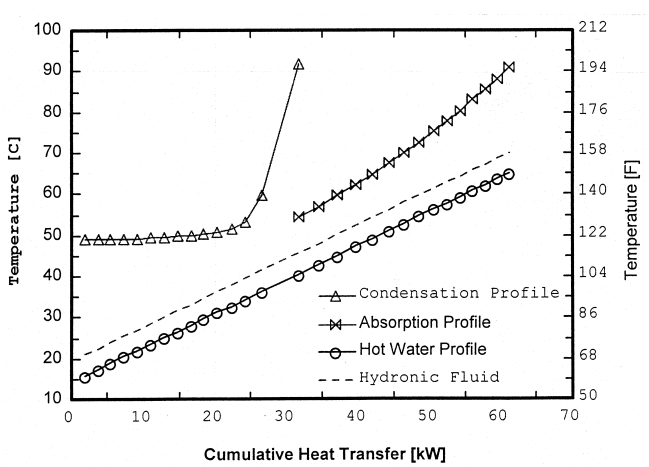

Figure 4 *Design Q-T analysis of condenser and WCA.*

PROTOTYPE PERFORMANCE

The objectives of the prototype development and testing were twofold.

1. Design and test high glide absorber.
2. Verify capacity and efficiency.

Figure 5 shows the breadboard prototype. A PC-based data acquisition system (DAS) was used to monitor system performance. The computer was also used to control liquid levels and flow rates. The measured steady-state experimental data were analyzed to derive solution concentrations, component heat duties, cycle COP, and transfer coefficients. The analysis begins with a heat, mass, and concentration balance on three separate interrelated control volumes: the generator/rectifier, the low pressure GAXs and absorber, and the refrigerant components. This procedure provides several independent calculations of internal parameters, ensuring thermodynamic consistency of the cycle solution. Then two additional independent checks are performed: the internally calculated generator duty is compared to the externally measured gas heat input, and, similarly, the internally calculated cooling duties are compared to the externally measured heat gained by the service water.

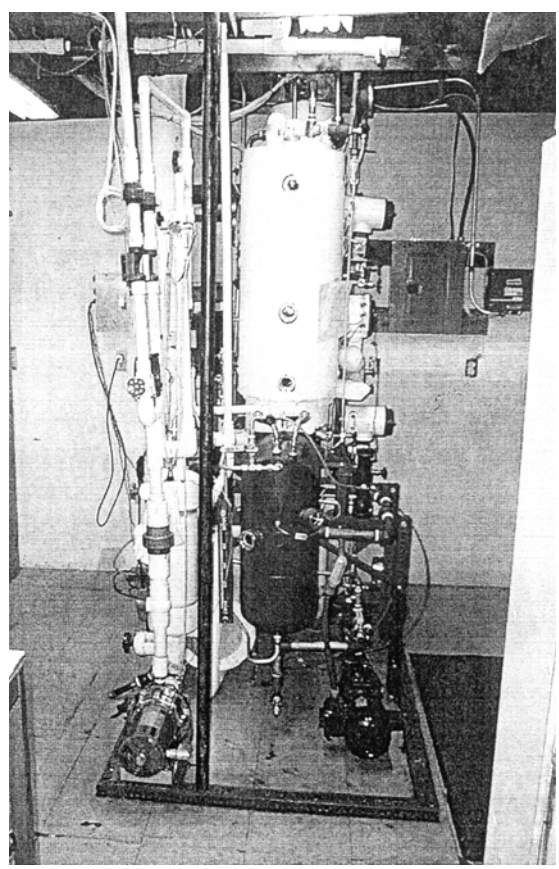

Figure 5 *CHWGHP breadboard prototype.*

TABLE 2
CHWGHP Breadboard Prototype Performance Data

RUN	Design	0530	0316	0314	0312	1218
Cooling capacity (kW)	28.1	27.34	26.3	28.2	24.8	27.8
Gas cooling COP	0.71	0.73	0.66	0.69	0.70	0.75
Gas heating COP	1.54	1.58	1.51	1.53	1.54	1.60
Effective lift (°C)	46.11	42.2	40.1	37.1	38.7	37.9
Condenser coolant flow (10^{-3} m^3/s)	0.289	0.2662	0.3578	0.4382	0.3938	-
Condenser coolant temperature in (°C)	19.44	17.5	20.8	20.5	24.9	-
Condenser coolant temperature out (°C)	45.61	43.0	39.12	36.22	40.03	-
Absorber coolant flow (10^{-3} m^3/s)	0.289	0.2662	0.1325	0.183	0.1514	0.3987
Absorber coolant temperature in (°C)	45.61	43.0	20.8	20.5	24.9	19.7
Absorber coolant temperature out (°C)	70	69.1	78.58	64.43	70.88	38.42
Condenser outlet temperature (°C)	48.9	45.98	44.1	42.9	44.0	45.1
Absorber outlet temperature (°C)	54.44	50.80	47.6	42.9	46.6	37.9
Evaporator inlet temperature (°C)	5.56	6.21	6.2	6.3	6.8	3.6
Condenser/high side pressure (bar)	19.42	18.18	17.8	17.28	17.49	17.83
Absorber/low side pressure (bar)	5.17	4.959	4.876	4.883	4.924	4.89

Results from five steady-state conditions are presented in Table 2. The design parameters are also included in the table for reference. In all cases, the system was operated at near design capacity. The coolant inlet temperature was maintained close to the design temperature of 19.44°C. The coolant inlet temperature has a strong effect on system performance. A low coolant inlet temperature causes lower condenser pressure and a colder absorber, both of which increase system efficiency. A higher inlet temperature conversely increases condenser pressure and absorber temperature and lowers the system efficiency.

