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ABSTRACT

This article compares four heating and cooling systems. The systems are: a high efficiency furnace and electric air conditioner; a ground source heat pump; an absorption air conditioner and direct heating; and a thermally driven heat pump; the last two systems use solar thermal energy and backup non-renewable energy. A comprehensive program was developed that predicted the entire life cycle cost, energy usage, exergetic efficiency, and exergy destruction, of all four systems operating in the same home figuratively placed in the cities of Louisville, KY; Houston, TX; Minneapolis, MN; Sacramento, CA; and Phoenix, AZ. The results showed that the vertical ground source heat pump always paid back in the shortest time, between 4-15 years in all five cities compared to the furnace and air conditioner system. The economic pay back period was the shortest between 4-7 years in the cities of Louisville, Minneapolis, and Phoenix, which have larger heating and/or cooling requirements. The thermally driven heat pump, which largely used renewable energy, had equal or greater exergetic efficiency than the ground source heat pump in each city, while the furnace and air conditioner always had the lowest exergetic efficiency.

INTRODUCTION

Today people are becoming increasingly concerned about the diminishing fossil fuel resources, energy costs, pollution, and climate change. In residential buildings, most energy is used in heating, cooling, and hot water; therefore, by determining the most efficient heating, cooling, and hot water systems, energy consumption will be decreased in residential buildings. This article presents a program that determines the energy used and exergy destroyed in conventional heating, cooling and hot water systems and compares this to the energy used and exergy destroyed in new proposed systems that obtain a portion of energy needed from renewable energy from thermal solar collectors. This article also presents a life cycle cost analysis of the conventional and newer systems to determine if the newer systems are currently cost effective, and if not what factors such as utility costs or costs of thermal solar collectors need to change to make the newer systems cost effective.

The systems studied are fully described in the section titled "Case Studies" and the systems include condensing furnaces, ground source heat pumps, absorption air conditioning, and thermally driven heat pumps. Because the systems include the devices listed above, previous studies that involve these heating and cooling devices are presented. A condensing furnace is typical natural gas furnace except that it cools the flue gases down so low that some of the water vapor condenses, and uses the heat from this process to add to the heating of the home. Wright et al. (1984) give details of prototype and performance data of gas furnaces using plastic heat exchangers, which are less expensive and more corrosion resistant than steel heat exchangers, to condense a portion of the flue gases exhausted from the combustion process; the prototypes exhibited thermal efficiencies of 92%. Cohen et al. (1991) studied the effect of condensing furnaces that replaced traditional furnaces in three US cities and found that the replacements saved 31-41 GJ/yr and were cost-effective.

Zogou and Stamatelos (1998) demonstrated that ground-source heat pumps performed better than air-source heat pumps in all climates. Shonder et al. (2002) also showed that there were savings from retrofitting air-source heat pumps with ground-source heat pumps.

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Absorption chillers utilize a refrigerant-absorbent binary mixture that is environmentally benign to cool the home. Florides et al. (2002) used computer modeling of a domestic-size absorption air conditioner powered by solar energy to determine the energy costs associated with cooling by the absorption process plus heating by a boiler. Life cycle cost analysis showed that the unit was not economically feasible at that time. Priedeman and Christensen (1999) suggested that absorption air conditioners may offer lower energy costs than electric air conditioners when natural gas rates are favorable compared to electric rates. Sumathy et al. (2000) reported on a lithium bromide absorption chiller that was developed and tested in China and powered by low-grade thermal energy between 60°C (140°F) and 75°C (167°F). This low-grade energy allowed for available solar energy to be an optimal energy source to power the absorption unit. Unfortunately, the performance of device was low when operating from low-grade energy, and reported a cooling COP between 0.31 to 0.39.

Santoso (1989) described a Rankine cycle engine-driven heat pump system where a compressor and pump were powered by a Rankine cycle, instead of electricity. A computer program was developed that simulated the conditions at various locations and determined the mechanical efficiency and coefficient of performance of the system at different operating conditions. The energy source for the boiler in the Rankine cycle could be natural gas or low grade energy such as exhaust or thermal solar energy. The boiler operating temperature ranged from 36.7°C (98 °F) to 61.1°C (142°F). Using R-22 for the vapor compression cycle and R-113 for the Rankine cycle, the coefficients of performance for heating and cooling were found to be as good as or better than other common systems.

The objective of the study and paper is to examine four heating and cooling systems operating in the same home figuratively placed in five different climate regions in the United States. The program developed evaluated these four systems using data entered by the user and then determined the energy efficiency, exergy efficiency, and cost effectiveness of each design in each region. The systems studied will be presented in detail in Case Studies section and were (1) a condensing gas furnace with a high efficiency electric air conditioner and natural gas hot water heater; (2) a vertical ground source electric heat pump which provides some hot water and has electric backup furnace and hot water; (3) solar collectors that provide thermal energy for an absorption air conditioner and direct heating with natural gas backup; and (4) solar collectors that provide thermal energy to a thermally driven ground source heat pump with electric backup heat pump and hot water heater. These systems were chosen because the furnace and AC system is the most commonly used system, and the ground source heat pump is highly efficient and currently available. The other systems were chosen because they use renewable thermal solar energy and the current status of these systems is that all the equipment exists for the absorption system, and the thermally driven heat pump is currently under development. The study and the paper are original in that previous researchers have analyzed some of these systems but these four systems have never been studied simultaneously and two of the four systems studied are likely the most promising for utilizing renewable energy in a residence. In addition, the study determines the energy efficiency, exergy efficiency, and cost effectiveness of newer systems compared to traditional systems and determines if it is currently viable to invest in these systems and if not how conditions need to change to encourage investment in efficient systems that partly use renewable energy. Furthermore, the study examines how exergy efficiency and destruction correlates to energy efficiency and cost effectiveness in residential heating and cooling systems which may encourage the use of exergy analysis in determining optimized systems.

