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Geothermal Heat Pumps in K–12 Schools: A Case Study of the Lincoln, Nebraska Schools

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NATIONAL LABORATORY
MANAGED BY UT-BATTELLE
FOR THE DEPARTMENT OF ENERGY

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CONTENTS

Part III. Comparing Life Cycle Costs for Commercial Heat Pump Systems

FIGURES

TABLES

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ABBREVIATIONS AND ACRONYMS

ACC	air-cooled chiller
ASHRAE	American Society of Heating, Refrigerating and Air-Conditioning Engineers
BHEx	borehole heat exchanger
BLCC	building life cycle cost (computer program)
Btu	British thermal unit
cfm	cubic feet per minute
CO_2	carbon dioxide
COP	coefficient of performance
CV	constant volume
diam	diameter
DOE	U.S. Department of Energy
DST	Duct Storage Model
EER	energy efficiency rating
EMS	energy management system
EWT	entering water temperature
GHP	geothermal heat pump
GHWB	gas-fired hot water boiler
GHX	ground heat exchanger
gpd	gallons per day
gpm	gallons per minute
GSB	gas-fired steam boiler
h	hour
HVAC	heating, ventilating, and air-conditioning
kBtu	thousand British thermal units
LCC	life cycle cost
MMBtu	million British thermal units
NIST	National Institute of Standards and Technology
NO_x	nitrogen oxide(s)
ORNL	Oak Ridge National Laboratory
PM	preventive maintenance
RH	relative humidity
SBM	Superposition Borehole Model
SO_2	sulfur dioxide
TMY	typical meteorological year
VAV	variable air volume (air-handling system)
WCC	water-cooled chiller

1. INTRODUCTION

Geothermal heat pumps (GHPs) have been shown to have a number of benefits over other technologies used to heat and cool buildings and provide hot water, combining high levels of occupant comfort with low operating and maintenance costs. Public facilities represent an increasingly important market for GHPs, and schools are a particularly good application, given the large land area that normally surrounds them. Nevertheless, some barriers remain to the increased use of GHPs in institutional and commercial applications. First, because GHPs are perceived as having higher installation costs than other space conditioning technologies, they are sometimes not considered as an option in feasibility studies. When they are considered, it can be difficult to compile the information required to compare them with other technologies. For example, a life cycle cost analysis requires estimates of installation costs and annually recurring energy and maintenance costs. But most cost estimators are unfamiliar with GHP technology, and no published GHP construction cost estimating guide is available. For this reason, estimates of installed costs tend to be very conservative, furthering the perception that GHPs are more costly than other technologies. Because GHP systems are not widely represented in the various softwares used by engineers to predict building energy use, it is also difficult to estimate the annual energy use of a building having GHP systems. Very little published data is available on expected maintenance costs either. Because of this lack of information, developing an accurate estimate of the life cycle cost of a GHP system requires experience and expertise that are not available in all institutions or in all areas of the country.

The lack of confidence in design methods has also led to the perception that GHPs have a high first cost. For example, ground heat exchangers can account for 20–30% of total system installation costs, and cost-effective design requires that they be sized as accurately as possible. A number of vendors have developed computer software to automate the calculations involved in sizing the ground heat exchangers. As shown in Chapter 5 of this report, these programs are now generally accurate; but because of a lack of confidence in the software, many system designers continue to rely on traditional rules of thumb, such as 150 bore feet per ton of cooling capacity installed. In most cases, this leads to oversized ground loops and a more costly installation.

In 1998, Oak Ridge National Laboratory (ORNL) entered into an agreement with the Lincoln, Nebraska, Public School District and Lincoln Electric Service, the local electric utility in the Lincoln area, to study four new, identical elementary schools built in the district that are served by GHPs. ORNL was provided with complete as-built construction plans for the schools and associated equipment, access to original design calculations and cost estimates, extensive equipment operating data [both from the buildings' energy management systems (EMSs) and from utility meters], and access to the school district's complete maintenance record database, not only for the four GHP schools, but for the other schools in the district using conventional space conditioning equipment. Using this information, we were able to reproduce the process used by the Lincoln school district and the consulting engineering firm to select GHPs over other options to provide space conditioning for the four schools. The objective was to determine whether this decision was the correct one, or whether some other technology would have been more cost-effective. An additional objective was to identify all of the factors that make it difficult for building owners and their engineers to consider GHPs in their projects so that ongoing programs can remove these impediments over time.

We began by comparing the annual energy use of the GHP schools with the energy use of the other schools in the district. We found that the four schools with GHPs are among the lowest energy consumers in the district (see Chapter 2). We then used as-built construction plans and site-monitored data to develop a calibrated engineering model of one of the schools, Maxey Elementary, using the TRNSYS modeling software (Chapter 3). This model was calibrated using the site-monitored data (Chapter 4). The calibrated TRNSYS model was then used to benchmark four commercially available software programs for sizing ground loop heat exchangers, to determine whether the methods agreed with one another, and to determine whether their designs were consistent with the ground heat exchangers installed at the site, given the site-monitored data (Chapter 5).

A detailed analysis of the district's maintenance database allowed us to determine per-square-foot planned and unplanned annual maintenance costs for the GHPs and for three other system types used in the district. Designs were developed for these three system types, as alternative space conditioning systems for the Maxey school. We developed new, independent estimates of the installation costs of these three systems and the GHPs, and the DOE-2 model was run to predict the annual energy consumption of each system type for providing heating and cooling for the school (Chapter 6). Finally, all of this information—installed cost, annual energy use and annual maintenance cost—was used to determine the life cycle cost of each of the space conditioning options, assuming a 20-year system life (Chapter 7). The GHPs were found to be the most cost-effective option for the school, with a life cycle cost some 13% lower than the next most attractive technology.

Part I. Modeling the Performance of Geothermal Heat Pumps in Lincoln Schools

2. COMPARISON OF ENERGY USE IN GHP AND NON-GHP LINCOLN SCHOOLS

2.1 INTRODUCTION

Currently, there are more than 600 GHP systems installed at public school facilities across the nation. In the fall of 1995, the Lincoln, Nebraska, school district opened four new elementary schools—Campbell, Cavett, Maxey, and Roper—served by GHPs. The schools have identical floor plans, each with about 69,000 ft^2 of space dedicated to classrooms, offices, meeting rooms, a cafeteria, and a gymnasium. Each school serves approximately 500 students. Figure 2.1 is a photograph of the front entrance of one of the schools, Maxey Elementary.

Fig. 2.1. Maxey Elementary School, one of four identical schools in Lincoln Nebraska, served by geothermal heat pumps.

In each of the schools, the classrooms are mostly situated on the perimeter of the building with the offices and meeting rooms situated near the core. The schools were designed with open floor plans, including low-rise walls and sliding wall partitions to allow for greater flexibility. The floor plan is shown in Fig. 2.2. The schools were also designed to provide significant natural lighting, with large windows in each classroom, skylights in the main corridors, and a courtyard in the center of the building. The school building is mainly of single-story design, but does include a small second floor near the gymnasium, where the main mechanical room for the building is located.

The performance of the GHP installations in the four Lincoln schools is well-documented by electric and gas utility data (in 15-min and monthly intervals) and 10-min energy management system (EMS) data. In addition, the situation at Lincoln is unique in that the district maintains records on facility design, energy performance, and maintenance activities for all facilities within the district. This information allows a comprehensive review of the design and performance of the GHP systems in the Lincoln Public Schools and a comparison of the performance of these schools to others within the school district.

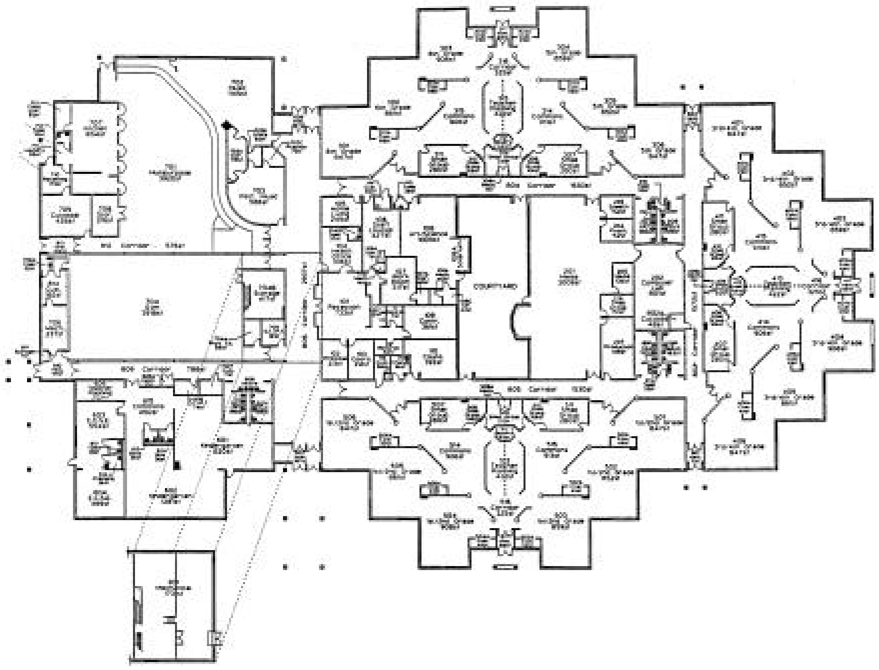


Fig. 2.2. Floor plan of GHP elementary schools in Lincoln, Nebraska.

2.2 SYSTEM DESIGN

The design of the schools' mechanical systems was the result of a collaborative effort between the engineer, the school district, and the local electrical utility (Bantam and Benson 1995). Before an HVAC system type was selected, life cycle costs for five alternative designs were analyzed using energy-consumption and demand profiles from simulations performed with a commercially available software package, operating costs estimated using utility rate schedules, and assumptions about maintenance and equipment replacement intervals and installed costs estimated with a commercially available cost estimating guide. These designs included a variable air volume (VAV) system with air-cooled chillers, a VAV system with water-cooled chillers, gas-fired absorption chillers, vertical GHPs, and water-loop heat pumps. Various time-consuming adjustments were needed during the feasibility study in order to overcome shortcomings in GHP representations in available software tools. Considering capital costs and likely operating and maintenance costs, vertical GHPs were determined to be the best alternative.

Because the schools' annual operating schedules may vary from year to year, heating and cooling loads were estimated on the assumption of full-year operation of the facilities. According to the loads calculated during the design process, under full-year operation, each building was expected to be dominated by cooling loads. The total block cooling load for the 32-zone building was estimated to be 150 tons, while the peak block heating load was 940,000 Btu/h (Kavanaugh 1994). Full-load heating and cooling hours were estimated at 500 and 750, respectively.

Fifty-four heat pumps of various sizes meet the heating and cooling loads at each of the schools. Table 2.1 summarizes the size, number, and capacity of the heat pumps installed at the four schools. Because the heat pumps at the Campbell and Maxey Schools are from one manufacturer, and those at Cavett and Roper from another, there is some difference in the nominal capacities installed, but this difference is minimal and is not expected to alter building performance in any significant manner.

The schools were designed to meet ASHRAE Standard 62-1989, which calls for at least 15 cfm of fresh air per person. At each school, preconditioned outdoor air is provided to classroom and office heat pump units by two nominal 15-ton heat pumps (with two 7.5-ton compressors each), located within a mechanical room. Each large unit operates on 100% outdoor air and feeds this conditioned air into local heat pumps through two central ducts running along the schools' main corridors. Additional outdoor air is provided to assembly areas, such as the multipurpose cafeteria and gymnasium, by a nominal 10-ton unit operating on 40% outdoor air and a nominal 4.5-ton unit with 45% outdoor air. In all outdoor air units, preheat is provided by a hot water coil when ambient temperatures fall below 40°F. Hot water is also supplied to terminal units located in vestibules and other perimeter areas. Hot water is produced by gas-fired boilers, four per school, each one with a capacity of 330,000 Btu/h.

The remaining heat pumps, ranging in size from 1.4 to 4.5 nominal tons, serve individual zones: classrooms, offices, and common group study areas. For the most part, these units are located above the central corridors outside the zones they serve and are easily accessible to maintenance personnel.

At all four schools, the heat pumps absorb and reject heat through a common loop to a borefield consisting of 120 bores arranged in a 12 × 10 pattern with 20-ft spacing. Figure 2.3 shows the layout of the system; the borefields are located under the schools' soccer fields. The bores are 240 ft deep, with diameters of 4.25 in. on the lower 220 ft and 6 in. on the top 20 ft. Fine gravel was used to backfill the boreholes to within 10 ft of the surface, at which point a bentonite plug was used to seal the borehole in order to prevent groundwater contamination from the surface (in compliance with Nebraska state regulations). Since bores at these sites do not penetrate multiple aquifers, a surface plug is sufficient to protect groundwater. Fine gravel pack was judged to provide adequate pipe thermal contact because the static water level was considered to be between 20 and 40 ft (and in most instances closer to 20 ft).

Table 2.1. Heat pumps installed at the four Lincoln schools

Campbell and Maxey Schools				Cavett and Roper Schools			
Designation	**No. of units**	**Tons per unit**	**Capacity (tons)**	**Designation**	**No. of units**	**Tons per unit**	**Capacity (tons)**
HP1H.1	1	1.4	1.4	HP1H.1	1	1.4	1.4
HP1H.2	4	1.4	5.6	HP1H.2	4	1.4	5.6
HP1V	1	1.4	1.4	HP1V	1	1.4	1.4
HP2H	4	2.0	8	HP2H	4	1.8	7.2
HP2V	2	2.0	4	HP2V	2	1.8	3.6
HP3H.1	1	2.0	2	HP3H.1	1	2.3	2.3
HP4H.1	34	3.5	119	HP4H.1	34	3.0	102
HP4H.2	1	3.5	3.5	HP4H.2	1	3.2	3.2
HP4V	1	3.5	3.5	HP4V	1	3.0	3
HP5V.1	1	4.5	4.5	HP5V.1	1	4.6	4.6
HP5V.2	1	4.5	4.5	HP5V.2	1	4.6	4.6
HP6V	1	10.0	10	HP6V	1	10.0	10
HP7V	2	15.0	30	HP7V	2	16.0	32
Total	**54**		**197.4**	**Total**	**54**		**180.9**

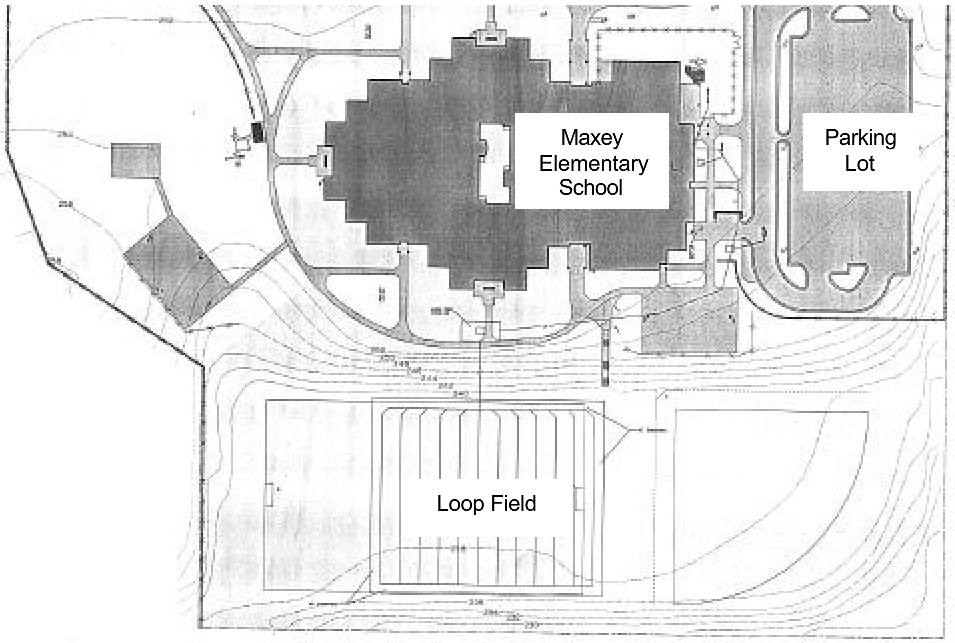


Fig. 2.3. Site plan, with borefield layout, for Lincoln GHP schools.

The loops consist of thermally fused, 1-in.-diam high-density polyethylene piping. At each school, approximately 10,000 gal of water containing 22% ethylene glycol are circulated continuously through the system by a 30-hp pump controlled by a variable-frequency drive.