Run 1218 was conducted with the refrigerant components bypassed, and the cooling capacity was inferred from the refrigerant vapor flow rate. The absorber coolant flow was set higher than design so as to check flow capacity through the absorber.

The other four runs were conducted with the refrigerant components online. For runs 0316, 0314, and 0312, the hydronic coolant inlet temperature to both the condenser and the absorber is the same because the hydronic flows through the condenser and absorber were in parallel rather than in series. The coolant flow through the condenser was adjusted to give the desired high side pressure. The absorber coolant flow was adjusted to achieve the desired outlet hydronic temperature. As the test data indicate, a substantial temperature glide was achieved on the hydronic stream, validating the high-glide absorber design. For all these runs, it was observed that the SCR component was over-rectifying the vapor, resulting in lower condenser temperature/pressure and lower efficiency.

After validating basic system operation and component performance, the hydronic piping was reconfigured for run 0530 to achieve serial hydronic flow through the condenser and absorber. The SCR over-rectification problem was addressed by reducing solution flow through the SCR. As the test data indicate, near design capacity, COP, and hydronic temperatures were achieved. The system gas cooling COP ranged from 0.66 to 0.75, compared to the design COP of 0.71. The corresponding gas heating COP ranged from 1.50 to 1.60, compared to the design COP of 1.54.

PAYBACK AND MARKET POTENTIAL

Test results reported above validate overall prototype design and individual component specifications. The design information was used to estimate the heat pump manufacturing cost, payback period, and market potential.

The cost of manufacturing the gas-fired hot water heat pump is estimated to be less than $200 per ton for the sealed absorption system (not including burner and pump). Adding all remaining components, overhead, and profit yields a manufacturer-selling price of $500 per ton. Thus, the cost from the distributor would be about $700/ton, and the installed cost to the final user about $750 ($6000 for the 8-ton CHWGHP). This cost estimate is based on a manufacturing volume of 25,000 units per year.

Table 3 presents a comparison of the installed cost of a conventional gas-fired commercial water heater, which delivers 61kW heating, to the installed cost of the CHWGHP system. The latter system is shown to carry a cost premium of $8,000, due to the extra storage and higher installation cost.

TABLE 3
Installed Cost Comparison,
61 kW (208.000 Btu/h Heating Capacity

Conventional System	
Gas-Fired Water Heater	$3,000
Installation	1,000
Total	$4,000
CHWGHP System	
CHWGHP	$6,000
Storage Tanks (2-120 gallons)	2,500
Auxiliary Water Heater (10 kW)	1,000
Installation	2,500
Total	$12,000

The cost saving to the customer is determined by the annual usage and the utility rates. The following calculation assumes a 33% duty cycle (CHWGHP running one-third the time) and typical national average gas ($5.62/MMBtu) and electricity rates (76 mills/kWh). Note that current trends have shown remarkable increase in utility rates; higher energy costs will further improves the heat pump economics and payback.

- Hours of operation: $8670 \div 3 = 2920$
- Gas usage, conventional system: $\frac{208{,}000 \text{ Btu/h}}{0.93 \text{ efficiency}} \times 2920 \text{ h} = 653 \times 10^6$ Btu
- Gas cost, conventional system: $653 \times \$5.62 = \$3670/\text{yr}$
- Gas usage, CHWGHP: $\frac{208{,}000 \text{ Btu/h}}{1.54 \text{ efficiency}} \times 2920 \text{ h} = 394.4 \times 10^6$ Btu
- Gas cost, CHWGHP: $394.4 \times \$5.62 = \$2216/\text{yr}$
- Value of cooling co-product of CHWGHP: $28.1 \text{ kW} \times 2920 \text{ h} = 82{,}052$ thermal kW/yr
- $82{,}052 \text{ kW (th)} \times \frac{1.4 \text{ kW (e)}}{3.517 \text{ kWth kWh(e)}} \times \frac{\text{ton}}{} \times \frac{\$.076}{} = \$2482/\text{yr/ton}$
- Electrical usage, CHWGHP: $1.1 \text{ kW} \times 2920 \text{ h} = 3212$ kWh
- Electrical usage cost, CHWGHP: $3212 \times \$0.076 = \$244/\text{yr}$
- Annual savings, CHWGHP: $3760 - 2216 + 2484 - 244 = \$3692$
- Hence, payback is: $8000/3692 = 2.17$ yr

DISCUSSION AND SUMMARY

The operation of the CHWGHP prototype has demonstrated cycle feasibility and satisfactory performance of both the overall prototype and the individual components. The tests verified prototype operation at the design capacity, temperature lift, water delivery temperature, and COP. When the cooling and hot water heat pumping capabilities are both fully utilized, this appliance has what is arguably the highest prime energy efficiency of any equivalent appliance. Manufacturing cost estimates and payback analysis indicate substantial savings to the customer. Work is in progress to develop and refine the control system and the load interface. Field-testing of the packaged prototype is planned in the next phase of the project.

ACKNOWLEDGMENTS

The authors would like to thank the Office of Building Technologies, U.S. Department of Energy, in Washington, D.C., and National Energy Technology Laboratory, Pittsburgh, for supporting this work. Advanced Mechanical Technology, Inc., performed the design and construction of the gas-fired generator. ECC shop staff is acknowledged for fabrication of the CHWGHP.

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