ANALYSIS

Heat Transfer

A commercial program was used to determine the heating and cooling required for the home in the five cities of Louisville, KY; Phoenix, AZ; Sacramento, CA; Minneapolis, MN; and Houston, TX. The commercial program accounted for the energy transferred by heat to or from the building due to outside temperature, sunlight, equipment (electrical appliances, such as stoves, dishwashers, computers, etc.), air infiltration, and people. The program also accounted for energy transfer due to latent energy such as moisture evaporation. Heat transfer was calculated by determining the thermal conduction resistances of the various building materials of the home and the thermal resistances associated with convection and summing up these thermal resistances in parallel or series as appropriate. Equations 1 and 2 show resistances in conduction and convection, respectively.

$$R_{cond} = \frac{L}{kA} \tag{1}$$

$$R_{conv} = \frac{1}{hA} \tag{2}$$

Summing up resistances in series is done by directly adding up each resistance; summing up resistances in parallel, such as a wood stud and insulation in a wall, is shown in Equation 3.

$$\frac{1}{R_{parallel}} = \frac{1}{R_{wood}} + \frac{1}{R_{insulation}} \tag{3}$$

The total thermal resistance of each of the walls was calculated in Equation 4 by accounting for inside and outside convection resistances and conduction resistances of sheetrock, insulation, wood, sheeting, and siding.

$$R_{wall} = R_{conv,in} + R_{sheetrock}$$

$$\frac{1}{\frac{1}{R_{wood}} + \frac{1}{R_{insulation}}} + R_{sheeting} + R_{siding} + R_{conv,out} \quad (4)$$

The thermal resistance of the floor, ceiling, and windows are shown in Equations 5 through 7. The outside convective resistance above the ceiling in the attic was different than outside the siding because it is a ventilated attic but not exposed to the outside air.

$$R_{floor} = R_{conv,in} + R_{floor} \quad (5)$$

$$R_{ceiling} = R_{conv,in} + R_{sheetrock} + R_{insulation} + R_{conv,out} \quad (6)$$

$$R_{window} = R_{conv,in} + R_{window} + R_{conv,out} \quad (7)$$

After the resistances were calculated the amount of heat transferred through each wall was calculated by Equation 8. This same form of equation was used to determine the heat transfer through the windows, ceiling, and floor. As seen, the heat transferred varies with respect to outside temperature, therefore the outside temperature was divided into discrete ranges and the heat transfer was calculated for each range of outside temperature.

$$\dot{Q}_{wall} = \frac{T_{in} - T_{out}}{R_{wall}} \quad (8)$$

The heating needed to maintain the temperature of the home was calculated from the heat leaving the home through the walls, ceiling, floor, windows, and air infiltration, as shown in Equation 9.

$$\dot{Q}_{heating} = \dot{Q}_{wall} + \dot{Q}_{ceiling} + \dot{Q}_{floor} + \dot{Q}_{windows} + \dot{Q}_{air} \quad (9)$$

Equation 10 determined the total energy required to be supplied to the heating system, which was calculated by multiplying the heat loss occurring from the home at a given outside temperature range by the amount of time (τ_{bin}) the outside temperature exists in that range, and divided by the efficiency of the system, which also may vary by outside temperature.

$$Q_{system} = \frac{\tau_{bin}\dot{Q}_{heating}}{\eta} \quad (10)$$

The calculation to determine the amount of cooling needed for the home was done the same way except that the home gained energy through the walls, ceiling, floor, windows; in addition, sunlight and the energy produced by people and appliances were included as energy gained by the home.

The amount of non-renewable energy needed by systems that largely use thermal solar collectors for heating and cooling was determined by first obtaining the amount of solar energy available each month in each of the five cities. The amount of solar energy available was based on the amount of daylight and cloud cover of each city. The cities chosen include a large variety of latitudes and cloud cover, therefore the amount of sunlight at different latitudes and climates were accounted for in the simulation. After this, it was determined how much of the available solar energy was collected, and incorporating the efficiency of each system that uses renewable energy, it was determined how much heating, cooling, and hot water was produced. If the system using renewable energy did not provide enough heating and cooling it was determined how much non-renewable energy was needed to meet the requirements by incorporating the efficiency of the system using non-renewable energy. The size and number of solar collectors of the absorption system were chosen such that renewable energy provided much of the cooling in the summer time, between 50 to 90% except in Sacramento, CA where 100% was provided, and a noticeable or significant amount of heating in the winter, between 30 to 70%. The thermally driven heat pump was sized to provide between 15 to 50% of cooling and 30 to 45% of heating except in Phoenix, AZ where 80% of heating was provided. The same size units were used in all locations because they are the smallest units available but less solar collectors were needed in Phoenix, AZ and Houston, TX to operate these units as there is more sunlight available in these climates. Less solar collectors is evident by the lower cost of solar collectors shown in the Results and Discussion Section. If the thermal solar collectors provided all the heating and cooling the excess solar energy collected was considered to be discarded and not used in other months. The power used by the pumps and controls of the thermal solar collectors were small and therefore neglected in the analysis.

Economics

The economics assumed that all systems lasted 20 years and that there was no value at the end of the 20 years. In addition, it was assumed that the price of each system did not vary throughout the country, thereby removing the effect of different costs of living. The economics, purchase and operating costs, of each system were determined by first obtaining the average cost of natural gas and electricity of each state of the corresponding cities that were studied as shown in Table 1 (EIA 2006). The energy obtained from 1 ft^3 of natural gas was determined to be 1027 Btu/ft^3. The escalation rates, ER, of natural gas and electricity, all with respect to 2006 dollars, were also obtained (EIA 2006), which stated that natural gas and electricity will escalate, in decimal form, at 0.003 and –0.002, respectively.

Table 1. Average 2006 Utility Rates of Selected States

	CA	KY	MN	AZ	TX
Natural Gas					
Rate ($/GJ)	12.5	16.8	12.3	16.2	13.2
($/MBtu)	11.89	15.97	11.70	15.39	12.59
Electricity					
Rate ($/kWh)	0.1436	0.0686	0.0872	0.0928	0.1258

The interest rate, after removing the effect of inflation, which was used in conjunction with the additional cost or savings of each of the newer systems, compared to the furnace and air conditioner base system, was calculated by averaging the certificate of deposit (*CD*) and mortgage rates, and then removing the effect of inflation, *F*, as shown in Equation 11 (Fuller et al. 2005).

$$I = \frac{[(1+M)+(1+CD)]/2}{(1+F)} - 1 \tag{11}$$

Inflation was removed from the analysis because it is unpredictable, and also the results of this analysis are in 2006 dollars, making it more intuitive. The most recent mortgage, *CD*, and inflation rates were used and were, in decimal form, 0.064, 0.054, 0.044, respectively. From this information the interest rate, removing the effect of inflation, and was calculated to be, in decimal form, 0.014.