2.3 ENERGY PERFORMANCE

Total annual energy consumption per square foot is presented for each school in Table 2.2. Because the schools consume both natural gas and electricity, the appropriate form of energy consumption used in benchmarking analyses is source energy. This format accounts for the average efficiency of producing electricity from fossil fuel and delivering it to the site, assumed to be 33%. With an average annual energy consumption of 86.1 kBtu/ft^2, Campbell School is the lowest consumer of energy among the four GHP schools. Maximum annual consumption (at 102.7 kBtu/ft^2) occurred at Cavett in 1997.

In order to compare the performance of the Lincoln GHP schools with that of the rest of the schools in the district, we collected energy consumption and cost data for all schools—37 elementary schools, 11 middle schools, and 5 high schools. We also collected data on the physical characteristics on each school, such as floor area, age and number of expansions, and HVAC system types and ages. Using multiple sources of information (utility account data, 1996 and 1997 Lincoln Public Schools' billing records, facility reports, and equipment inventories) to perform a rigorous verification of the energy and building characteristics information, we derived a qualified data set of 50 schools.

Figure 2.4 compares the schools' average annual source energy use per square foot with that of other schools in the Lincoln School District. The data indicate that the GHP schools are exceptionally low energy users. Campbell school is the lowest: only 12% of the schools in the district use less energy per square foot. Even for the highest average energy user of the four, Maxey School, only 30% of the schools in the district use less energy per square foot. These numbers are even more impressive when it is considered that most of the schools that use less energy than the GHP schools cool less than 15% of their total floor area (only two cool between 70 and 90% of their floor area). In fact, 48% of the schools in the Lincoln district cool less than 25% of their respective floor areas. Only seven schools (including the four GHP schools) air-condition 100% of their floor space. The average source energy use for schools that air-condition less than 25% of their floor space is 100.9 kBtu/ft^2; for schools that air-condition more than 70% of their floor space, the average source energy use is 120.4 kBtu/ft^2. The average annual source energy used consumed by each of the four GHP schools is 93.7 kBtu/ft^2.

Table 2.2. Annual energy consumption (1996, 1997, and average) for four Lincoln GHP schools

School	Source energy consumption (kBtu/ft^2)		
	1996	**1997**	**Annual average**
Campbell	84.7	85.8	85.3
Roper	95.9	93.5	94.8
Cavett	90	102.7	96.4
Maxey	101.4	96.1	98.2

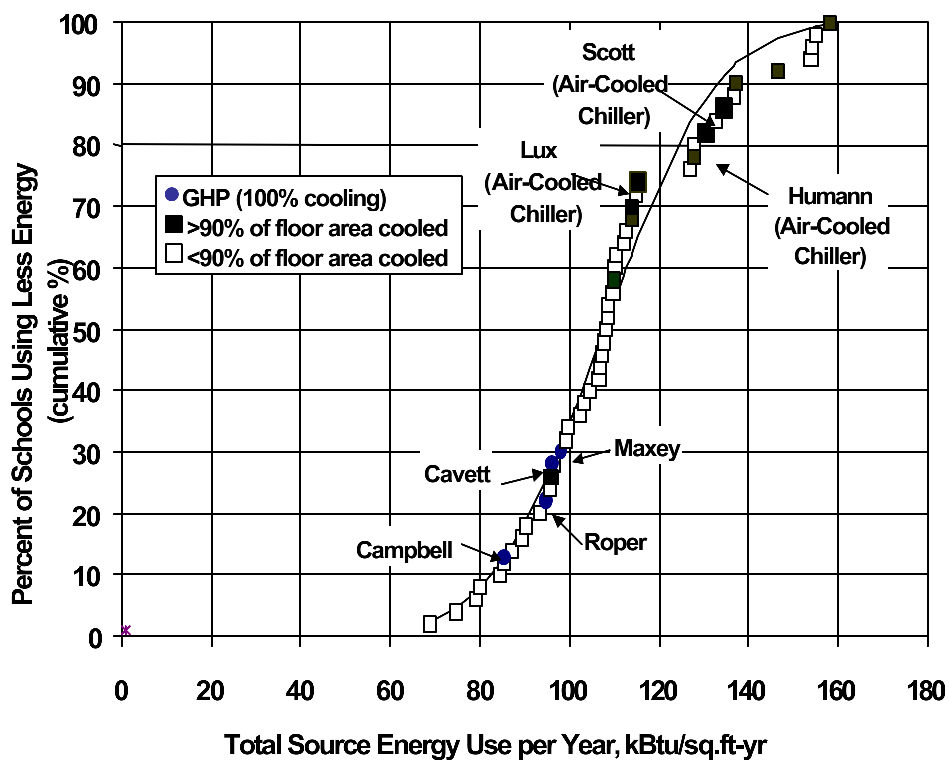


Fig. 2.4. Distribution of average annual source energy consumption for K–12 schools in the Lincoln School District. Solid data markers identify schools that provide space cooling to over 90% of their total floor area. Three schools built in the 1990s with conventional HVAC systems are also identified.

In addition to floor space cooled, older schools do not deliver the same levels of outdoor air per person. Figure 2.4 also allows a comparison of source energy use for all Lincoln schools built in the 1990s. This is useful because both the percentage of floor space cooled and ventilation air delivered are similar for the new schools. The non-GHP schools rely on either air-cooled chillers or air-cooled condensers to cool 79–100% of their floor area. On average, the GHP schools use 26% less source energy per square foot per year than the non-GHP schools.

2.4 EMS DATA ON ENERGY USE

Each of the four Lincoln GHP schools has an energy management system (EMS) that records a wealth of information about the operation of the HVAC system at 10-min intervals. A partial listing of the data available from the system is given in Table 2.3.

Table 2.3. Data collected by Lincoln schools energy management systems at 10-min intervals

Item	Data collected
Heat pumps with outdoor air	Return air temperature
	Outdoor air temperature
	Preheat coil discharge air temperature
	Mixed air temperature
	Supply air temperature
	Supply air humidity
	Compressor status (on/off)
	Reversing valve status (heating/cooling)
	Fan status (on/off)
	Zone heat pumps
Zone heat pumps	Space temperature
	Compressor status
	Reversing valve status
	Fan status
Loop field	Supply temperature
	Return temperature
	Water flow rate
Building energy use	Total electric use
	HVAC electric use

While EMS data were collected for all schools from system initialization in 1995 through 1997, the data set collected in 1996 for Maxey was the most complete and reliable (Carlson 1998). A review of this data provides some additional insight into the operation and performance of Maxey's GHP system.

The GHP systems at Lincoln were designed to satisfy the cooling-dominated loads expected for a schedule based on full-year operation. While some activities do occur during the summer months, Maxey generally operates on a traditional September through June schedule. As a result of this schedule shift, the actual loads exhibited by the building slightly favor heating. This is verified by loop flow rate and temperature data collected by the EMS. In 1996, the total amount of energy rejected by the ground loop was 880 MMBtu, while the total energy absorbed was 1004 MMBtu, a 13% difference. By contrast, the original design documents indicate that, on a full-year operating basis, annual cooling loads were expected to be nearly three times the size of annual heating loads.

As a consequence of traditional September through June operation, the loop fluid temperatures entering the heat pumps never approach the cooling design temperature of 90°F. Figure 2.5 illustrates the distribution of entering water temperatures (EWTs) throughout the year. According to the EMS data, loop temperatures remain below 65°F for nearly 90% of the year. (This distribution is also affected by the use of nighttime setback, during which EWTs remain relatively constant.) The maximum recorded value of EWT was 78°F, while the minimum was 45°F.

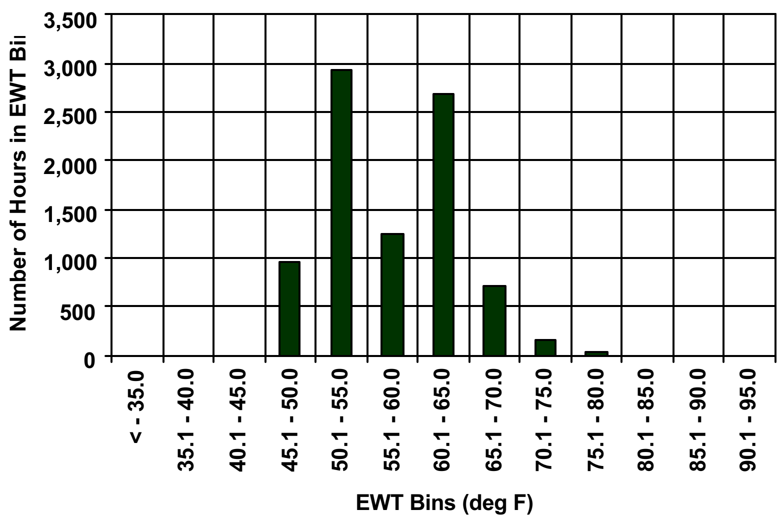


Fig. 2.5. Distribution of 1996 heat pump entering water temperatures (EWTs) as recorded by the Maxey EMS. Total hours of operation were 8760.

A 30-hp pump controlled by a variable-frequency drive varies the loop water flow to maintain supply/return differential pressure as two-way motorized ball valves open and close with compressor cycles at each heat pump. According to test and balance reports and EMS data, the total loop flow through the 54 heat pumps, with all two-way valves open was 552 gpm. This corresponds to approximately 2.8 gpm per installed ton of capacity and about 4.4 gpm per ton of peak block load. Figure 2.6 shows the distribution of loop flow rates in 1996. The maximum flow rate recorded was 552 gpm. Loop flow rates remain below 300 gpm for 78% of the year, and below 150 gpm for 50% of the year. Again, a nighttime system setback impacts this distribution with minimal after-hour flow rates and larger flow rates during recovery periods. Clearly, the potential for energy savings is significant with the use of the variable-frequency drive. Unfortunately, Lincoln reports that the packing on many of the motorized two-way valves has failed, causing variable flow operations to be halted while valves are left in the open position. EMS data confirms that flow rates for Maxey remained constant after April 1997.

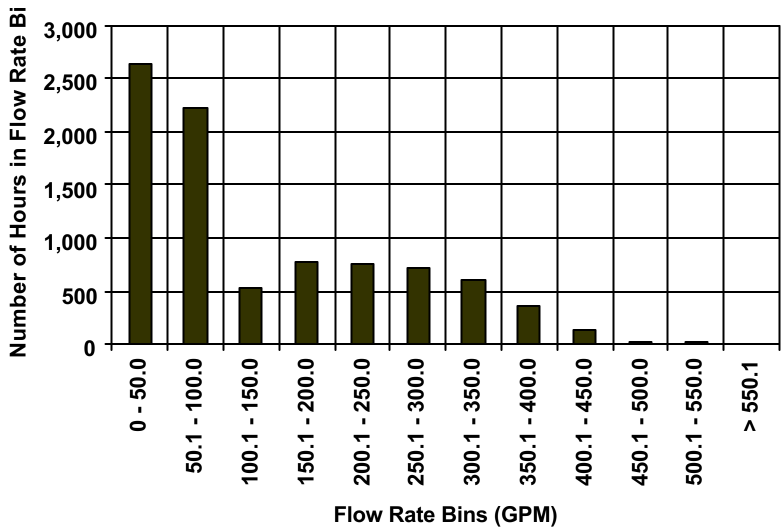


Fig. 2.6. Distribution of 1996 loop flow rates as recorded by the Maxey EMS. Total hours of operation were 8760.

2.5 CONCLUSIONS

Energy data collected over a two-year period indicate that the four new elementary schools in Lincoln, Nebraska, outperform others in the school district. Of 50 schools studied, only 5 consume less energy than the best-performing GHP school; however, these 5 schools cool less than 10% of their total floor area, while the GHP schools cool 100% of their floor area. When compared to other new schools with similar cooling and ventilation loads, the GHP schools used approximately 26% less source energy per square foot of floor area. A variation in performance between the four GHP schools is evident, and is most likely due to differences in occupancy and equipment loading and scheduling.

3. DEVELOPING AN ENGINEERING MODEL

3.1 INTRODUCTION

Because each of the four Lincoln schools with GHP systems has a building EMS, we have extensive data on the operation of the mechanical systems collected by the EMSs (see Table 2.3). This data provided an opportunity to validate the design of the mechanical systems and the borefield, to compare the results of various software packages for the design of borehole heat exchangers in a commercial application, and to assess the accuracy of representations of GHP systems in building energy analysis software packages. In previous work (Hughes and Shonder 1998), similar methods were used to develop a calibrated engineering model of a residential GHP installation at Fort Polk, Louisiana.

The first step in pursuing these opportunities was to develop an engineering model of one of the GHP schools and its associated mechanical systems. Using detailed measured data, along with utility billing information, from Maxey Elementary, we then developed calibrated, detailed TRNSYS-based simulation models of the school building and its heating and cooling systems.

This chapter describes the TRNSYS-based engineering model, and how the model was developed and calibrated.

3.2 THE TRNSYS SOFTWARE

We used the public-domain simulation software TRNSYS (Klein et al. 1996) as the platform for the detailed models because it can operate at any time step, which allows the use of ground heat exchanger models that require small time steps for stability and accuracy, and because it is relatively easy to drive with measured data. TRNSYS is a modular system simulation package in which the user describes the system's components and their interconnections. Components may be equipment, such as a pump or thermostat, or utility modules, such as occupancy forcing functions, weather data readers, integrators, or printers. New component models for the heat pump and borehole heat exchanger (BHEx) are easily added to the existing component libraries to expand the capabilities of the program to include commercial GHP systems.

Unlike most of the commonly used building energy analysis tools, TRNSYS can operate in either energy-rate or temperature-level control (Klein et al. 1996). In energy-rate control, the heating and cooling loads are calculated only on the basis of the net heat losses or gains from the conditioned space. The user specifies the setpoint temperatures for heating and cooling. The program then calculates the amount of energy required to keep the conditioned space at these setpoints. The calculated loads are then passed to the conditioning equipment, which exactly meets these loads at every time step. The advantage of using energy-rate control is that the loads for a given structure can be calculated once and then reused in subsequent equipment and plant simulations. However, the detailed interaction between the conditioned space and the equipment is not treated directly as it is in temperature-level control. Although this interaction is sometimes desirable to model, in this case, given the size of the simulation, it proved impossible, and energy rate control was used.

3.3 THE DETAILED ENGINEERING MODEL

The performance of each of the system's components must be characterized before the system can be modeled. In this case, the components were defined as the building and its characteristics, the load-producing factors within the building, the HVAC equipment, and the BHEx and its associated components. The operation of these components and the weather forcing function are described below.

3.3.1 Building Model

The thermal performance of the school was modeled using a TRNSYS Type 56 multizone building model. The school is divided into 65 thermal zones: 54 zones are served by GHPs, while the remaining 11 have either hot-water unit heaters or are unconditioned. The characteristics of the walls, windows, doors, floors, and ceilings (size, material, orientation, etc.) in each zone were obtained from as-built architectural drawings. The multizone building model in TRNSYS allows the user to build wall types from layers, in which each layer is a unique material. The thermal properties of each layer (thermal conductivity, density, specific heat, and width) are entered by the user or chosen from an existing library. For Maxey School, 19 unique wall types, 4 unique window types, and 26 unique layer types were required.

Each wall in the model is specified as an external wall (one which thermally interacts with the ambient), an adjacent wall (one which thermally interacts with another zone), a boundary wall (one which has a known boundary condition), or an internal wall (one which is just used for capacitance effects). For external walls, the view factor to the sky (for radiation exchange) must be supplied. External walls also require an estimation of the convective and radiative heat transfer coefficient to the ambient. For this study, the following correlation was used:

$$h = 5.7 + 3.8 \times \mathrm{WV} \ , \qquad \text{(Eq. 3.1)}$$

where h is in units of $\mathrm{W/m^2} \times {}^\circ\mathrm{C}$ and the wind speed, WV, is in units of meters per second.

The heat transfer to the ground from the floor slab in the multizone building model was calculated based on the method presented in chapter 27 of the 1997 ASHRAE Fundamentals (ASHRAE 1997). The effective U-value for the slab was the ASHRAE UA value divided by the total floor area of the building. To get our model to recognize this effective U-value for our slab, we had to add a layer of fictitious insulation until the calculated U-value for the slab matched the calculated effective U-value.

The windows at the school were modeled in the Window 4.1 program from Lawrence Berkeley National Laboratory. The outputs from this program can be read into the multizone building model.

The multizone building model requires the user to supply the incident solar radiation on every unique wall or window orientation at each timestep. For most buildings, these are simply the cardinal directions north, south, east, and west plus horizontal. Maxey Elementary School, however, has a significant number of overhangs and wingwalls, each of which required the use of the TRNSYS overhang and wingwall model.