The utility rates, *UR*, of natural gas and electricity were calculated for each year, *N*, by knowing the initial rates and the escalation rate, *ER*, for each utility, based the recursive formula of Equation 12.

$$UR_N = (UR_{N-1}) \times (1 + ER) \tag{12}$$

The operating costs, OC, for each system and year were then calculated using Equation 13 from the utility rate and the amount of energy used by each system, which is Q_{system} in Equation 10 while keeping separate electricity and natural gas. As seen in Equation 13, the utility rates of natural gas and electricity escalated each year but the consumption of natural gas and electricity remained the same each year.

$$OC_N = (UR_{NG,N} \times E_{NG}) + (\mathrm{UR}_{\mathrm{EL,N}} \times E_{EL}) \tag{13}$$

After the operating costs of each system of each year were determined, the life cycle costs analysis was completed in the following way. The initial additional cost of each system compared to the base system was calculated and was called recovery cost, *RC*, and was recorded as a negative value. The difference in operating costs, *DOC*, a positive value, equal to the operating cost of the base system minus the operating costs of the newer systems, was calculated and added to the recovery cost of each system, making the recovery cost a smaller negative value. At the end of each year, interest was calculated and added to the recovery cost of each system. If the recovery cost was still negative, the newer system had not paid back compared to the base system, the interest was also negative, which created a larger recovery cost. If the recovery cost was positive, the newer system had paid back compared to the base system, the interest was also positive and the recovery cost became larger and is considered the added savings, *AS*, of the system at that year. The payback period, *PP*, is when the added savings became a positive value. Equations 14 through 16 calculate the recovery cost, *RC*, added savings, *AS*, and payback period, *PP*, for each of the 20 years.

$$RC = \begin{cases} \text{Year 1}, = (AC + DOC_1) \cdot (1+I) \\ \text{Year } N > 1, = (RC_{N-1} + DOC_N) \cdot (1+I) \end{cases} \tag{14}$$

$$AS = \begin{cases} If\ (RC < 0), 0 \\ Else,\ RC \end{cases} \tag{15}$$

$$PP = \begin{cases} \text{Year 1} \begin{cases} If\ AS_1 > 0, \text{“}yes\text{”} \\ Else,\ 0 \end{cases} \\ \text{Year } N \begin{cases} If\ [(AS_N > 0), \text{ AND } (AS_{N-1} = 0)],\ yes \\ Else,\ RC \end{cases} \end{cases} \tag{16}$$

Exergy

Exergy indicates the usefulness of the energy to do work, exergy transfer accompanies energy transfer and when energy is transferred in an irreversible process the exergy at the end is less than at the beginning of the process, therefore exergy is destroyed. Exergy transfer accompanying heat transfer is shown in Equation 17 (Moran and Shapiro 2004).

$$\text{Energy transfer from heat} = \left(1 - \frac{T_0}{T_b}\right) Q \tag{17}$$

The exergy balance of a closed system, such as the overall heating and cooling systems and the home, is shown in Equation 18. In the equation work, *W*, is positive going out of the system and energy in the form of work, such as electricity, directly transfers the same amount of exergy.

$$\begin{aligned} 0 &= (Exergy\ ) - (Exergy\ out) \\ &\quad - (Work) - (Exergy\ destroyed) \end{aligned} \tag{18}$$

Exergetic efficiency is described as the useful exergy of the process, such as the exergy needed to heat or cool the home, with respect to the total amount of exergy input to the system. Exergetic efficiency describes how well matched the input energy is to that of the end use and ranges between zero and unity. When the input energy is well matched to the end use and the system is relatively internally reversible, the exergetic efficiency approaches unity.

Others (Rosen and Dincer 2004a, 2004b; Dincer and Rosen 2000) have used exergy efficiency and exergy destruction to study heating, cooling, and power cycles. These researchers determined that exergy analysis provides enhanced understanding of systems and assists in design and improvements efforts of cycles by examining the exergy efficiency or exergy destruction of each component or of the entire system when comparing entire systems.

The following sources of exergy were accounted for in this study. The exergy transfers from the heat lost and gain by the house for the heating and cooling seasons were included for all houses in all locations and all heating and cooling systems. Exergy of the flue gas from the furnace and of the heat leaving the condenser in the air conditioner were accounted for. The exergy from the transfer of heat to or from the ground was accounted for in the ground source heat pump system. The exergy from heat transfer in the absorber and condenser of the absorption unit was considered as well as the exergy from the heat of the flue gas when the system operated in backup mode and the exergy from the solar energy obtained by the solar thermal collectors. The thermally driven heat pump system included the exergy from the heat transfer to or from the ground and the exergy from the solar energy obtained by the solar collectors. There was no exergy included from solar energy for the systems that did not use solar collectors.

Many temperatures were needed during the exergy analysis. The temperature of the combustion products was determined by estimating that the natural gas was methane and it combusted with 20% excess air. The temperature of the energy leaving the absorber and condenser of the absorption unit was obtained from a current manufacturer of absorption units. The temperature of the heat leaving the condenser of the air conditioner and of the heat transferred between the ground source heat pump and the ground was obtained by consulting an HVAC company (Jacobs 2006). The temperature of the water produced by the thermal solar collectors was used to determine the exergy of that energy and was obtained by consulting an advisor to a company that produces thermal solar collectors (Henkel 2006).

The exergy transfer associated with using energy to fabricate the four different heating and cooling systems was not determined or accounted for. The exergy analysis included the high temperature combustion gases of the furnace or of the absorption unit operating in backup mode but not the change in chemical exergy of the natural gas fuel.

CASE STUDIES

Locations and Climates Observed

The purpose is to model a home in five different U.S. cities and compare the four different heating and cooling systems for each location. These five different cities represent the most common types of climate in the United States. The cities chosen were Louisville, Kentucky; Houston, Texas; Minneapolis, Minnesota; Sacramento, California, and Phoenix, Arizona that represents the climates of mixed-humid; hot-humid; cold; marine; and hot-dry, respectively as shown in Figure 1 (EEBA 2007).

Figure 1 *Climate map of the United States.*

House Specifics

The average size of an American home is approximately 232 m^2 (2500 ft^2) (EIA 2001) and the simulated house is 236 m^2 (2535 ft^2). The house is a "simple" rectangular, one-story, slab-on-grade home. There is no basement because it presents a simpler comparison for those who live in California, Florida or other places that may not have basements because of earthquakes or flooding.

The modeled home faces north and has a length (north, south direction) of 11.9 m (39 ft) and width (east, west direction) of 19.8 m (65 ft). The ceiling height in the home is 2.44 m (8 ft). The foundation sides are insulated to an R-5 value. The outside walls have an overall insulation factor of R-21; this insulation factor is obtained from R-19 fiberglass insulation, sheetrock, plywood, siding. The windows are double-pane with solar control low-e with a wood frame. Solar control low-e reduces infrared radiation heat transfer therefore requiring less heating in the winter and less cooling in the summer. Figures 2a and 2b show the basic drawing of the house. The simulated house has a certain window-to-wall ratio on each of its sides. The window area facing north is 7.57 m^2 (81.5 ft^2) and the window area facing south is 13.3 m^2 (142.9 ft^2). The south facing windows allow more sun in during the winter and less sun inside during the summer with proper overhangs on the south side of 0.61 m (2 ft) and also on the north side to be symmetric. There is a 2.74 m^2 (29.5 ft^2) window on each east and west wall. The interior walls are not insulated, the attic is unconditioned, and the attic floor (just above the ceiling) is insulated to a value of R-60. The front door is a steel/wood/steel construction. The entire home is one zone of temperature control because this is common in residential buildings.

The house orientation and design and materials chosen were considered to be energy-saving and therefore assume passive methods of energy conservation. This may lead to a problem that decreasing the total energy consumption of the house by passive methods, may lead to a smaller variation in performance of the systems, thereby decreasing the overall

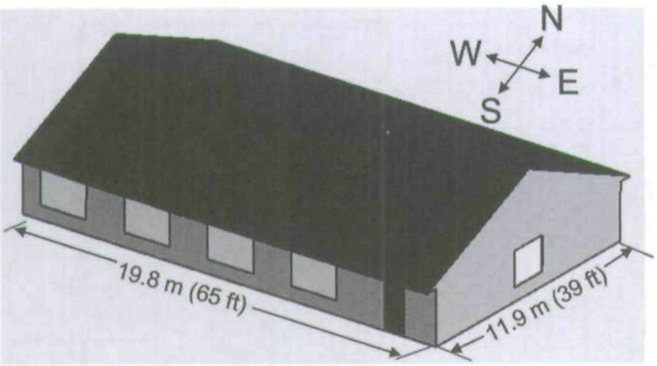

Figure 2A *Drawing of the model house facing south and east.*

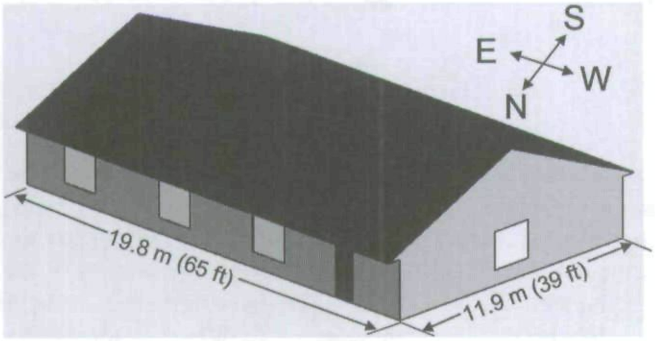

Figure 2B *Drawing of the model house facing north and west.*

Table 2. Annual Heating and Cooling Requirements of the Home in Five Cities

	Heating GJ (MBtu)	Cooling GJ (MBtu)
Louisville, KY	70.1 (66.4)	29.0 (27.5)
Phoenix, AZ	31.6 (29.9)	44.2 (41.9)
Sacramento, CA	54.9 (52.0)	20.4 (19.3)
Minneapolis, MN	123.4 (117.0)	14.9 (14.2)
Houston, TX	27.9 (26.4)	49.0 (46.5)

advantages to the newer technologies and extending their payback periods (McCabe and Purcell 1983). Even with this potential drawback it is best practice and appropriate to perform the simulation with the energy efficiency attributes that are currently available. The heating and cooling loads of the home in each location are shown in Table 2.

Systems

Sixty-nine percent of new homes built in 2004 used natural gas for heating and 90% of new homes in the United States have central air conditioning (U.S. Census Bureau 2006). This is why a condensing gas furnace and high efficiency air conditioner is a great standard to compare against other heating and cooling systems. The systems were assumed to have a life of 20 years at which time there is no value left to them and that all maintenance costs were equal for all systems. The smallest, high-efficient systems that used non-renewable energy were selected because they met the maximum rate of heating and cooling needed for the home in all climates. These systems all had two stages and will work in all climates, what will be different between climates is how often both stages operate. It is also advantageous that all the systems are the same for comparison purposes. The smallest systems that used renewable energy were also selected.

Condensing Furnace and Central Air Conditioner

The first system studied was a condensing gas furnace and a high-efficiency air conditioner with a natural gas hot water heater. An air conditioner utilizes vapor compression refrigeration cycle to cool a home. The equipment chosen for modeling was a two-stage condensing gas furnace with 93% efficiency and a high efficiency air conditioner (Jacobs 2006).

Ground Source Heat Pump

The second system being examined was a geothermal heat pump that uses vertical loops buried in the ground. The vertical loops are 100 to 300 ft deep (Wright and Covin 1993) in order to reach a depth where the earth temperature is constant. These vertical loops transfer heat to and from the ground instead of the air. For cooling, the heat is removed from the home and transferred into the ground. For heating, the heat is removed from the ground and transferred to the home.