3.3.1.1 Internal Gains

The occupancy data for each zone were obtained from the school for the 9-month school year in 1996. This information was validated and placed into a typical 365-day profile for use with TRNSYS. The students were assumed to be in the building from 8 A.M. to 3 P.M. and in the cafeteria from 11 A.M. to 1 P.M. The school staff was assumed to be in the building from 7 A.M. to 5 P.M. Because the multipurpose room and the gymnasium were used after school and on weekends for most of the year, these areas were assumed to have additional occupants from 5 P.M. to 9 P.M. on weekdays and from 10 A.M. to 9 P.M. on weekends.

From documentation provided by the school, we tabulated the number of students assigned to each classroom for the three years of school operation. The three years were then averaged to obtain a "typical year" occupancy for each classroom. The average total number of students per year during the 3-year period was 434. Since students do not spend their entire day in the classroom, but rather, spend some time in the library, in the hallways, in the art room, in the computer room, etc., we assumed that the students were in their classrooms two-thirds of the time. This left a third of the students to be "distributed" throughout the remaining rooms of the school. The staff (approximately 50 adults each year) were distributed in a similar way.

The sensible and latent gains for the occupants were taken from ASHRAE (ASHRAE 1997) and are summarized in Table 3.1. The values for students were assumed to be 80% of the quoted value for adults.

Table 3.1. Sensible and latent gains for school occupants

Gain	Convective (kJ/h)	Radiative (kJ/h)	Latent (kg/h)
Students in building	144	72	0.088
Adults in building	180	90	0.110
Gymnasium users	355	178	0.3989
Multipurpose room users	180	90	0.110

Another important consideration in the modeling of the building is the internal heat generation due to lights and equipment. The lighting data were taken from the architectural drawings, while equipment gains were estimated, based on the function of the zone. Lighting gains were assumed to be 100% radiative, while the equipment gains were assumed to be 80% convective and 20% radiative. Lighting and equipment schedules were based on occupancy schedules for each zone.

3.3.1.2 Infiltration

The infiltration into the school is the great unknown in the TRNSYS simulations. For this reason, the infiltration parameter was used to "tune" the building to better match the measured data. The infiltration is based on an earlier ASHRAE method where the infiltration is a function of the difference between indoor and outdoor temperature and the wind speed. The formula for determining infiltration is

$$\text{Infiltration} = k1 + k2 \cdot \text{ABS}(T_{\text{inside}} - T_{\text{ambient}}) + k3 \cdot \text{wind speed}\,, \qquad \text{(Eq. 3.2)}$$

where the infiltration is measured in air changes per hour, the temperatures are in Celsius, and the wind speed is in meters per second. In most TRNSYS building calibration exercises, the values of $k1, k2$, and $k3$ would have been varied until the simulated energy consumption closely matched the measured energy consumption. However, because the infiltration in the school is very heavily influenced by the students' going into and out of the school doors, the simulated infiltration is also a function of whether the students are in the building. For this simulation, the following infiltration function was used:

$$\text{Infiltration} = 0.25 + 0.02 \times \text{ABS}(T_{\text{inside}} - T_{\text{ambient}}) + 0.049 \times \text{wind speed} \quad . \qquad \text{(Eq. 3.3)}$$

The infiltration was also classified by zone type because perimeter zones with windows and doors receive more infiltration air than core zones. To account for this, we established multipliers on the base infiltration rate for different types of zones. Table 3.2 summarizes the assumed zone infiltration for reference.

Table 3.2. Assumed zone infiltration

Type of zone	Multiplier when students are in the school	Multiplier when students are not in the school
Vestibule, entryway	5.00	1.00
Classrooms, zones with a window	1.00	0.30
Zones with a window and a door (media, multipurpose, receiving)	1.50	0.40
Interior zones	0.25	0.05
Mechanical rooms	2.00	2.00

3.3.1.3 Capacitance

The Type 56 multizone building model in TRNSYS allows the user to account for the heat capacitance of furnishings and other items in the zones by using a multiplier on the zone air heat capacitance. This value is relatively difficult to estimate, but its effect is rather small on the overall energy consumption. Table 3.3 shows the multipliers were used in the simulation for the different room types.

Table 3.3. Multipliers used in the simulation for different room types

Room type	Room air capacitance multiplier
Classroom	3
Computer, media, art, science	3
Teacher planning	1.5
Corridors	1
Common area	2
Mechanical rooms	1
Vestibules	1
Offices	2

3.3.1.4 Weather Data Inputs

Although ambient temperature and relative humidity were measured at Maxey School for the duration of the monitoring period, no solar data or wind speed information were recorded. This information was not available for Lincoln in 1996. Since the TRNSYS building model requires solar data and wind speed to accurately calculate the loads to be met by the heat pumps, an alternative weather data set was required to drive the simulations. We decided to use typical weather data for the simulations and utilized the TMY2 weather data [typical meteorological year (TMY) summaries by the National Renewable Energy Laboratory)] for Lincoln. The TMY2 weather, which is a monthly best-fit average of 30 years of weather data, contains ambient temperature, relative humidity, incident solar radiation, and wind speed values at hourly increments for a year. Although the use of TMY weather did not allow direct comparison of simulation results to measured values on items like the observed maximum entering water temperature (EWT), it does provide information on how the school would operate in a typical year. Table 3.4 shows the TMY weather data used by the simulations and the weather recorded at the school on an annual basis for the maximum and minimum temperatures and on a monthly basis for the heating and cooling degree days (65°F base).

3.3.1.4 Load Calculation

Table 3.4. TMY weather data for Lincoln and measured data at Maxey School

Data set	Minimum ambient temperature (°F)	Maximum ambient temperature (°F)
TMY for simulations	−14.4	98.1
Measured data	−8.5	97.0

The Type 56 multizone building model in TRNSYS calculates the temperature of each zone in the building, taking into account all the factors previously described (solar radiation, adjacent zone temperatures, ambient temperature, internal gains, etc.), and then calculates the sensible and latent energy required to keep the building temperature at or between the heating and cooling setpoints (74°F occupied and 50°F unoccupied for heating; 76°F occupied and 80°F unoccupied for cooling). For this simulation, the dehumidification setpoint was assumed to be 50% in cooling mode. These sensible and latent loads are then passed to the heat pumps and unit heater models in order to determine the energy use required to meet these loads.

3.3.2 Equipment Models

3.3.2.1 Heat Pumps

There are 54 water-source heat pumps installed in the Maxey School, all located on a common ground heat exchanger loop. Each of the 54 heat pumps interfaces with the EMS for overall building system control (main setpoints, occupied mode, fan control, etc.).

Two of the larger heat pumps (located in the mechanical room) are dedicated to providing preconditioned outside air to 44 of the zone heat pumps. Six of the zone heat pumps receive no outside air; these are primarily located in the corridors of the building. The two remaining heat pumps (the ones for the multipurpose room and for the gymnasium) bring in their own outside air. Each of the four heat pumps that see unconditioned outside air is equipped with a preheat hot water coil to help prevent heat pump unit lockout by safety controls.

The preconditioned outside air is distributed to the zone heat pumps by a ducting network located above the main school corridors. This conditioned outside air is ducted to each heat pump, where it mixes with return air from the space before being conditioned by the zone heat pumps.

The installed sizes and models of the 54 heat pumps were taken from the mechanical drawings of the school. The common loop flow rate, air flow rate, and outside air flow rate were taken from the manufacturer's inspection reports.

Ventilation heat pumps. Two twin-compressor WaterFurnace WXV084 heat pump units (15 nominal tons each) provide the preconditioned outside air (approximately 5000 cfm each) to the zone heat pumps. These two units operate continuously during occupied periods and are inactive during unoccupied times. They operate with 100% outside air, except for a 1-h morning warmup on heating season weekdays when the units condition 100% return air. During this period, the hot water coil completely meets the load (bringing the return air from its set-back condition of 50°F to the occupied setpoint of 74°F), and no compressor operation is allowed. During all other occupied weekday periods, the preheat coil raises the 100% outside air stream to its minimum setpoint of 60°F. The heat pumps then condition the outside air to the building setpoint.

The two ventilation heat pumps are also equipped with electric humidifiers set to 30% relative humidity (RH). Whenever the conditioned air stream falls below 30% RH, the humidifier injects moisture until the RH setpoint is reached.

Outside-air zone heat pumps. The multipurpose room and the gymnasium have zone heat pumps that process their own outside air. Like the two ventilation heat pumps, these heat pumps are equipped with preheating hot water coils and a humidifier. During occupied periods, the heat pump fan runs continuously and the compressor cycles to meet the load. During unoccupied periods, the fan and compressor cycle to meet the load.

Zone heat pumps. Each classroom has its own heat pump and its own thermostat for control of the heat pump. The 50 zone heat pumps take return air (which is a mixture of conditioned outside air and room air during occupied periods) and condition it as demanded by the thermostat. These units have no hot water coils and no humidifiers. During occupied periods, the heat pump fan runs continuously and the compressor cycles to meet the load. During unoccupied periods, the fan and compressor cycle to meet the load.

3.3.2.2 Loop Pump

The common loop pump at Maxey is a variable speed pump that at top speed is rated at 575 gpm and 30 hp at 30 ft of head pressure. During 1996, the pump ran continuously during the day and operated at a minimum flow rate when all the heat pumps were off. Pump data consisted of a plot of normalized power versus normalized flow rate.

To represent this observed pump operation in TRNSYS, a polynomial expression was fit to this data of the form

$$\text{Normalized power} = 0.3540712 + 0.2139276 \times \text{normalized flow rate} + 0.4825143 \times \text{normalized flow rate}^2 \ . \qquad \text{(Eq. 3.4)}$$

This expression was then used to calculate the loop pump power consumption.

The flow rate for the loop pump was determined by summing the individual heat pump flow rates, which were determined by multiplying the heat pump's calculated part load ratio at a given timestep by the rated flow rate across the heat pump at the supply/return pressure differential maintained by the variable speed pump (found in the manufacturer's inspection reports). The power for this flow rate was determined by using the polynomial expression given above.

According to the documentation, the loop pump has a minimum flow rate of 50 gpm for the entire 1996 period—even when all of the heat pumps were off.

Figure 3.1 shows a plot of the simulated daily energy consumption of the loop pump vs daily average ambient temperature. Notice the significant differences between occupied days and unoccupied days. A plot of the binned loop pump flow rate derived from the measured data and simulated data is shown in Fig. 3.2. The differences between the simulated and measured flow are

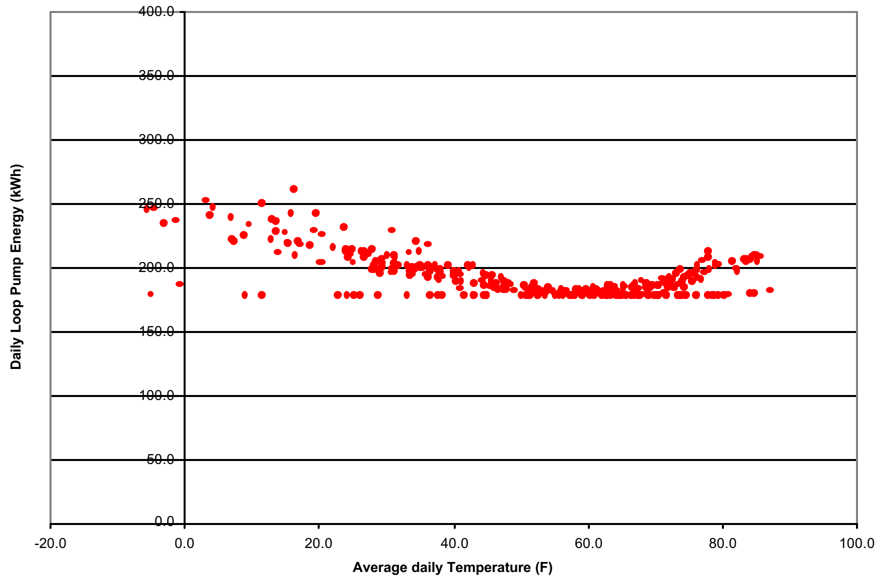


Fig. 3.1. Simulated daily energy consumption of the loop pump vs daily average ambient temperature.

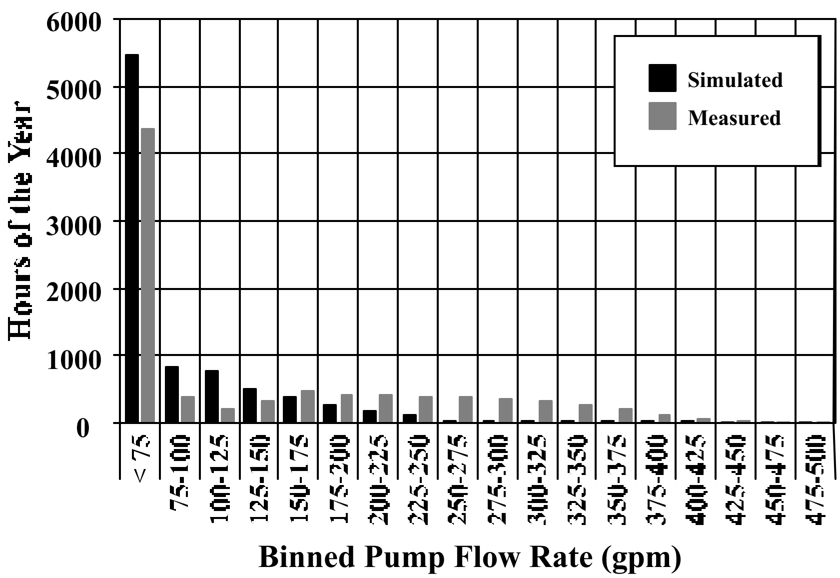


Fig. 3.2. Binned loop pump flow rate derived from simulated and measured data.

probably caused by two important factors: (1) weather differences between the simulated case and the measured case and (2) the tendency in the TRNSYS energy rate control method of calculating the "averaged" flow rate over the timestep to lessen the maximums. For example, if in reality the heat pumps are all on simultaneously for a given 10-min period within an hour, the maximum flow within the hour will be quite high for the measured case. The simulated case will have a smaller flow rate over the hour because the heat pumps all have part load ratios less than

1 for the hour (heat pump flow rate = maximum heat pump flow rate part load ratio). This is one of the inherent disadvantages to using "energy rate control."

3.3.2.3 Humidifiers

The four heat pump units that condition the outside air (the two ventilation heat pumps, plus the gymnasium and multipurpose room units) are equipped with humidifiers to keep the ventilation air stream above 30% RH. The amount of energy required to operate the humidifiers is recorded in the simulation and shown in Fig. 3.3. No data were available to calibrate the humidifier energy consumption against reality.

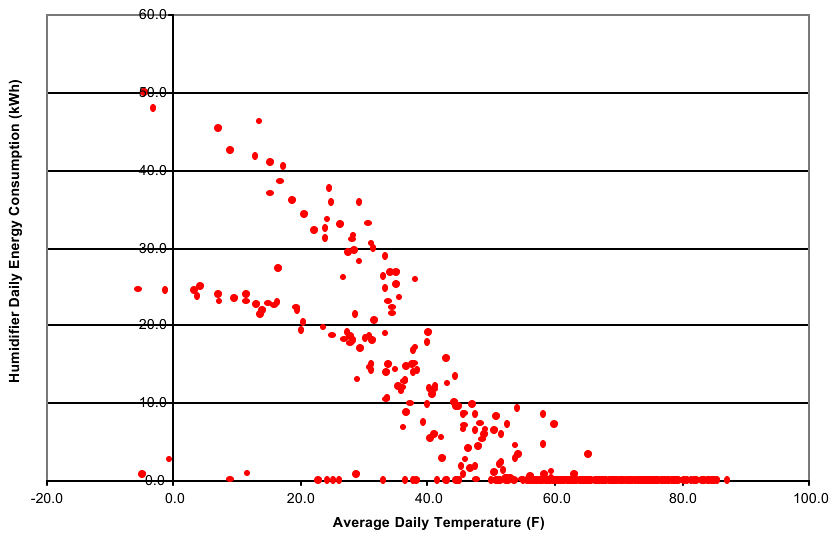


Fig. 3.3. Energy required to operate humidifiers as derived from the simulation.

3.3.2.4 Ventilation Pumps

The two large outside air heat pumps are equipped with preheat hot water coils to keep the heat pump units from locking out on safety controls. Each of these hot water coils is served by a 42-gpm, 0.5-hp pump to ensure proper flow to the coils. These pumps are controlled to operate whenever the ambient temperature falls below 40°F. The hot water for the preheating coils is provided by the gas boiler.

Because there was no data to calibrate against, the value found in the mechanical drawings was assumed for the ventilation heat pumps; thus when the pump is energized, it is assumed to draw 0.37 kW (= 0.5 hp).