The specific vertical ground source heat pump chosen is a two-stage heat pump with electric resistance backup heating. The heating and cooling COP for the actual heat pump varies with location based on ground temperature. The ground temperature was based on the average water temperature. Also, it is assumed that all heat generated from the electric backup resistance heater enters the house. (Jacobs 2006)

Absorption Air Conditioner with Direct Heating

The third system investigated was a unit consisting of thermal solar collectors that provided energy to an absorption air conditioner, direct heating and hot water heater with natural gas backup for heating, cooling, and hot water. As seen in Figure 3 (Moran and Shapiro 2004), the absorption air conditioner has the same components as a vapor compression cycle, but the compressor is replaced at least with an absorber, pump, heat exchanger and refrigerant generator that operates with energy provided by solar collectors. The absorption unit being studied uses a water/lithium bromide solution where water is the refrigerant and lithium-bromide is the absorbent.

In heating mode, the water/lithium bromide solution is heated by solar energy in the refrigerant generator; the water vapor is separated from the lithium bromide and travels to the low pressure apparatus, heating the water within the copper tubes. Then the water, operating as refrigerant, condenses and is returned to the high temperature refrigerant generator, and starts the process again. For instances where there is not enough solar energy available, heating and cooling is accomplished by a backup natural gas system.

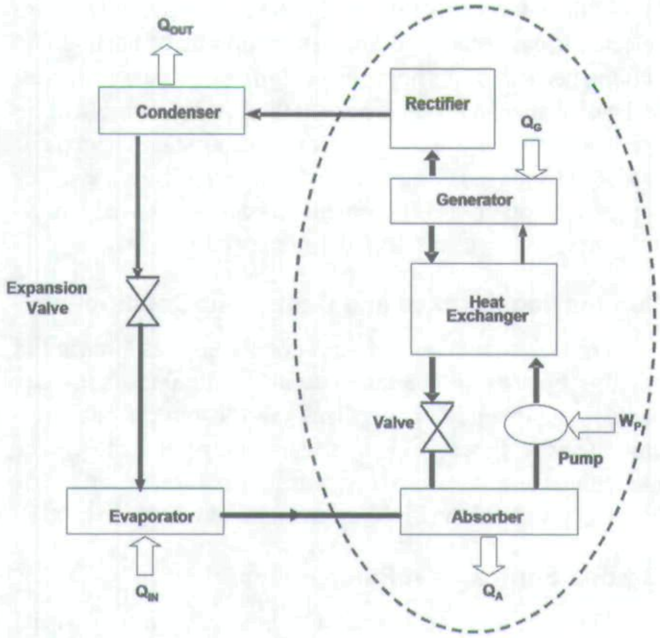

Figure 3 *Diagram of absorption air conditioning cycle.*

One China-based company is the only known company that produces a residential-sized absorption unit. The absorption unit is a double effect, 14.1 kW (4 RT) absorption cooler. Stationary parabolic solar collectors were a good fit to provide solar energy to this unit because the solar collectors have provided temperatures up to 160°C (320°F) in the cooling season which is high enough to operate the absorption unit and 76.7°C (170°F) in the heating season which provides space heating. The cooling COP for the absorption unit is 1.1 and the heating COP is 1.0 (Henkel 2006). There is concern that this unit is oversized for this particular home that requires about 7 kW (2 RT) of cooling in most of the climates; however the cost of absorption units do not decrease much below a certain cooling capacity rate, therefore since the cost does not change much, it was acceptable to model a larger unit. One disadvantage of the larger unit is that it would not provide as much dehumidification as a more appropriately sized unit.

Thermally Driven Heat Pump (TDHP)

Figure 4 shows a diagram of the thermally driven heat pump, TDHP (Feutz et al. 2003).The TDHP is a heat pump that uses a hydraulic compressor that operates from liquid refrigerant that was pressurized by a pump and then vaporized from the thermal solar energy. This refrigerant enters the high pressure side of the hydraulic compressor and compresses the low pressure refrigerant vapor coming from the evaporator by a double-ended hydraulic piston and check valves. Both fluids used in the hydraulic compressor are the same refrigerant and both are then routed to the condenser. When insufficient solar energy is available the system is powered by an electric compressor or both compressors can operate simultaneously. Also, the TDHP only requires energy at 57.2°C (135°F), therefore less expensive solar collectors were used in the simulation compared to the collectors needed for the absorption system. However, the cooling COP of the TDHP is low averaging 0.35, while in heating mode the system is better than simply direct heating with a COP of 1.25. When the system operates with the electric backup compressor the COP is the same as the electrically powered ground source heat pump (Henkel 2006).

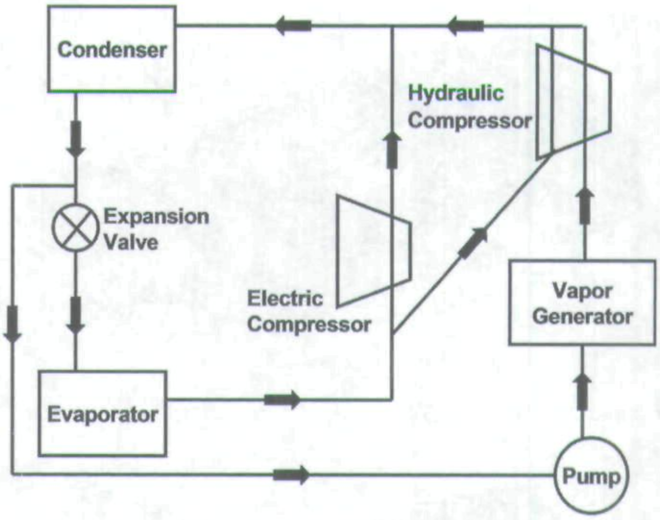

Figure 4 *Diagram of thermally activated heat pump cycle.*

RESULTS AND DISCUSSION

Economics

Table 3 includes the initial and operating costs of all systems in all locations and how each of the newer systems compare to the base system. The vertical ground source heat pump, GSHP, pays back in all five locations between 4 and 15 years depending on the climate. The simulation of the GSHP was very favorable in climates that required a significant amount of cooling and/or heating such as in Louisville, KY, Minneapolis, MN, and Phoenix, AZ where the ground source heat pump paid back in 4, 7, and 7 years and produced added savings of more than \$19,000, \$9000, and \$8000 in 2006 dollars, respectively. The absorption unit, ABS, does not pay back in any of the five cities within 20 years. The TDHP paid back, compared to the furnace but likely not the GSHP, in Louisville, KY and Minneapolis, MN where there is a larger heating load than the other three cities. The TDHP had a favorable heating COP but relatively poor cooling COP making it beneficial in climates that need substantial amount of heating and some air conditioning.

A simulation was performed to determine what the escalation rate of natural gas and electricity must be to have systems break even if they did not pay back during the original study. The results for this analysis are seen in Table 4. Most of these escalation rates are impractical, if the Department of Energy's forecast is accurate the price of electricity and natural gas will not change much; however, if there are drastic

Table 3. Economic Comparison of Four Systems Simulated in Five Cities

		FURNACE	GSHP	ABS	TDHP
KY	System cost ($)	11,211	14,639	11,500	16,789
	Solar panel cost ($)	N/A	N/A	15,840	10,000
	2006 electricity operating cost ($)	117	514	0	332
	2006 natural gas operating cost ($)	1416	0	1,121	0
	Payback Year	N/A	4	N/A	15
	Added savings after 20 years ($)	N/A	19,888	N/A	7888
AZ	System cost ($)	11,211	14,639	11,500	16,789
	Solar panel cost ($)	N/A	N/A	13,860	10,000
	2006 electricity operating cost ($)	310	517	0	270
	2006 natural gas operating cost ($)	747	0	485	$0
	Payback Year	N/A	7	N/A	N/A
	Added savings after 20 years ($)	N/A	8383	N/A	N/A
CA	System cost ($)	11,211	14,639	11,500	16,789
	Solar panel cost ($)	N/A	N/A	15,840	10,000
	2006 electricity operating cost ($)	186	777	0	493
	2006 natural gas operating cost ($)	830	0	513	0
	Payback Year	N/A	15	N/A	N/A
	Added savings after 20 years ($)	N/A	1689	N/A	N/A
MN	System cost ($)	11,211	14,639	11,500	16,789
	Solar panel cost ($)	N/A	N/A	15,840	10,000
	2006 electricity operating cost ($)	75	1271	0	677
	2006 natural gas operating cost ($)	1,729	0	1,279	0
	Payback Year	N/A	7	N/A	15
	Added savings after 20 years ($)	N/A	9266	N/A	6545
TX	System cost ($)	11,211	14,639	11,500	16,789
	Solar panel cost ($)	N/A	N/A	13,860	10,000
	2006 electricity operating cost ($)	365	630	0	439
	2006 natural gas operating cost ($)	570	0	632	0
	Payback Year	N/A	12	N/A	N/A
	Added savings after 20 years ($)	N/A	2926	N/A	N/A

changes in prices, some of these rates might be feasible. To complete this simulation, a relationship was needed between the escalation rates of each utility, since the projected escalation rates were both very small, the escalation rates were set equal to each other. With this relationship the escalation rate of the utilities were calculated to have the systems that did not payback in twenty years in the original simulation, most of the systems that largely use renewable energy were in this category.

Table 5 indicates the prices of solar panels that are necessary for the systems that largely use renewable energy to pay back in areas where they did not pay back in the original study. In most cases, the solar panel prices needed for the systems to pay back are reduced by about 50%. However, the solar panel price needed for the TDHP in Phoenix, AZ to pay off was only slightly less than the current price.

Figures 5 and 6 show the total cost of operating each of the systems in the locations of Louisville, KY and Sacramento, CA, respectively. These costs are shown because in Louisville, KY the newer systems provide significant energy and cost savings and the GSHP pays back in the shortest time and in Sacramento, CA the newer systems are not that economically advantageous and the GSHP pays back in the longest time of any location.

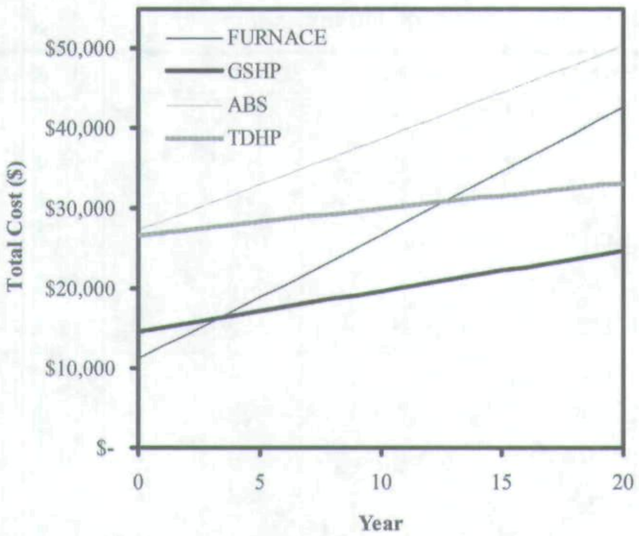

Figure 5 *Total purchase and operating costs of all systems in Louisville, KY.*

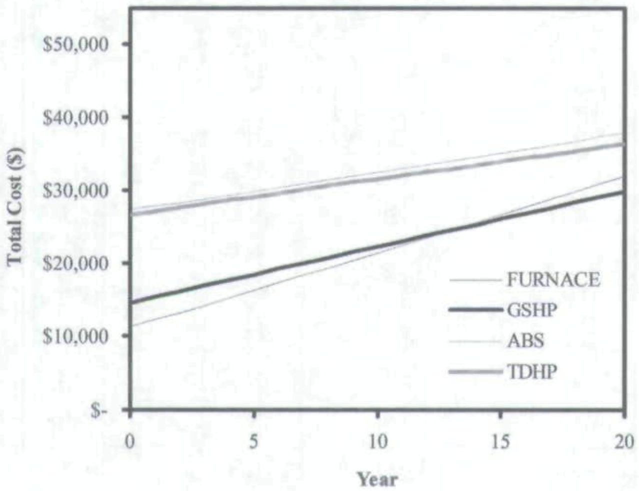

Figure 6 *Total purchase and operating costs of all systems in Sacramento, CA.*

Energy and Exergy

Table 6 shows the first-law coefficient of performances or efficiencies of each of the systems as well as the second-law exergetic efficiencies. The first-law efficiencies were calculated by two methods: the first method was relative to the amount of non-renewable energy used, while the second method was relative to the total amount of energy used. The GSHP and TDHP each had about the same amount of exergy destruction in all locations, and had the greatest difference in Minneapolis, MN, and always had less exergy destruction than the other two systems. The TDHP and GSHP also had the highest exergetic efficiencies in all locations; the exergetic efficiencies of the two systems were approximately equal in Phoenix, AZ, Sacramento, CA, and Houston, TX and the TDHP had a slightly higher efficiency in Minneapolis, MN,