3.3.2.5 Unit Heaters

The vestibules in the school, as well as several perimeter areas (e.g., the receiving area), are served by hot-water unit heaters, with hot water provided by the boiler. Because of limited gas usage data and the lack of unit heater fan energy consumption data, the unit heaters could not be calibrated against measured data. Therefore, the equipment specification values were used for the simulations.

3.3.2.6 Hot Water Pump

A 3-hp, 155-gpm pump is used to distribute hot water for preheat hot water coils and the unit heaters in the perimeter areas of the school. This pump is controlled to operate whenever the ambient temperature falls below 65°F. There were no data to verify the operation of this pump or to calibrate the simulated pump against reality. For this reason, the pump was assumed to draw 2.4 kW (= 3 hp) whenever it was operating.

3.3.2.7 Boilers

The hot water for the unit heaters and preheat hot water coils is provided by four modular boilers connected in parallel. The boilers are gas-fired and heat a mixture of propylene glycol (22%) and water to 200°F. The specifications of the boiler were an input capacity of 1600 kBtu/h with an output capacity of 1320 kBtu/h, resulting in an overall efficiency of 82.5%. Limited gas usage data prevented calibration of the boilers, so the values specified in the mechanical drawings were used.

3.3.2.8 Domestic Hot Water

The domestic hot water for the school is provided by two gas-fired water heaters. One unit serves the majority of the school areas, while the second unit serves the kitchen area. The specifications of the hot water heaters are shown in Table 3.5.

Table 3.5. Water heater specifications

School areas	Storage volume (gal)	Heating capacity (kBtu/h)
Restrooms, lounges, most of school	100.0	250.0
Kitchen	100.0	197.0

The school is equipped with a hot water distribution network designed to keep hot water available on short notice throughout the school. This is accomplished by use of two 0.33-hp 15-gpm pumps that continuously circulate hot water. To accurately model the gas consumption due to the hot water heaters, an estimate of the hot water draw was required. For the building water heater, an ASHRAE correlation was found that suggests that 0.6 gpd per building occupant is a reasonable estimate for restrooms, lounges, and similar areas. For the kitchen water heater, another ASHRAE correlation suggested that 0.7 gpd per meal served by the cafeteria is a reasonable estimate for the kitchen hot water heater. However, because Maxey School does not have a full kitchen (the meals are delivered by a service company), we halved this value for our simulations (0.35 gpd per meal served). The hot water draws for the simulation use these correlations plus the schedule of students and staff in the building to generate the hot water draw profile.

Another important factor in the hot water heater gas consumption is the heat loss in the distribution lines throughout the school. We used an ASHRAE correlation that suggests loss coefficients for different pipe sizes and types of insulation to generate effective U-values for the pipes in the school simulations. The simulated effective U-values for the school piping were0.74 Btu/hr-ft^2-ºF for the 2.5-in. piping and 0.81 Btu/hr-ft^2-ºF for the 2-in. piping.

Although no detailed gas usage data exists for the school, there is a way to check the validity of the TRNSYS hot water assumptions. For months of the year with no unit heater operation and no preheat hot water coil operation (like the summer months), the school's gas consumption is strictly due to hot water usage. During these periods in 1996, the simulated monthly gas consumption is within 2% of the utility bill data.

3.3.3 The Ground Heat Exchanger

3.3.3.1 Configuration of the BHEx

Each of the 54 heat pumps at Maxey School rejects heat to and absorbs heat from one common-loop ground heat exchanger. The common loop working fluid is a water solution containing 22% propylene glycol by volume. The heat pumps are in parallel in this installation, drawing flow from the supply loop, adding or removing heat, and then sending the flow to the return loop. A conceptual schematic of this configuration is shown in Fig. 3.4, which leaves out the piping details that balance flow across the parallel heat pumps and the parallel U-tubes in the borefield. The variable speed common loop pump is controlled to maintain a design pressure differential across the heat pumps. Two-way valves at each unit open to allow flow through the unit when the compressor is operating.

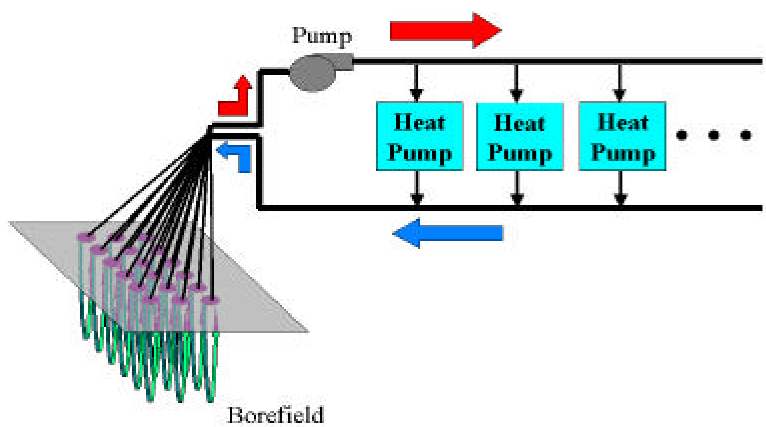


Fig. 3.4. Conceptual piping configuration at Lincoln schools.

The ground heat exchanger is made up of 120 bores in a 12 × 10 pattern (Fig. 3.5). The borefield is located several hundred feet from the school beneath a soccer field on the school grounds. The boreholes in the borefield are spaced 20 ft apart between centers and are each 240 ft deep. The boreholes are plumbed in parallel and each contains one U-tube ground heat exchanger. The U- tube ground heat exchangers are comprised of 1-in. nominal diameter SDR-11 polyethylene pipe. The boreholes themselves are each 4.25 in. in diameter for the first 220 ft and 6 in. in diameter for the top 20 ft. The bores are backfilled with fine gravel (Fig. 3.6), and the top 10 ft are sealed with a bentonite plug. The static water level was reported to be around 20 ft.

Three sets of supply and return lines lie between the school and the borefield header pit. Each of the three line pairs serves four of the twelve legs of the borefield. Originally, the borefield was configured such that a third of the borefield may be shut down in periods of little operation to allow that portion of the borefield to "recover." This option was never used during 1996 and was ignored in the simulation. The supply and return lines were designed to allow an equal distribution of flow to each of the ten ground heat exchangers on one of the legs of the 12 × 10 array. This design, shown in Fig. 3.7, utilizes sections of 2-in., 1.5-in., and 1-in. pipe in a reverse return configuration. For simulation purposes, the flow through each of the 120 borehole ground heat exchangers was assumed to be identical.

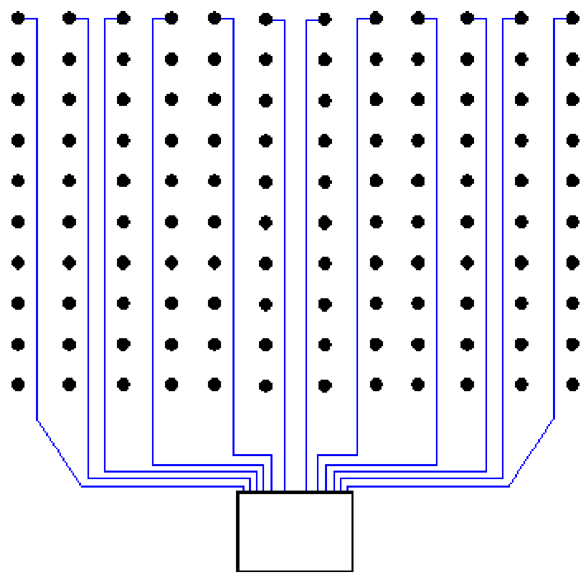
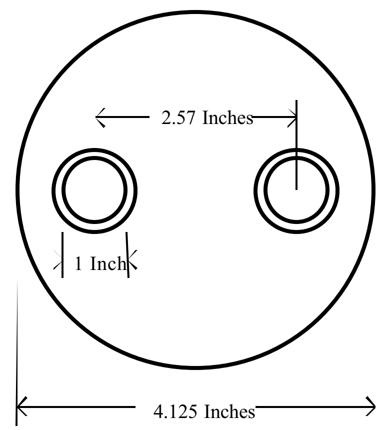


Fig. 3.5. Ground heat exchanger pattern.

Fig. 3.6. Top view of a U-tube vertical ground heat exchanger.



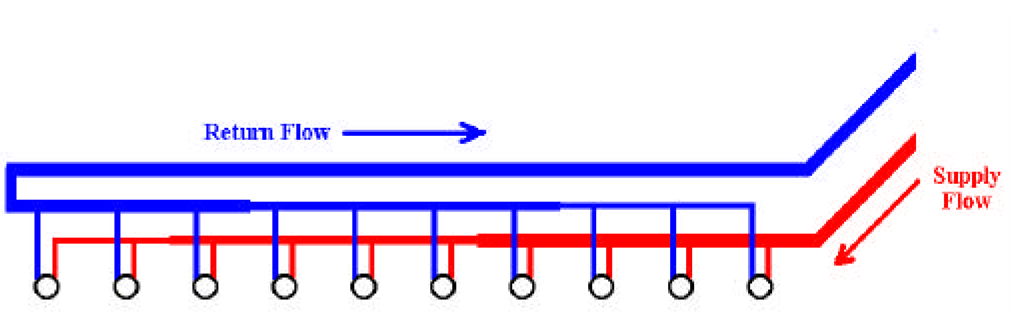


Fig. 3.7. Design of supply and return lines.

3.3.3.2 Modeling the BHEx

For vertical GHP systems, the most important component model is the BHEx. Two subroutines were available to model the borehole heat exchangers: the Superposition Borehole Model (SBM) (Eskilson 1986) and the Duct Storage Model (DST) (Hellstrom, Mazzarella, and Pahud 1996). SBM is intended to model ground heat flow in situations where a number of thermally interacting bores are present, whereas in DST multibore interaction is treated heuristically. The two subroutines were found to give essentially the same results, which indicates either that thermal interactions between the bores at the Maxey School site are minimal, or that the method DST uses to treat multibore interactions is adequate for this application. Ultimately the DST subroutine was included in the simulation because of its significantly shorter run time.

In DST, a BHEx is defined as a system in which heat or cold is stored directly in the ground. The heat transfer from the borehole system to the surrounding ground is approximated by pure conduction. The storage volume (the volume of earth containing the boreholes) has the shape of a cylinder with a vertical axis of symmetry. The boreholes are assumed to be uniformly placed within this storage volume. There is convective heat transfer in the boreholes and conductive heat transfer in the ground. It is convenient to treat the thermal process in the ground as a superposition of a global problem onto a local one. The global problem is to handle the large-scale heat flows in the storage and the surrounding ground, whereas the local problem is to account for the heat transfer between the heat carrier fluid and the storage. The local problem uses local solutions around the boreholes and a steady-flux part, by which the number of local solutions, and thus computation time, can be reduced without significant loss of accuracy. The global and the local problems are solved with the use of the explicit finite-difference method, whereas the steady-flux part is solved analytically. The total temperature at one point is obtained by a superposition of these three solutions.

The short-term effects of the injection/extraction through the boreholes are simulated with the local solutions, which depend only on a radial coordinate and consider a cylindrical volume exclusively ascribed to each borehole. As the model assumes a relatively large number of boreholes, most of the boreholes are surrounded by other boreholes. Consequently, a zero heat flux at the outer boundary attributed to the symmetrical positions of the neighboring boreholes is prescribed. This assumption may lead to the under-prediction of heat transfer to the ground in cooling mode for installations with only a few boreholes.

The heat transfer from the fluid to the ground in the immediate vicinity of the borehole is calculated with a heat transfer resistance. A steady-state heat balance (performed each time step) for the heat carrier fluid gives the temperature variation along the flow path. The local solution may take into account a radial stratification of the storage temperatures (due to a coupling in series of the boreholes), as well as increased resolution in the vertical direction. The local heat transfer resistance from the fluid to the ground (or borehole thermal resistance) may depend on the flow conditions, that is, it can be dependent on both temperature and flow. It may also take into account the unfavorable internal heat transfer between the downward and upward legs of the U-tube in a borehole.

The three-dimensional heat flow in the ground is simulated using a two-dimensional mesh with a radial and a vertical coordinate. The model assumes homogeneous and constant thermal properties within a horizontal ground layer. Several ground layers are permitted, and the thermal properties may vary from layer to layer. Insulation may be placed on the top and sides of the storage volume. A time-varying temperature is given for the ground surface.

4. CALIBRATING THE ENGINEERING MODEL

4.1 INTRODUCTION

An engineering model calibrated to site-monitored data is a useful tool because its outputs can be used for a variety of purposes:

- to investigate whether the ground heat exchangers were sized correctly;
- to provide consistent inputs for the comparison of multiple vertical ground heat exchanger sizing methods;
- to investigate the energy consequences of other HVAC design alternatives;
- to investigate alternative control strategies for the building; and
- to predict the energy consumption of the building in other climates and locations.

For Maxey School, each of the HVAC equipment components in the simulation (heat pumps, loop pumps, etc.) was calibrated to measured data (where available) so that the simulated behavior of the individual component closely matched observed operation. These calibrated components were then combined into a complete system simulation of the school and its HVAC equipment. The infiltration into the school (which directly affects the heating and cooling loads of the school) was used as the calibration parameter to match the simulated HVAC energy consumption with the measured HVAC energy consumption.

With the building and ventilation loads and ambient conditions available as forcing functions, the TRNSYS system model was run in order to estimate the power consumption of the entire HVAC system. The infiltration was then adjusted and the building loads recalculated until the simulated energy consumption matched the measured energy consumption.

4.2 MEASURED DATA

The measured site data collected at the school is critical for the calibration. The measured data taken from 1995 through 1997 varied in completeness, with 1996 being the most complete year by far. For this reason, the simulation was compared to the EMS data (measured at 10-min intervals) for the period from January 1 through December 31, 1996. Because the school opened in August 1995, one might expect a slight bias due to the cumulative net heat exchange with the ground prior to January that would affect comparisons of measured and modeled maximum and minimum simulated EWTs. (The simulation assumes that the school began operation on January 1, 1996.) It was found, however, that during the first 5 months of operation, the borefield had only modest cumulative net heat exchange; this was not large enough to significantly affect the soil thermal property calibration.

The EMS status data for the heat pumps for 1996 was reasonably complete, but the EMS energy consumption did not match that shown on the utility bills (especially for natural gas). After the EWT and flow-rate data were corrected to alleviate some problems, we used these data to calibrate the ground heat exchanger model. By far the most useful data set was the daily integrated HVAC and non-HVAC electricity consumption information from the school. Besides allowing us to calibrate our models on a daily basis, these data provided some unique clues in deciphering the actual operation of the building for 1996.

A plot of the measured daily energy consumption versus average daily temperature is shown in Fig. 4.1. This figure reveals the typical "U" shape commonly found in these types of plots, with separate curves for occupied days (in this case weekdays) and unoccupied days (weekends and some holidays). Occupied days are defined as weekdays when the building is using the standard heating and cooling setpoints of 74°F and 76°F, respectively, from 6 A.M. to 5 P.M., and heating and cooling setback/setup temperatures of 50°F and 80°F, respectively, from midnight to 6 A.M. and from 5 P.M. to midnight. Unoccupied days are defined as those days where the heating setpoint temperature is 50°F (its setback condition) and the cooling setpoint temperature is 80°F (its setup condition).

A closer look at the data, however, revealed that there were actually three curves hidden within this scatter plot. During two periods of the year (January 18–29 and November 17–December 31) the building was operated in occupied mode 24 h a day—that is, the thermostats were not set back to their normal unoccupied setpoint of 50°F. One possible explanation is that during exceptionally cold weather, it can take a long time for the building to reach the occupied setpoint in the morning. Eliminating night setback resolves the problem, but leaving the building with normal setpoints during the Christmas holidays was probably an oversight. Figure 4.2 shows the energy consumption of the building in each of its three modes.

4.3 HEAT PUMP CALIBRATION

Because the 10-min measured data did not contain enough information, it was not possible to calibrate individual heat pumps. Therefore, manufacturers' catalog data was used to simulate the performance of each of the heat pumps in the building. A new heat pump model was written for TRNSYS for this project. The operation of the model is summarized below.

1. The EWT and loop flow rate are inputs to the model.
2. The zone temperature, zone humidity, and zone loads (sensible and latent) are read from a file previously written by the building simulation.
3. The heat pump performance indicators (capacities and power) are interpolated from catalog data based on the zone conditions, EWT, flow rate, and mode of operation (heating or cooling).
4. The heat pump rejection/absorption is calculated from an energy balance using the heat pump power and the zone loads that were met.
5. The leaving water temperature and flow rate are calculated.