Table 4. Utility Escalation Rates and Prices Needed for Renewable Energy Systems to Pay Back in 20 Years

		ABS	TDHP
	Escalation Rate (%)	8.22%	N/A
KY	Natural Gas @ Year 20 ($/MBtu)	$69.74	N/A
	Electricity @ Year 20 ($/kWh)	$0.308	N/A
	Escalation Rate (%)	3.78%	1.46%
AZ	Natural Gas @ Year 20 ($/MBtu)	$31.15	$20.28
	Electricity @ Year 20 ($/kWh)	$0.188	$0.122
	Escalation Rate (%)	6.34%	5.63%
CA	Natural Gas @ Year 20 ($/MBtu)	$38.23	$33.64
	Electricity @ Year 20 ($/kWh)	$0.462	$0.406
	Escalation Rate (%)	5.92%	N/A
MN	Natural Gas @ Year 20 ($/MBtu)	$34.88	N/A
	Electricity @ Year 20 ($/kWh)	$0.260	N/A
	Escalation Rate (%)	9.84%	6.12%
TX	Natural Gas @ Year 20 ($/MBtu)	$74.85	$38.93
	Electricity @ Year 20 ($/kWh)	$0.748	$0.389

Table 5. Solar Panel Prices Needed for Systems to Pay Back in 20 years

	ABS ($/m^2)	TDHP ($/m^2)
Current	$600	$400
KY	$263	N/A
AZ	$417	$335
CA	$322	$158
MN	$340	N/A
TX	$209	$132

and Louisville, KY. The TDHP had equal or greater exergetic efficiency, and therefore less exergy destruction, because it used low grade solar thermal energy for the useful purpose of heating and cooling and when it was using non-renewable energy it operated as the ground source heat pump system. The base system of natural gas furnace and electric air conditioner had the largest amount of exergy destruction of any system in all locations due to the high temperature of combustion of natural gas that is used inefficiently to only provide heat at room temperature.

Figures 7 and 8 give the amount of natural gas and electricity respectively that is being consumed annually during the heating and cooling seasons. In Figure 7, the ground source heat pump and the thermally driven heat pump are not listed because those systems do not use natural gas; whereas the absorption unit does not use significant amounts of electricity therefore it is not shown in Figure 8.

CONCLUSIONS

The currently available ground source heat pump (GSHP) is economically viable in all locations simulated and is especially economically advantageous in climates with significant

Table 6. Efficiency or COP and Exergy Comparison of Four Systems Simulated in Five Cities

		FURNACE	GSHP	ABS	TDHP
KY	Heating energy efficiency (%) or COP relative to non-renew energy	93%	5.0	158%	8.7
	Heating energy efficiency (%) or COP relative to total energy	93%	5.0	93%	2.0
	Cooling COP relative to non-renewable energy	5.1	6.5	2.6	8.0
	Cooling COP relative to total energy	5.1	6.5	1.1	1.5
	Overall energy efficiency relative to non-renew energy (decimal)	1.21	4.67	1.79	8.53
	Overall energy efficiency relative to total energy (decimal)	1.21	4.67	0.98	1.83
	Cooling exergetic efficiency (decimal)	0.048	0.062	0.020	0.052
	Heating exergetic efficiency (decimal)	0.066	0.250	0.096	0.311
	Overall exergetic efficiency (decimal)	0.064	0.210	0.078	0.237
	Annual exergy destruction (GJ)	65	17	50	15
AZ	Heating energy efficiency (%) or COP relative to non-renew energy	93%	5.3	388%	32.7
	Heating energy efficiency (%) or COP relative to total energy	93%	5.3	93%	1.4
	Cooling COP relative to non-renewable energy	4.7	5.2	7.2	8.7
	Cooling COP relative to total energy	4.7	5.2	1.1	1.0
	Overall energy efficiency relative to non-renew energy (decimal)	1.81	4.97	5.39	12.25
	Overall energy efficiency relative to total energy (decimal)	1.81	4.97	1.03	1.12
	Cooling exergetic efficiency (decimal)	0.135	0.137	0.081	0.131
	Heating exergetic efficiency (decimal)	0.045	0.203	0.110	0.264
	Overall exergetic efficiency (decimal)	0.068	0.163	0.093	0.174
	Annual exergy destruction (GJ)	41	16	21	14
CA	Heating energy efficiency (%) or COP relative to non-renew energy	93%	5.3	204%	10.9
	Heating energy efficiency (%) or COP relative to total energy	93%	5.3	93%	2.0
	Cooling COP relative to non-renewable energy	5.0	7.0	*	14
	Cooling COP relative to total energy	5.0	7.0	1.1	0.7
	Overall energy efficiency relative to non-renew energy (decimal)	1.19	5.45	2.80	11.53
	Overall energy efficiency relative to total energy (decimal)	1.19	5.45	0.98	1.31
	Cooling exergetic efficiency (decimal)	0.091	0.137	0.073	0.074
	Heating exergetic efficiency (decimal)	0.045	0.195	0.077	0.240
	Overall exergetic efficiency (decimal)	0.048	0.184	0.077	0.185
	Annual exergy destruction (GJ)	52	12	30	12
MN	Heating energy efficiency (%) or COP relative to non-renew energy	93%	4.4	136%	6.4
	Heating energy efficiency (%) or COP relative to total energy	93%	4.4	93%	2.4
	Cooling COP relative to non-renewable energy	5.2	9.0	11.4	12.9
	Cooling COP relative to total energy	5.2	9.0	1.1	1.1
	Overall energy efficiency relative to non-renew energy (decimal)	1.02	3.08	1.51	6.77
	Overall energy efficiency relative to total energy (decimal)	1.02	3.08	1.00	2.08
	Cooling exergetic efficiency (decimal)	0.037	0.066	0.022	0.040
	Heating exergetic efficiency (decimal)	0.081	0.211	0.108	0.342
	Overall exergetic efficiency (decimal)	0.080	0.206	0.103	0.314
	Annual exergy destruction (GJ)	108	37	79	20
TX	Heating energy efficiency (%) or COP relative to non-renew energy	93%	5.6	219%	17.2
	Heating energy efficiency (%) or COP relative to total energy	93%	5.6	93%	1.6
	Cooling COP relative to non-renewable energy	5.0	6.2	2.2	7.4
	Cooling COP relative to total energy	5.0	6.2	1.1	1.7
	Overall energy efficiency relative to non-renew energy (decimal)	1.90	5.76	2.18	9.31
	Overall energy efficiency relative to total energy (decimal)	1.90	5.76	1.09	1.67
	Cooling exergetic efficiency (decimal)	0.055	0.068	0.022	0.060
	Heating exergetic efficiency (decimal)	0.047	0.232	0.088	0.270
	Overall exergetic efficiency (decimal)	0.049	0.132	0.045	0.129
	Annual exergy destruction (GJ)	32	12	32	12

*The absorption system provides all the cooling needed from renewable energy therefore neglecting energy used in pumps and controls, as previously discussed, the cooling COP relative to non-renewable energy is infinite.

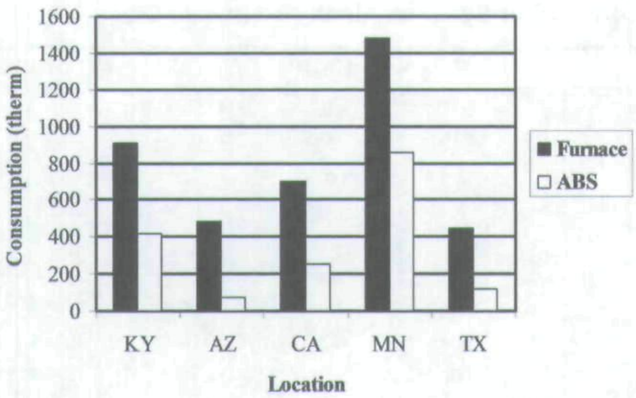

Figure 7 *Natural gas consumption of heating systems in five cities.*

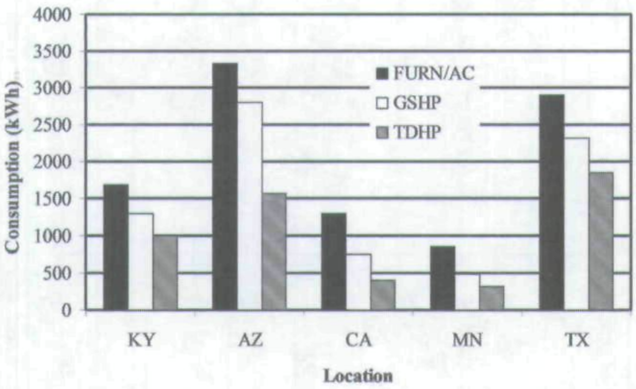

Figure 8 *Electricity consumption of cooling systems in five cities.*

heating and cooling requirements. Homeowners and construction personnel should be informed about the potential significant economic savings that exist from GSHP systems. The systems (ABS and TDHP) that use renewable energy to provide a portion of the heating and cooling requirements conserve more energy in climates with significant heating and cooling requirements but were, in general, not economically advantageous compared to the furnace system and even less economically advantageous if these systems were compared to the GSHP as the base system. In addition, the estimation of how high energy costs would need to raise to make systems that use renewable energy economically viable is much greater than predicted even when comparing them to the base system of the furnace and air conditioner. Also, the estimation of the decrease in price of thermal solar collectors to make systems that use renewable energy economically competitive also shows a significant reduction in price needed when operating in most climates studied. To make systems that use renewable energy economically viable there must be a significant increase in the cost of energy or a significant breakthrough in technology of thermal solar collectors that noticeably reduces the price of the collectors.

Exergetic efficiency and exergy destruction show that TDHP and GSHP better match the usefulness of the incoming energy with the end use compared to the FURNACE and ABS systems. The GSHP accomplishes this by using electricity, which is high grade energy, to efficiently pump heat between the house and the ground while the TDHP accomplishes this by using low-grade, thermal solar energy to also pump heat. The exergetic efficiency was low for the ABS system because it does not pump heat in heating mode and it requires more useful higher temperature energy from the thermal solar collectors in cooling mode compared to the TDHP. The FURNACE system had low exergetic efficiency because of the very high temperature combustion products that were only used to the heat the house near room temperature. Using exergetic efficiency to determine optimum systems was relatively successful and valuable and the TDHP using renewable energy did have the highest exergetic efficiency but only slightly higher than the GSHP and the initial additional cost of the thermal solar collectors was too large to make the TDHP economically advantageous in many climates and not economically advantageous when compared to the GSHP.

ACKNOWLEDGMENTS

Angela Bolling received support by the Bridge to Doctorate Fellowship of the National Science Foundation to complete this work. Tom Henkel and Jim Jacobs provided valuable information.

NOMENCLATURE

A	=	area of heat transfer, m^2
AS	=	added savings, \$
CD	=	certificate of deposit rate
DOC	=	difference in operating costs, \$
E	=	energy used, MBtu or kWh
EL	=	electricty
ER	=	escalation rate of energy
F	=	inflation rate
I	=	interest rate
k	=	thermal conductivity, W/(m·K)
L	=	thickness, m
M	=	mortgage rate
N	=	year number, year
η	=	efficiency
OC	=	operating cost of the system, \$
PP	=	payback period, years
$\dot{Q}_{heating}$	=	rate of heat transfer, MBtu
$\dot{Q}_{system}$	=	energy supplied to system, MBtu or kWh
RC	=	recovery cost, \$
RT	=	refrigerant tons 1 RT = 12,000 Btu/h = 3517 W
τ	=	time outside temperature is in certain temperature rangehour
T_0	=	temperature of energy source, K
T_b	=	temperature of boundary, K

T_{in} = inside temperature, °C
T_{out} = outside temperature, °C
Therm = heat energy, 10 therm = 1 MBtu
UR = utility rate, \$/kWh or \$/MBtu

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DISCUSSION

Don Gibbs: Have you considered utilizing low-temperature solar-heated water to supplement the GSHP during the winter (heating) season?

James A. Mathias: No, the project did not include solar-heated water to supplement the GSHP during the winter. Thank you for the suggestion and insight about the possible improvement.