A plot of the simulated daily heat pump energy consumption versus the average daily temperature is shown in Fig. 4.3. Note that the three modes of operation are quite distinct on the plot, with the 24-h occupied days using significantly more electricity than the standard occupied days.
Figs. 4.4 and 4.5 are plots of the simulated hourly cooling and heating loads met by the heat pumps versus the entering water to heat pumps. These plots show that the heat pumps are sized fairly well; peak coincident operation is within the expected range. The closest the school came to its capacity constraints was 86.3% of capacity in the heating mode and 56.8% of capacity in the cooling mode.

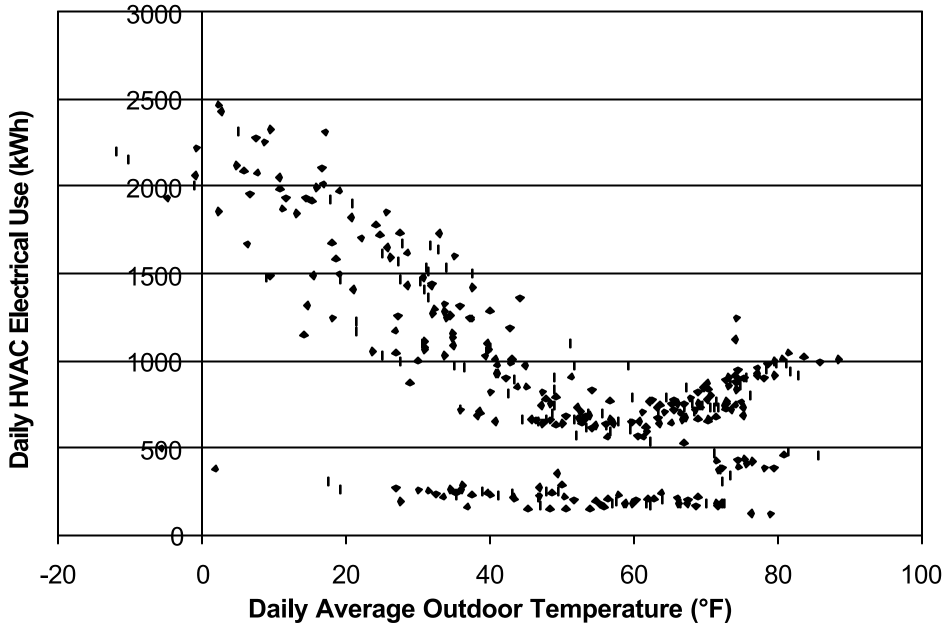


Fig. 4.1. Measured daily electric consumption vs average daily outdoor temperature.

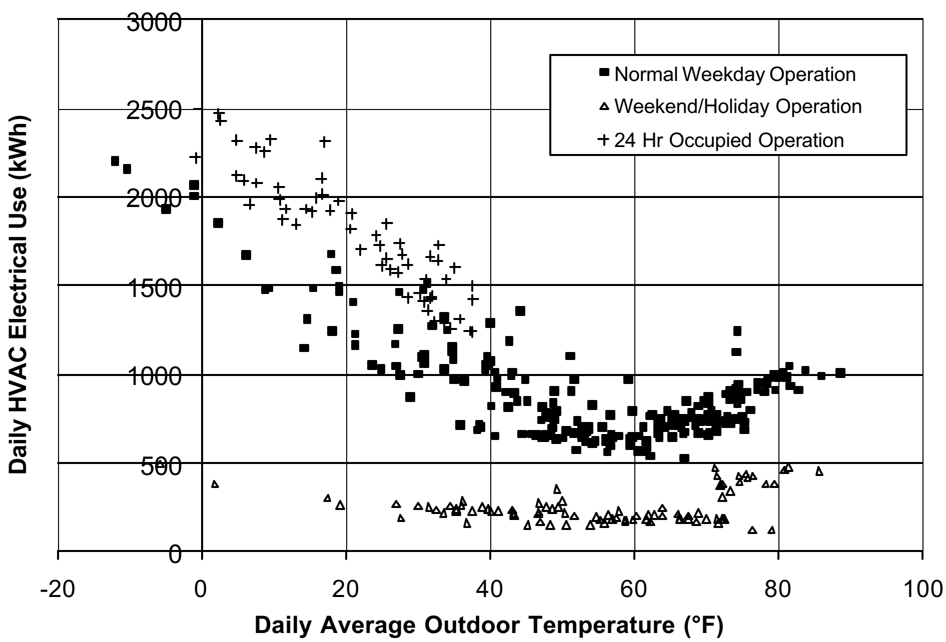


Fig. 4.2. Electrical use by HVAC system in three occupancy modes.

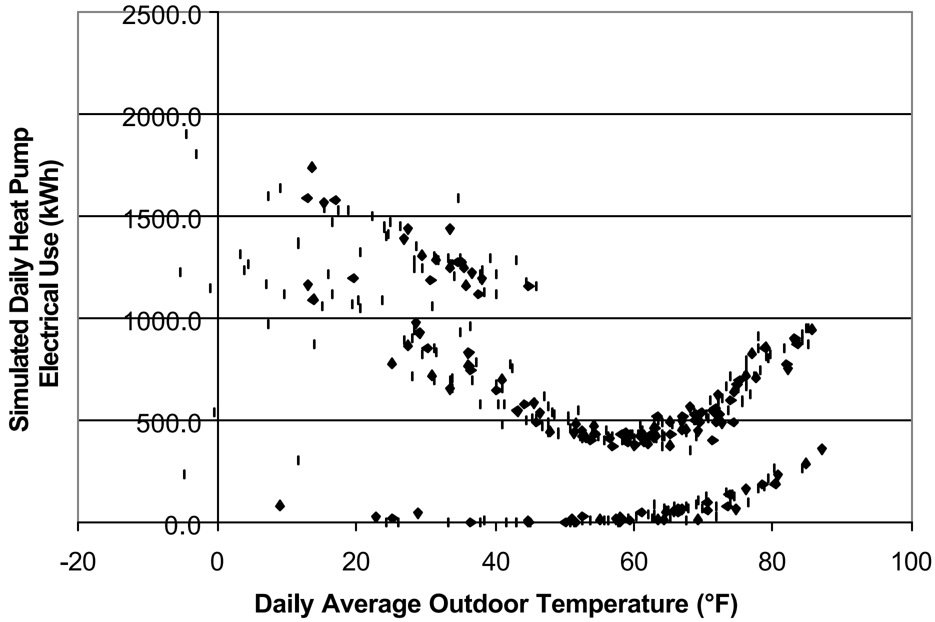


Fig. 4.3. Simulated daily electrical use by heat pumps only vs daily average outdoor temperature.

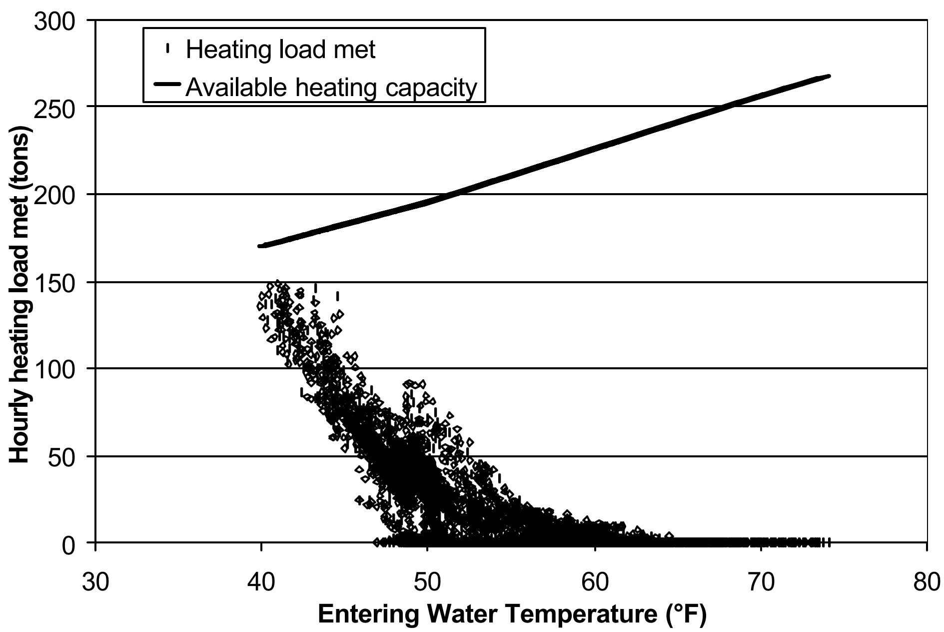


Fig. 4.4. Simulated heating load met by heat pumps vs EWT.

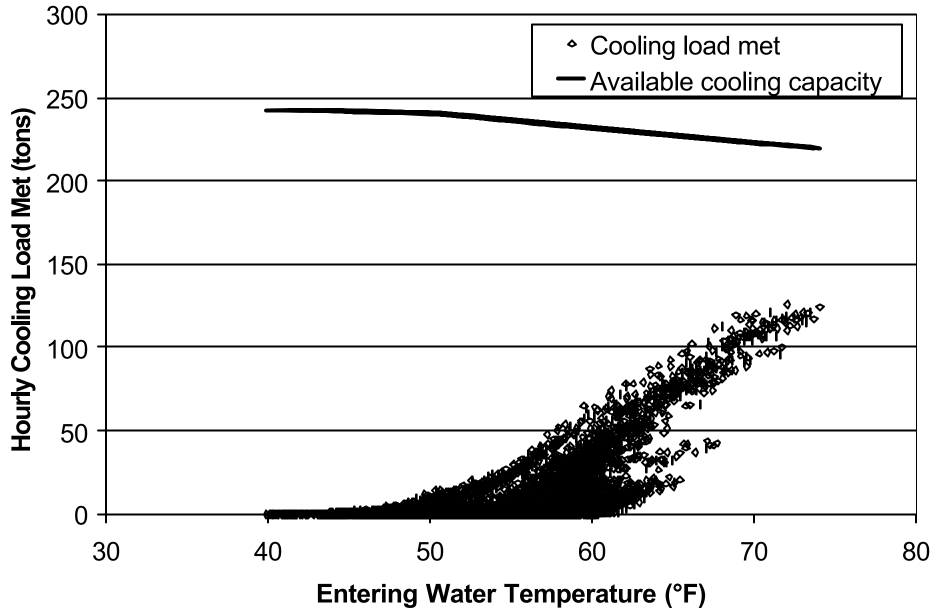


Fig. 4.5. Simulated cooling load met by heat pumps vs EWT.

4.4 SOIL CALIBRATION

Once the ground heat exchanger model was created (Sect. 3.3.3.2), the calibration of the model proceeded without problems. Where possible, we used known values of the ground heat exchanger parameters. These values included geometric data on the heat exchanger (borehole diameter and depth, header depth, borehole spacing, U-tube pipe sizes, and shank spacing) and the thermal properties of the polyethylene pipe and the bore backfill material (thermal conductivity, density, and specific heat). The detailed simulation did not include the piping runouts to the ground heat exchangers or the horizontal buried pipes between the ground heat exchangers. The horizontal runouts were investigated and found to have little effect on the predicted maximum and minimum heat pump EWTs and were therefore ignored.

The undisturbed far field earth temperature was set to a value of 55.7°F. This temperature was measured during a short-term in situ soil thermal conductivity test at the school in mid-1998 (Shonder and Beck 2000).

The remaining parameters—soil thermal conductivity, and the product of soil density times specific heat product—were varied to achieve a "best-fit" soil, the properties of which result in the best match between the recorded data for heat pump EWT versus the model predictions for heat pump EWT for the entire year. This best-fit effective soil also includes any vertical variations in soil properties and the impact of the horizontal runouts and the horizontal buried pipe between the ground heat exchangers.

The goal of this study was not to try to demonstrate how accurate the ground heat exchanger model predictions can be when the soil thermal property assumptions are best-fit to data (since this best-fit process tends to mask any errors or poor assumptions in the ground heat exchanger model), but rather, to develop an accurate prediction of the performance of this ground heat exchanger over a range of operating conditions. The “tuned” heat exchanger model can then be expected to provide reasonably accurate simulation results when coupled with other calibrated component models.

The measured loop-temperature and flow-rate data from the site were recorded in 10-min intervals. With these inputs from the monitored data, and with the soil thermal conductivity and the product of specific heat and density varied as parameters, the predicted temperature from the ground heat exchanger model in TRNSYS was compared to the outlet temperature from the ground heat exchanger (heat pump EWT) from the collected data to determine best-fit properties for the soil.

The simple statistical comparison to determine best fit was done by taking the square of the difference between the predicted values and the collected data for each 10-min interval when the loop flow rate was greater than 75 gpm. At these higher flow rates, the effect of the horizontal pipes is small. In addition, the measured temperature from the ground heat exchangers was taken inside the mechanical room (conditioned space) and was strongly affected by the conditioned space temperature at small flow rates. The differences were summed over the length of the simulation (the entire 1996 period), and the soil properties with the lowest differences were selected as the best-fit values. While other error criteria could have been used for the optimization (like the square of the error, for example), with different results, the goal was to minimize the differences in operation between the simulation and measurement over the entire year of operation (and not, for example, to reduce the effect of bad points, as a square of the error would accomplish).

The optimization of soil properties was based on the downhill simplex method in two dimensions (Press et al. 1992). The method does not require derivatives, but simply function evaluations. In this case, the function evaluations were the sum of the squared errors over the annual simulation between the measured and simulated EWT to the heat pumps. This simplex algorithm acts as an optimization routine that stands above TRNSYS and uses the simulation as a function call.

The results of the soil properties calibration for this unit correspond closely to the ASHRAE heavy saturated soil: a density-specific heat product of 44.9 Btu/ft^3-°F (ASHRAE heavy saturated soil is 40 Btu/ft^3-°F) and a thermal conductivity of 1.396 Btu/hr-ft-°F (ASHRAE heavy saturated soil is 1.40 Btu/hr-ft-°F). With the known borehole geometry and the best-fit soil properties, the DST model calculated a pipe-to-soil thermal resistance of 0.1854 hr-ft-°F/Btu.

With these effective soil properties, the average temperature difference between measured and predicted values over the year (for time intervals when the loop flow rate was greater than 75 gpm) was 0.945°F. These reverse-engineered soil properties confirm the thermal conductivity measured in an in situ test at the site (Shonder and Beck 2000). Using a cylindrical source model, the in situ test measured a “best-fit” soil thermal conductivity of 1.35 $\pm$ 0.15 Btu/hr-ft-°F, which compares well with the 1.396 Btu/hr-ft-°F value found from the annual TRNSYS optimizations. The ORNL analysis was based on a 50-h data set in which a thermal test unit was used to heat and pump water through a ground heat exchanger similar to the ones installed at the Maxey School site. The supply and return temperatures were measured and the soil thermal conductivity calculated for the test site on the basis of a cylindrical-source model. The property calibration presented here is based on all cycles of an operating system for the period of one year and includes the impact of the horizontal runouts and the horizontal buried pipe between the ground heat exchangers.

4.5 RESULTS

The goal of this phase of the project was to calibrate the models of the individual energy-consuming HVAC devices in the school to measured data and then connect the calibrated models together to form a model of the entire school. The simulated results from the school should then match the measured data taken from the school. Although some of the individual components could not be calibrated because of a lack of measured data, enough was known about the system's operation to allow an accurate estimate of the its performance.

There are really three unique criteria for evaluating the performance of the "whole system" model: the EWTs to the heat pump (used for the ground heat exchanger sizing evaluation), the total electrical consumption of the HVAC equipment, and the total gas consumption of the HVAC equipment. Of these, the most difficult to calibrate is the EWT because this depends strongly on load and system operation that is similar to that observed in actuality.

4.5.1 Electrical Energy Use

The electricity consumed by the HVAC equipment at the school was recorded on a daily basis. Electric consumption data was available for the heat pumps, the loop pump, ventilation hot water pumps, the hot water pump, unit heater fans, humidifiers, and domestic hot water pumps. The electricity consumed by the same pieces of equipment in the simulation was integrated on a daily basis so that it could be compared against the measured data. The simulated and measured daily HVAC electric consumption as a function of the average daily ambient temperature is shown in Fig. 4.6. It can be seen that the simulation does an excellent job of predicting the performance in all three modes of operation—occupied days, unoccupied days, and days when the controller was set to constant occupied mode.

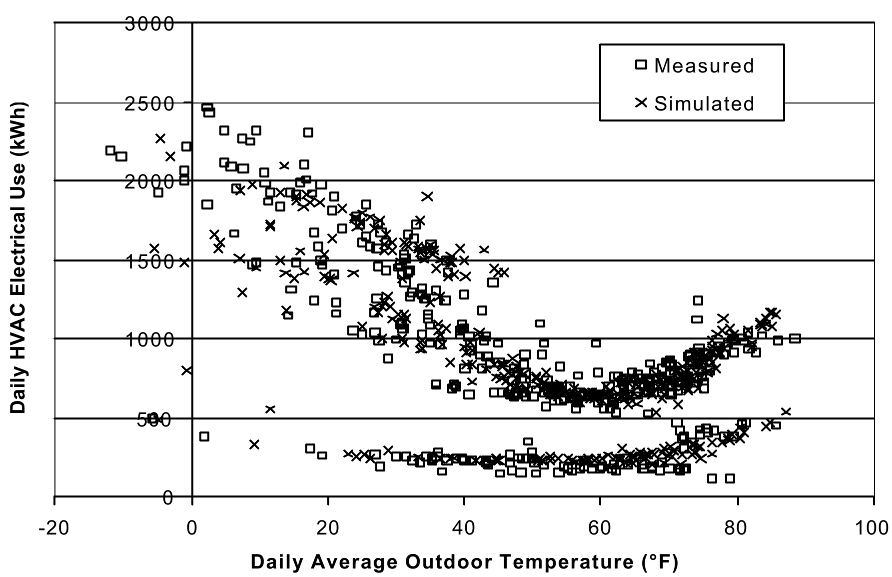


Fig. 4.6. Simulated and measured daily HVAC electric use vs daily average outdoor temperature.

Not only does the simulated electric consumption closely match the measured consumption, but the annual totals closely match as well. The measured daily data were also fit to a TMY using a five-parameter estimation technique, and this result is included as well. Fitting the measured data to a TMY allows an even comparison of the measured data to the simulated data, at least in terms of weather. The result is that the TRNSYS simulation, when calibrated to measured data, was able to predict the annual electric consumption of Maxey School in a TMY to within 1% of the measured consumption (Table 4.1).

Table 4.1. Performance of calibrated model in simulating HVAC electric consumption

Data set	Annual HVAC consumption (kWh)
Measured 1996 data	323,232
Measured data fit to TMY	311,372
Simulated typical year	314,901

An energy consumption breakdown by device is shown in Fig. 4.7. As was expected, the heat pump system consumes most of the electrical HVAC energy at the school (72% for the heat pumps and 22% for the loop pump).

4.5.2 Electrical Demand

During 1996, the measured peak electrical demand at Maxey School was 255 kW. For a TMY, the TRNSYS simulation predicts a peak demand of 286 kW, which is about 12% higher than the measured value. As is shown in Fig. 4.8, the simulated load duration curve does not match well with the load duration curve based on actual data. One obvious reason for this is that the TRNSYS simulation used typical year weather, rather than the actual weather experienced during the year. Another reason for the difference is that in the TRNSYS simulation, lighting and equipment loads are assumed to be constant during occupied periods. (A different constant load is assumed for unoccupied periods.) In reality, lights and equipment are turned on and off throughout the school day according to the needs of teachers and students. Thus, when total electrical demand is considered, the TRNSYS simulation appears to have less load diversity during occupied periods. However, as is shown in Fig. 4.9, when only the demand of the HVAC equipment is considered, there is excellent agreement between the actual and simulated load duration curves.

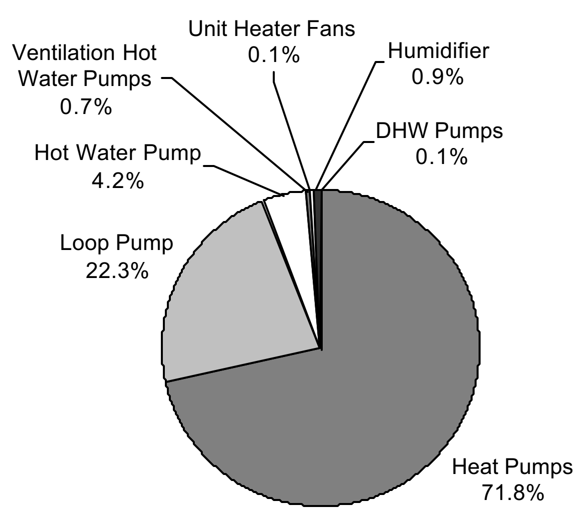


Fig. 4.7. Annual electrical energy use by major HVAC devices.

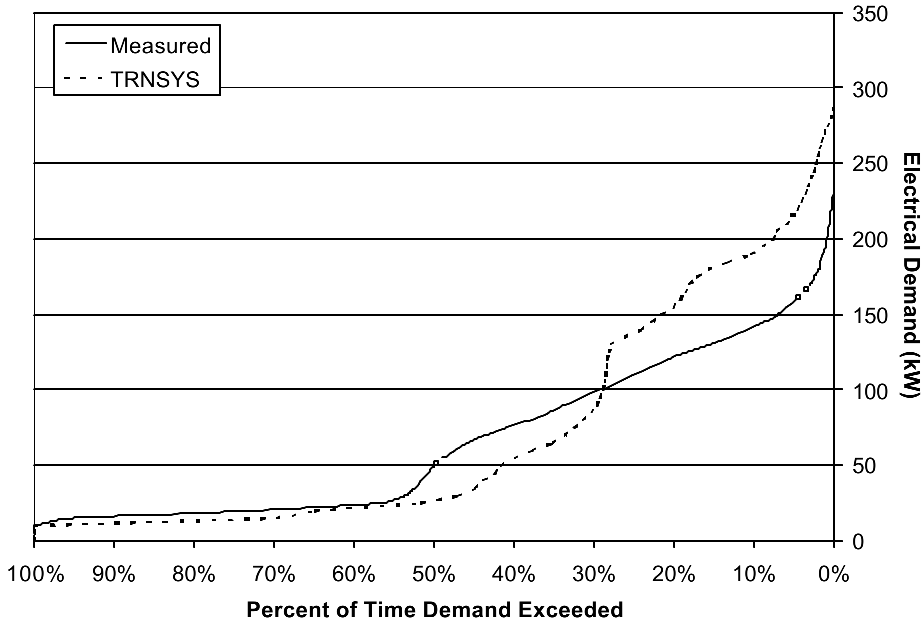


Fig. 4.8. Actual 1996 and TRNSYS-simulated load duration curves for total electrical demand (lights, equipment, and HVAC).

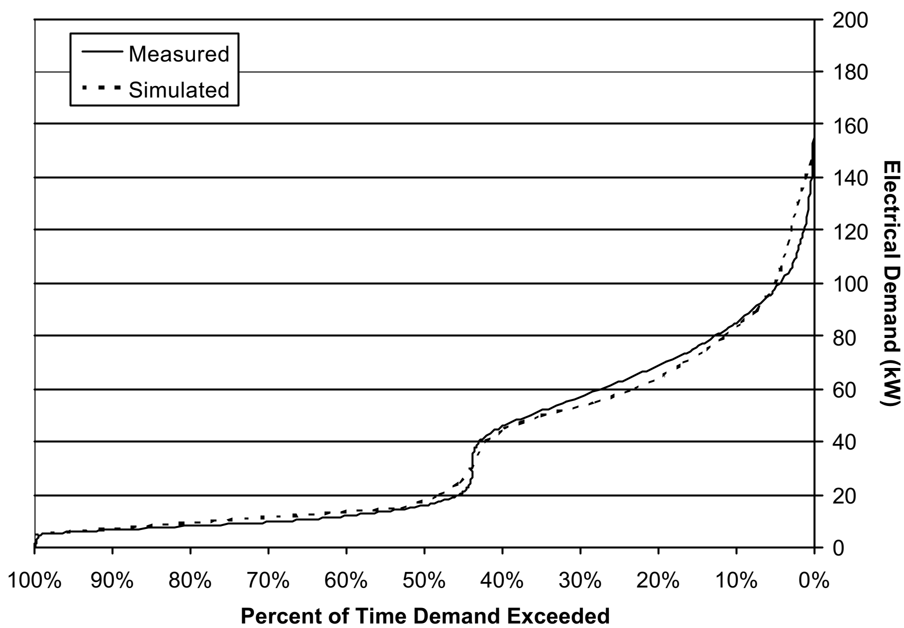


Fig. 4.9. Actual 1996 and TRNSYS-simulated load duration curves for HVAC electrical demand only.

4.5.3 Natural Gas Use

The only data available for natural gas use were for monthly consumption. Figure 4.10 presents a comparison of actual 1996 monthly natural gas use versus heating degree days in the billing period, and predicted natural gas use for a TMY versus heating degree days. The model predicts a gas use of 12,424 therms (364.2 MWh) in a TMY. For comparison, a correlation of the 1996 data versus heating degree days per month predicts that the school would use 12,787 therms in a TMY. The two values agree to within 3%.

4.5.4 Entering Water Temperatures

A comparison of the predicted and measured EWTs for a mild weather week during February is shown in Fig. 4.11. As can be seen from the figure, the model does an excellent job of predicting reality when calibrated. The periods of low-flow operation when the predicted values diverge from the measured values can be easily explained. The simulated EWT is "measured" at the outlet of the ground heat exchanger (in the earth). The measured EWT from the site was actually measured in the conditioned mechanical room of the school. At low flow rates, the conditioned space temperature affects this measurement. Although not shown, during periods of no-flow operation, the measured EWT decays to the conditioned space temperature, not to the soil temperature.

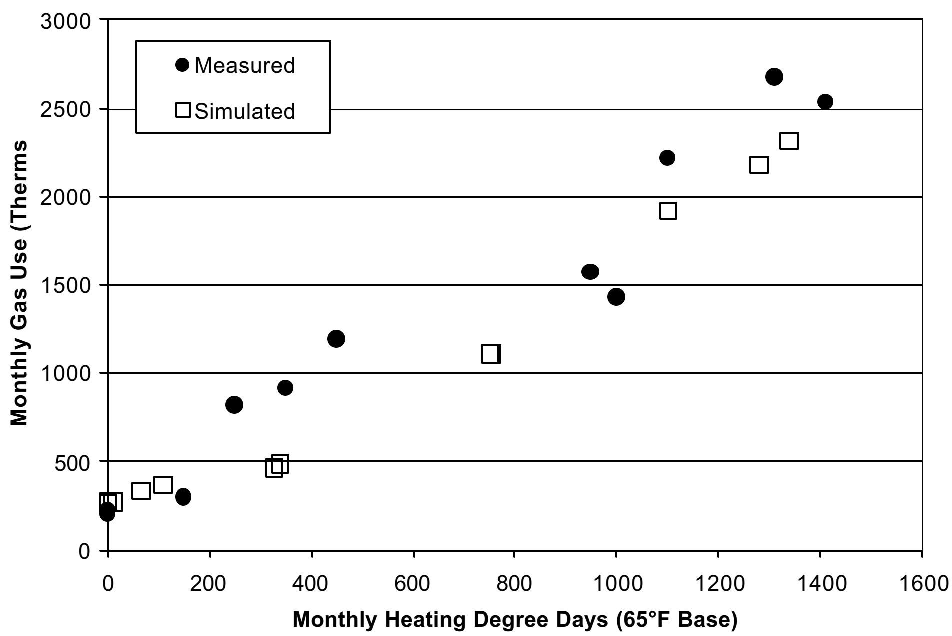


Fig. 4.10. Actual 1996 and TRNSYS-simulated monthly natural gas use vs base-65°F heating degree days per month.

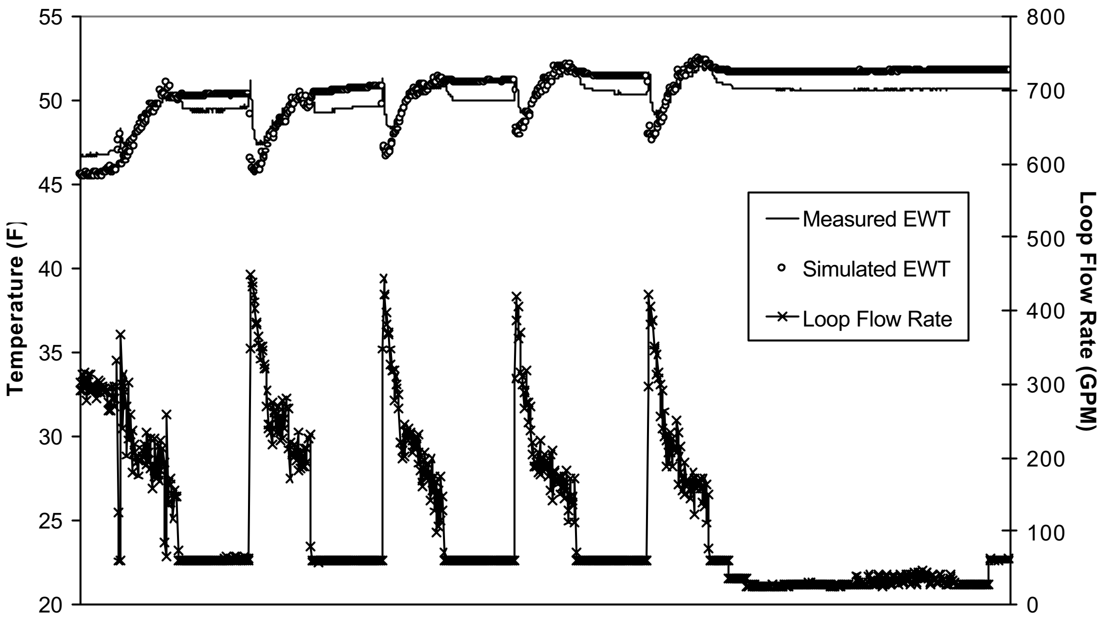


Fig. 4.11. Predicted and measured EWTs for a mild winter week.

The minimum and maximum EWTs for the simulated year and for the measured year are shown in Table 4.2 and Figs. 4.12–4.13. While there are differences, which can be attributed primarily to weather, the figures generally show good agreement. The maximum measured EWTs reported are for times when the pump flow rate was greater than 75 gpm. At lower flow rates, the effect of the measurement being made inside the mechanical room masks the true extremes. The simulated year has fewer heating degree days (6241, compared with 6924 for the actual year) but has a much colder minimum ambient temperature (–4.4 to –8.5°F). The simulated year also has a slightly higher maximum ambient temperature (98.1 to 97.0°F) and a higher number of cooling degree days (1215 compared with 1069 for the actual year).

Table 4.2. Minimum and maximum EWTs (°F) for simulated and measured year

Data set	Minimum EWT	Maximum EWT
Simulation results	39.9	74.2
Measured results	45.2	68.9

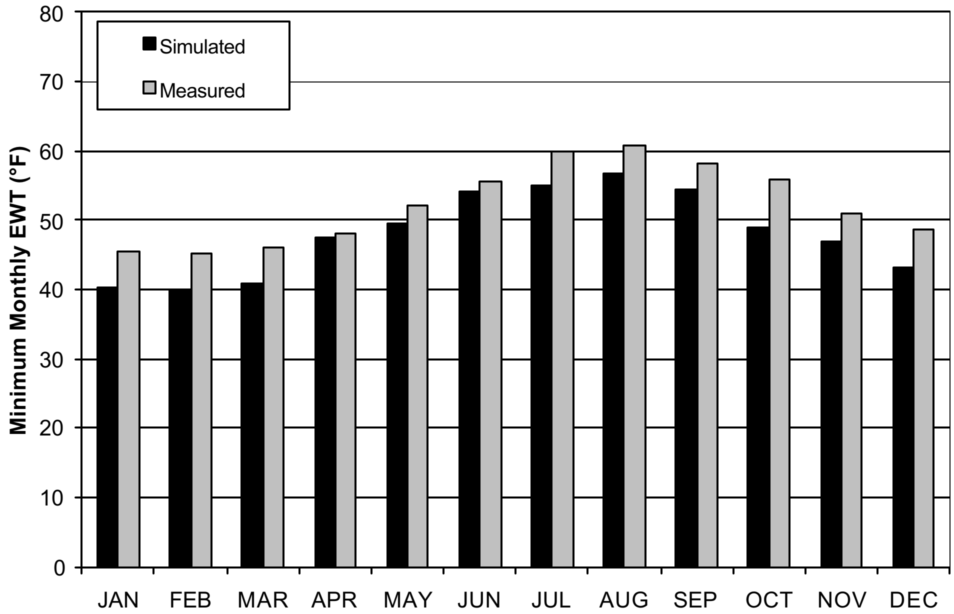


Fig. 4.12. Minimum EWTs by month for simulated and measured year.

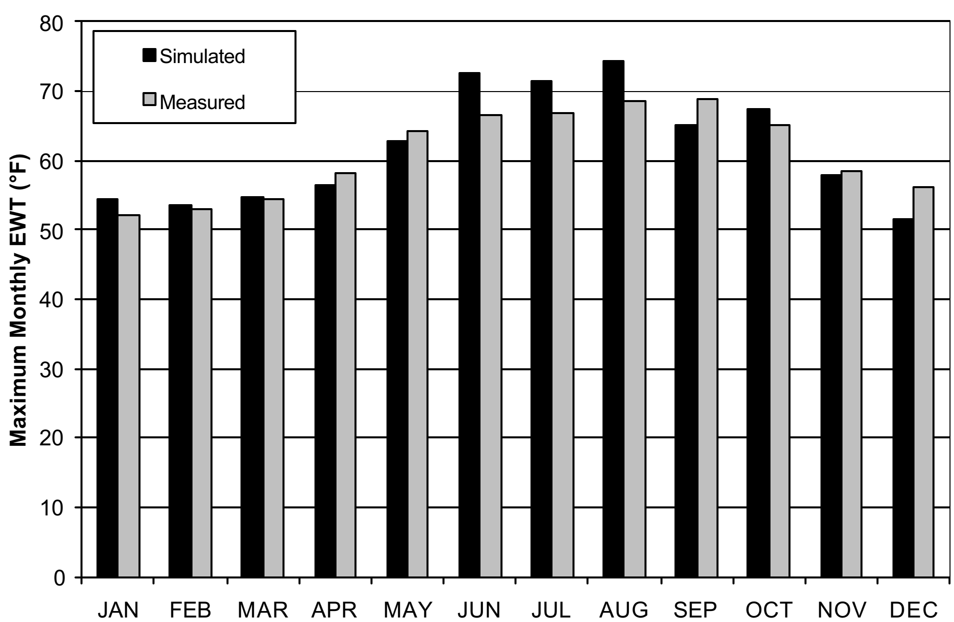


Fig. 4.13. Maximum EWTs by month for simulated and measured year.

4.6 CONCLUSIONS

The final results for the simulated and measured energy consumption for Maxey School show excellent agreement. According to the monitored data, during calendar year 1996 the school's HVAC system used a total of 323,232 kWh of electrical energy. A correlation of daily energy use versus daily average temperature indicates that in a TMY, the HVAC system would have used 311,372 kWh. When run with TMY weather data, the calibrated model predicts HVAC electrical use of 314,901 kWh. The two values agree to within 1%. Actual natural gas use, when fit to a TMY, agrees with the model to within about 3%

Another measure of the accuracy of the calibrated simulation is how well it is able to predict annual minimum and maximum EWTs. Unfortunately, not enough 1996 weather data was available to drive the calibrated simulation, so it is necessary to compare the model's predictions for a TMY with actual data from 1996. The comparison is presented in Figs. 4.12 and 4.13. Because the cooling season for 1996 was less severe than the TMY (1069 base-65°F cooling degree days in 1996 versus 1215 for the TMY), and the 1996 heating season was more severe than the TMY (6924 base-65°F heating degree days in 1996 versus 6241 for the TMY) the measured and simulated EWTs are not directly comparable. Nevertheless, the two appear to agree within about 5°F.

While the completeness of the 1996 data made it useful for calibration purposes, the set did have some problems. In Fig. 4.2, the daily electrical energy use by the HVAC system is plotted against daily average temperature for every day in 1996. Three modes of energy use are evident: normal weekday operation when school is in session (filled squares), weekend/holiday operation (open triangles), and an 8-week winter period during which the HVAC system used much more energy than normal. As shown in Fig. 4.6, we were able to model this behavior by eliminating night setback during the 8-week period and heating the school at its daytime setpoint 24 h per day. Apparently the control system was inadvertently set this way for the period in question. In order to generate inputs for the design methods, the controller was assumed to operate with the normal night setback.

Part II. A Comparison of Design Algorithms for Borefield Sizing

5. VERTICAL GROUND HEAT EXCHANGER DESIGN SOFTWARE FOR COMMERCIAL APPLICATIONS

5.1 INTRODUCTION

In previous studies, we have compared a number of commercially available programs for the design of vertical borehole heat exchangers (BHEx) in residential applications (Thornton, McDowell, and Hughes 1997; Shonder et al. 1999). This chapter extends the comparison to commercial applications by comparing four BHEx design programs and a benchmark simulation for a commercial application. In the previous chapter we described the development and calibration of an engineering model for Maxey Elementary School. The school's operation was then simulated for TMY at the site. The outputs from this simulation were used as inputs to the four design programs described here. Since loads at the school are dominated by heating, the programs were exercised to design borefields with minimum inlet water temperatures of 30, 35, and 40°F.

Calibrated models are necessary for these comparisons because all of the design programs use different algorithms to size the BHEx. The programs call for different inputs (e.g., monthly heat absorption/rejection to the ground, peak heating and cooling loads per month, equivalent full-load heating and cooling hours, and others), and there is no consistent way to derive all of these inputs from a given monitored data set. Even if it were possible to develop inputs for all of the methods from site-collected data, those data represent the behavior of the system only for the period during which the data was collected. BHEx design algorithms require load information for a typical year at the site. Without a calibrated model, there is no generally accepted method of predicting typical year performance using data from an actual year.

5.2 GROUND HEAT EXCHANGER SIZING PROGRAMS

As in the previous residential comparisons, the sizing programs are referred to by letter designation. In this case, only those programs suitable for commercial borefield sizing (A, B, C, and F) were evaluated.

Each of the four programs requires a different set of user inputs. The general factors which influence the design size of the ground heat exchanger (GHX) are the building design loads, monthly and annual heating and cooling loads, soil thermal and temperature properties, heat exchanger geometry and pipe thermal properties, and the capacities and efficiencies of the heat pumps connected to the ground loop. The inputs used and the method of deriving these inputs from the detailed simulation model are presented below.

5.2.1 Program A

Program A allows the user to select a horizontal or vertical heat exchanger configuration from a set of standard arrangements or a rectangular borefield of any dimension. We stipulated a 120-bore GHX with a 10 × 12 configuration and 20-ft bore-to-bore spacing to match the geometry of the actual Maxey borefield. Program A also requires the distance between the U-tubes, the U-tube pipe material and nominal diameter, the distance from the surface to the top of the U-tube, and specification of the fill material (grout). Values corresponding to those of the actual site were used for all of these inputs except for the grout material. The thermal conductivity of the fill material used in the detailed simulation was 1.0 Btu/hr-ft-°F. From the menu of choices for grout material available in Program A, we chose a 64% solids thermal grout as the closest match to the actual material. This choice was based on data in Smith and Perry (1999), who reported a thermal conductivity value for a 63.5% solids grout of 0.85 Btu/hr-ft-°F. Soil type, soil thermal properties, and ground temperature data are selected from menus. A new soil with the "best fit" properties of the Maxey site was added to the Program A menu and used. A user-defined ground temperature data set with Maxey site data was added to the built-in menu and used.

Program A also gives users the option of selecting a heat transfer fluid from a built-in menu or adding their own. We added to the menu a new fluid with the thermal properties of the actual fluid used at the site. Maximum overall design flow rate must be input along with total heating and cooling capacity (at ARI-330 standard rating conditions) and average heating coefficient of performance (COP) and cooling energy efficiency rating (EER) (at design EWTs) of the building heat pumps. Values from the detailed simulation were available for the system flow (460 gpm), for the average heat pump efficiency and capacity at the simulation minimum and maximum EWTs (40°F and 74.1°F, respectively), and for the average capacity at standard rating conditions (EWTs of 32 and 77°F). These are presented in Table 5.1. For other EWTs used in the sizing cases, heat pump efficiency was adjusted from the detailed simulation values based on data for the most prevalent heat pump unit used at Maxey. Rated performance data for this heat pump are presented in Table 5.2.

Table 5.1. Average seasonal performance data from detailed simulation

Heating values:	
Heating seasonal COP	4.00
Minimum EWT	40°F
Ave. rated heating capacity at minimum EWT	2039 kBtu/h
Ave. rated heating capacity at 32°F	1788 kBtu/h
Cooling values:	
Cooling seasonal COP	4.62
Maximum EWT	74.1°F
Ave. rated cooling capacity at maximum EWT	2635 kBtu/h
Ave. rated cooling capacity at 77°F	2610 kBtu/h
Maximum system flow rate	469 gpm

The only other input required for Program A was either monthly ground heat absorption and heat rejection or monthly building loads. Values from the detailed simulation for both options are listed in Table 5.3.

Program A offers two methods for computing design lengths: an "average monthly load" basis and a "peak load" basis. The peak option requires two additional inputs: winter and summer peak run time ratios. The user's manual for Program A defines these simply as the run time ratios during peak conditions. From the detailed simulation maximum and minimum EWTs, the run time ratios for the peak hourly loads were determined to be 1.00 for heating and 0.58 for cooling based on the standard rated capacities. In discussions for clarification, the program developers indicated that their intent is that run time ratios should be the average values for the peak two-day

Table 5.2. Capacity and efficiency data for most prevalent heat pump model used at Maxey School site

EWT (^{o}F)	COP (heating) or EER (cooling)	Capacity (kBtu/h)
	Heating	
25.0	3.46	28.2
30.0	3.67	30.8
35.0	3.86	33.4
40.0	4.09	36.1
	Cooling	
74.1	16.07	45.1
90.0	13.60	42.9
95.0	12.43	40.5
100.0	11.36	38.2

Note: Values for fluid flow rate of 11 gpm and air flow rate of 1450 cfm.

Table 5.3. Monthly total and peak loads and heat absorption and rejection for Maxey School, simulated for TMY

Month	Total heating (kBtu)	Total cooling (kBtu)	Peak heating (kBtu/h)	Peak heating hours	Peak cooling (kBtu/h)	Peak cooling hours	Heat absorbed (kBtu)	Heat rejected (kBtu)
Jan	343,444	3,895	1,786	11	152	2	253,894	10
Feb	240,506	1,495	1,696	5	59	1	178,038	41
Mar	145,415	11,024	1,693	2	301	1	100,429	4,480
Apr	70,820	25,348	905	3	396	2	41,360	17,517
May	26,609	92,481	490	1	941	2	6,111	95,376
Jun	6,386	170,592	227	1	1,508	3	96	199,240
Jul	5,323	227,177	210	1	1,432	5	3	266,829
Aug	4,807	235,520	154	1	1,491	7	34	276,723
Sep	20,600	88,148	287	6	863	2	3,555	92,602
Oct	58,755	43,803	1,081	1	1,101	3	30,595	37,860
Nov	203,565	7,045	1,098	2	217	2	147,112	645
Dec	379,297	3,962	1,274	4	104	2	282,797	0

periods (again based on the standard rated capacity of the heat pumps). Peak two-day average run time ratios were determined to be 0.405 and 0.25 for heating and cooling, respectively, from the detailed simulation.

Sizing runs were made using both methods and with both peak-hour and average two-day run time ratios for the peak method. Using the peak-hour run time ratios, the peak load option produced designs more than twice as long as those produced by the average monthly load method. Using the average two-day peak run time ratios, peak method designs were about 30% longer than those of the monthly method. The peak load method, using two-day average run time ratio values, is the most consistent with the other three design programs. Therefore, this report presents designs from that method only. It is evident that careful determination of the run time ratio input values is critical to achieving accurate loop designs with Program A's peak option because its results are extremely sensitive to this parameter.

5.2.2 Program B

Program B requires the same basic design parameters as Program A. In addition, it requires the user to input a value for borehole resistance (fluid-to-ground heat transfer resistance). In the absence of a value from the detailed simulation, a utility program included with Program B was used to compute a value of 0.211 hr-ft-°F/Btu. Operating data for the most prevalent heat pump used at Maxey are included in the Program B database. This was used for the required heat pump input.

One drawback to Program B is that it is limited to a maximum borefield size of 10 × 10. This meant that the size of the actual Maxey borefield could not be input directly. We used the 10 × 10 grid but adjusted the loads and total flow rate as discussed below.

Program B also requires a bore spacing-to-depth ratio. Although the actual value for Maxey is 0.083 (20/240), Program B only allows the user to choose from a built-in library of discrete values. The closest value available to the actual was 0.10, and this was used.

For each month in a design year the program requires total heating and cooling load and peak heating and cooling loads. In order to account for the borefield size limitation noted above, we adjusted values obtained from the detailed simulation by a factor of 0.833 (100/120), as recommended by the developers of Program B. In addition, the program requires the number of hours of occurrence of peak load in any one month. A default of 6 h is recommended by the developers. From the detailed simulation, maximum hours of occurrence for heating and cooling peaks were determined to be 11 and 7, respectively (see the discussion under Program F). These values were input to Program B.

A number of heat transfer fluid choices are built into Program B. The 23.5% propylene glycol option was chosen as the closest match to the actual fluid used (22% propylene glycol by volume). The maximum fluid flow from the detailed simulation, 460 gpm, was adjusted by the 0.833 factor to account for the smaller effective borefield.

5.2.3 Program C

Program C requires basic information about the ground heat exchanger array: the nominal diameter of the U-tube pipe, the thermal conductivity of the fill material, heat transfer fluid flow rates (in gpm/ton), bore separation distance, and borefield dimensions. The program also requires the user to specify whether the fluid flow at design conditions is turbulent, transitional, or laminar. The flow was determined to be turbulent for the Maxey case. Given this information, the program calculates a borehole resistance of 0.196 hr-ft°F/Btu.

Operating data for the Maxey heat pumps were available in the program's database. We used data for the most prevalent size unit as input. Best fit soil properties from the detailed simulation were used.

Rather than the monthly loads required by the other design programs, Program C requires the average loads in each of four time periods (blocks) for a heating and a cooling design day. Design days were determined from the detailed simulation as those days during which daily heating and cooling loads were a maximum. These block loads are given in Table 5.4. The program also requires annual equivalent full-load heating and cooling hours. These were determined by taking the annual loads from the detailed simulation and dividing them by the average rated capacity of the system heat pumps at the simulation maximum and minimum EWTs (see Table 5.1 for values). This calculation yielded 738.3 full-load heating hours and 345.5 full-load cooling hours. (For comparison, in the simulation model the average run-time for all the heat pumps was 537 hours in heating mode and 397 hours in cooling mode.) It should be noted that full-load hours will vary somewhat depending on the EWT used to determine heat pump capacity. Fortunately, bore length results from Program C are not very sensitive to this parameter. Increasing the cooling load hours by 18% resulted in a change in calculated bore length of less than 1%.

Table 5.4. Average block loads (kBtu/h) on peak heating and cooling days

Block	Ave. heating load	Ave. cooling load
8 A.M. – noon	1671	1322
Noon – 4 P.M.	1305	1441
4 P.M. – 8 P.M.	113	283
8 P.M. – 8 A.M.	547	286

5.2.4 Program F

With the exceptions of the block design loads (Table 5.4), full-load hours, heat pump capacity, and monthly ground loads, Program F requires all of the information needed by the other three programs. The heat pump information required by Program F consists of an estimate of the heating and cooling seasonal performance factors. For our purposes, these were determined from the detailed simulation and are presented in Table 5.1. For EWTs other than the simulation minimum and maximum, these values were adjusted in the same manner as discussed under Program A. If it is desired to consider peak load periods in the design analysis, then the hours of occurrence of peak heating and cooling loads must be provided for each month of the design year. For purposes of the present study, this information was determined from the detailed simulation by taking the number of hours where the heating or cooling load was within 95% of the absolute hourly peak. Peak hours of occurrence are included in Table 5.3.

From the fluid, the U-tube material, and the fill material information given, Program F calculated a borehole resistance of 0.177 hr-ft-°F/Btu.

The program gives the user the option of including or not including monthly peak loads in the sizing analysis. We ran the program using both approaches. Design bore lengths obtained when ignoring peak effects were about 30 to 40% as long as those derived when the peak loads were included. Since the "peak load" design lengths were the most consistent with those of the other three design models, only those design lengths are reported here.

5.3 COMPARISON OF RESULTS FROM THE FOUR DESIGN PROGRAMS

Table 5.5 compares the results of one-year heat exchanger designs from the four programs and the calibrated benchmark. In all cases, the system was determined to be dominated by the heating load; thus, designs were produced to limit the minimum EWTs to 30, 35, and 40°F. These are the heat exchanger lengths required such that the minimum EWT does not fall below the given temperature in the first year of operation. One-year design lengths are most appropriate for applications where heat rejection and extraction roughly balance over the year, but are sometimes used for commercial sizing when the borefield has modest multiyear effects. The lengths are plotted in Fig. 5.1. On average, there is a difference of ±16% between the designs from the four programs and the TRNSYS benchmark design. Note, however, that Programs A, B, and C agree more closely with the TRNSYS benchmark than does Program F. On average, there is a difference of ±12% between the designs of Programs A, B and C and the benchmark, while the designs of Program F differ by ±25%, on average, from the benchmark.

Table 5.5. One-year design lengths in bore ft/ton from the four design programs and TRNSYS

Min.	Design program				
EWT	A	B	C	F	TRNSYS
30°F	73.5	75.3	91.8	102.9	78.8
35°F	91.2	92.9	115.3	129.4	103.5
40°F	118.8	120.6	152.4	170	142.9

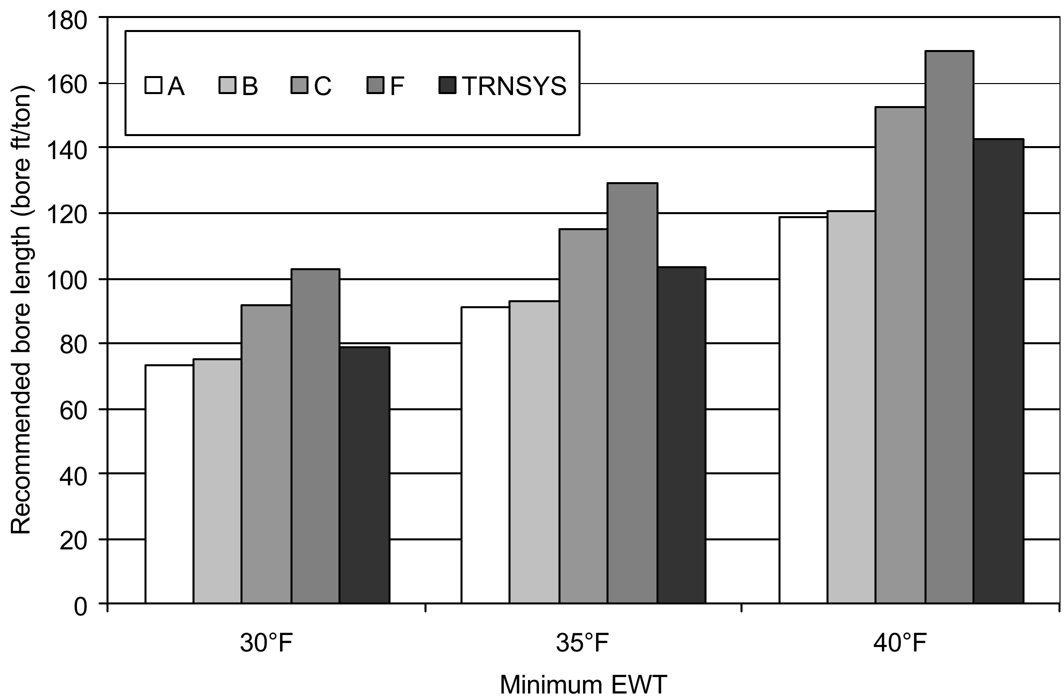


Fig. 5.1. One-year heat exchanger lengths for various minimum EWTs from the four design programs and the TRNSYS benchmark.

The 10-year heat exchanger design lengths are presented in Table 5.6 and plotted in Fig. 5.2. On average, these lengths are about 7% higher than the one-year lengths, indicating only modest multiyear effects. Overall, the heat exchanger lengths differ by an average of ±12% from the TRNSYS benchmark, somewhat less than the ±16% difference for the one-year lengths. As with the one-year lengths, the designs of Programs A, B, and C agree more closely with the benchmark than do the designs of Program F. On average, there is a difference of ±10% between the designs of Programs A, B, and C and the benchmark, while the designs of Program F differ by ±17%, on average, from the benchmark.

Table 5.6. Ten-year design lengths in bore ft/ton from the four design programs and TRNSYS

Min	Design program				
EWT	A	B	C	F	TRNYS
30°F	85.3	77.6	97.6	100.6	84.1
35°F	105.9	97.1	121.8	126.5	109.4
40°F	137.1	127.1	160	168.8	148.2

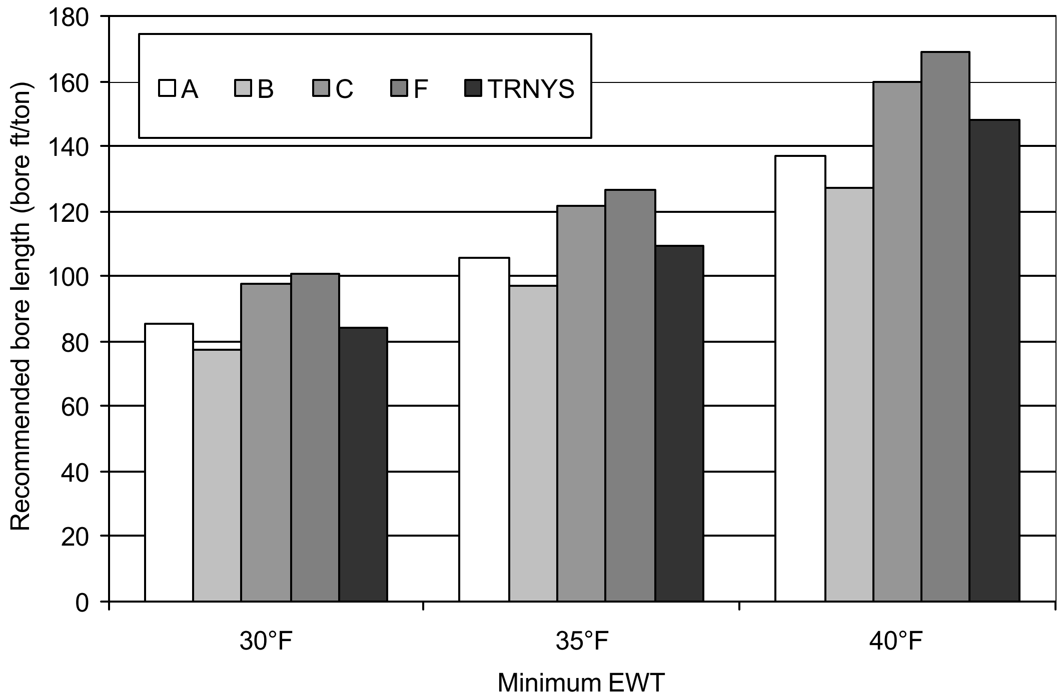


Fig. 5.2. Ten-year heat exchanger lengths for various minimum EWTs from the four design programs and the TRNSYS benchmark.

With 120 boreholes drilled to a depth of 240 ft, the borefield actually installed at Maxey School is sized at approximately 192 bore ft/ton. While this is much larger than the size recommended by any of the four design programs tested, recall that the original design was based on year-round operation rather than the actual 9-month operating schedule used to develop this comparison. As noted in Section 2.4, operating the school on a 12-month schedule changes the annual heat balance from the existing slightly heating-dominated condition to a highly cooling-dominated one. Thus, the designs produced by the four programs in this section cannot be directly compared with the actual installation at the school. Nevertheless, when the 10-year TRNSYS benchmarks are extrapolated to 45°F (the minimum entering water temperature seen in the 1996 data), they indicate that a borefield of about 200 bore ft/ton would result in this annual minimum. This is only 4% larger than the installed borefield. Although the two numbers are not directly comparable due to weather differences between 1996 and the TMY for the site, the result does provide some confidence. Thus, in a qualitative sense it can be said that the four design methods do match the existing borefield, to the extent that their recommended designs match the TRNSYS benchmarks.

5.4 CONCLUSIONS

Three of the programs tested—Programs A, B, and C—agreed with the one-year benchmark lengths to within about ±12%, which is comparable to the accuracy seen in the most recent comparison of designs for residential systems. Program F differed by ±25% on average from the one-year benchmark designs. These results indicate that the publishers of this program may need to reexamine the method used to calculate the design lengths.

Part III. Comparing Life Cycle Costs for Commercial Heat Pump Systems

6. MAINTENANCE COSTS

6.1 INTRODUCTION

In conjunction with the study of the energy consumption of all of Lincoln's schools, we reviewed their maintenance request database to learn more about actual maintenance costs for GHPs and to compare these with costs for more conventional HVAC systems found in the district's schools.

Maintenance costs for commercial HVAC systems, including costs for buildings with GHP systems, have been examined in some previous studies. A recent study of the annual maintenance costs for 25 buildings with GHP systems by Cane, Morrison, and Ireland (1998a,b) focused on maintenance activities considered to be either responses to failures (repair or service) or part of a planned maintenance program (preventive and corrective). The sample included 15 schools, 3 offices, 4 multifamily homes, 2 warehouses, and 1 restaurant. Average annual costs ranged from 9.3¢/ft^2 for in-house labor to 10.95¢/ft^2 for contracted work. For schools, average annual maintenance costs ranged from 4.69¢/ft^2 for in-house work to 6.97¢/ft^2 for contracted labor. The age of these schools ranged from 3 to 17 years, with an average age of 6.2 years.

An older study of conventional HVAC systems commissioned by ASHRAE (Dohrmann and Alereza 1986) is the basis for the maintenance costs listed in the 1995 *HVAC Applications Handbook* (ASHRAE 1995). This study covered a sample of 342 commercial buildings located across the United States, with ages ranging from 2 to over 25 years. GHP systems were not included because they were not commonly available at the time. Average annual maintenance costs for the entire sample were 32¢/ft^2, with a median cost of 24¢/ft^2.

Using maintenance records and procedures from the school district, we were able to study repair, service, corrective, and preventive maintenance requests for a sample of 20 schools. Sampling criteria included those schools with cooling provided to over 70% of total floor area[1] and the use of one of the following heating and cooling systems:

- vertical-bore geothermal heat pumps (GHPs),
- an air-cooled chiller with gas-fired hot water boiler (ACC/GHWB),
- a water-cooled chiller with gas-fired hot water boiler (WCC/GHWB), or
- a water-cooled chiller with gas-fired steam boiler (WCC/GSB).

The review of maintenance costs was divided into two components: unplanned (repair, service, and corrective maintenance) and planned (preventive maintenance) actions.

As the first component of this approach we reviewed a database containing maintenance work orders for repair, service, and corrective actions (Martin, Durfee, and Hughes 1999). In an effort to evaluate total maintenance costs for the sample of 20 schools, we also studied a second component of maintenance costs, those from planned or preventive actions. The data used in this analysis are contained in a preventive maintenance database. The results of the two components were combined to calculate total maintenance costs for the schools studied.

[1] Some schools may provide small amounts of supplemental heating or cooling for temporary portable classrooms or underconditioned spaces. The maintenance activities for these supplemental systems are included in the maintenance costs; however, the contribution to total costs was insignificant.

6.2 SOURCES OF DATA

6.2.1 Data on Repair, Service, and Corrective Actions

In order to compare HVAC system maintenance costs for all schools in the Lincoln School District, an understanding of the physical characteristics and equipment installed at each school was necessary. Characteristics such as floor area, facility age, and number of additions, as well as HVAC system types, capacities, and commissioning dates, were provided by the district. Table 6.1 lists the categories of data collected for the building characteristics database. Table 6.2 provides a basic summary of building and system data for schools with the four types of HVAC systems studied. Total floor areas ranged from 22,150 to 367,826 ft^2, school ages from 3 to 75 years, cooling plant ages from 2 to 32 years, and heating plant ages from 3 to 70 years.

The school district maintains a database of maintenance requests that were submitted within the past 2–3 years for all facilities within the district. Maintenance records include the date of request, date of completion, request category (or work codes), craft(s) requested, labor rates, hours and costs, material costs, and a brief description of the problem. The database contains over 300 work codes that identify the category of request. Examples of work codes include heating, cooling, EMS, plumbing, and telephone repair.

Based on a query performed by the school district, over 7600 maintenance requests using HVAC-related work codes were identified. Of these, 2934 were verified as legitimate HVAC-related activities. A record-by-record review of the HVAC-related requests found that many labeled as HVAC work codes were actually concerned with water fountain or restroom repairs. This verification process also determined that the database requests for maintenance actions were mainly repair and service responses to equipment failures or corrective maintenance. Planned actions, such as preventive maintenance, are not included in the database, nor were any capital renewal projects for complete replacement of older HVAC equipment. Following verification, the data were then subdivided by school and examined.

The majority of the database requests indicate that most work was performed by in-house Lincoln Public Schools labor. The Lincoln school district relies heavily on its in-house workforce to handle most maintenance jobs and rarely uses contractors. In addition, because the first-year system warranties had expired, all GHP requests were completed by in-house staff. The average in-house base wage (including fringe benefits) reported for HVAC requests is $14 per hour. Maintenance requests may be handled by a variety of skill levels, depending on the requirements of the task. On-site custodians, mechanical equipment technicians (based at all high schools), and mobile craft specialists have base labor rates, including fringe benefits, that range from $13.50 to $18.75 per hour. The corresponding national averages for in-house base labor rates (Means 1998a), including fringe benefits, are slightly higher, at $15.50 for common maintenance laborers and $27.30 for skilled workers. Including workers' compensation and overhead, the national common and skilled labor rates are $21.14 and $36.85, respectively. Labor costs presented here are normalized to a national basis and include fringe benefits, workers' compensation, and overhead.

In addition to labor rates, hours, and costs, the database provides information on material costs. Because the school district enjoys tax-exempt status, the material costs provided do not include any form of sales tax.

Table 6.1. Data collected to establish database of school characteristics

Type of data	Characteristic	Method of description
Building data	Original floor area	ft^2
	Original age	year
	Number of additions	#
	Additional floor area	ft^2
	Number of portables	#
	Portable floor area	ft^2
	Total floor area	ft^2
Primary cooling	Primary cooling equipment	type
	Age of primary cooling plant	year
	Portion of floor area served	%
	Rated output capacity	tons
Secondary cooling	Secondary cooling equipment	type
	Age of secondary cooling plant	year
	Portion of floor area served	%
	Rated output capacity	tons
Primary heating	Primary heating equipment	type
	Age of primary heating plant	year
	Portion of floor area served	%
	Rated output capacity	MMBtu/h, BHP, kW
Secondary heating	Secondary heating equipment	type
	Age of secondary heating plant	year
	Portion of floor area served	%
	Rated output capacity	MMBtu/h, BHP, kW
Primary distribution	Primary distribution equipment	type
	Age of primary distribution plant	year
	Portion of floor area served	%
Secondary distribution	Secondary distribution equipment	type
	Age of secondary distribution plant	year
	Portion of floor area served	%

Table 6.2. Building, heating, and cooling characteristics for 20 schools in Lincoln, Nebraska

School	School type	Total floor area (ft^2)	Age of school (yrs)	Age of primary cooling system (yrs)	Floor area cooled (%)	Age of primary heating system (yrs)
Group A: Geothermal heat pumps (vertical-bore)						
Campbell	Elem.	69,670	3	3	100	3
Cavett	Elem.	72,550	3	3	100	3
Maxey	Elem.	69,670	3	3	100	3
Roper	Elem.	72,550	3	3	100	3
Minimum		69,670	3	3	100	3
Maximum		72,550	3	3	100	3
Average		71,110	3	3	100	3
Std. deviation		1,663	0	0	0	0
Group B: Air-cooled chiller and gas-fired hot water boiler						
Belmont	Elem.	104,724	75	5	87	5
Humann	Elem.	89,523	8	8	79	8
Minimum		89,523	8	5	79	5
Maximum		104,724	75	8	87	8
Average		97,124	42	7	83	7
Std. deviation		10,749	47	2	5.66	2
Group C: Water-cooled chiller and gas-fired steam boiler						
East	H.S.	367,826	31	2	85	31
West Lincoln	Elem.	66,963	42	21	69	42
Minimum		66,963	31	2	69	31
Maximum		367,826	42	21	85	42
Average		217,395	37	12	77	37
Std. deviation		212,742	8	13	11.31	8
Group D: Water-cooled chiller and gas-fired hot water boiler						
Zeman	Elem.	52,640	24	24	96	24
Everett	Elem.	91,163	70	6	83	70
Fredstrom	Elem.	60,732	15	15	73	15
Goodrich	M.S.	118,632	29	8	90	29
Hill	Elem.	56,016	22	22	86	22
Kahoa	Elem.	54,282	26	26	89	26
McPhee	Elem.	47,784	33	3	100	33
Morley	Elem.	56,391	37	23	78	37
Park	M.S.	191,081	72	8	92	8
Pyrtle	Elem.	44,276	34	32	100	3
Rousseau	Elem.	73,065	34	2	91	34
Bryan	H.S.	22,150	42	26	100	42
Minimum		22,150	15	2	73	3
Maximum		191,081	72	32	100	70
Average		72,351	37	16	89.83	29
Std. deviation		44,596	18	10	8.74	17

6.2.2 Data on Preventive Maintenance

Preventative maintenance (PM) requirements for HVAC equipment are contained within a database that is used to identify and schedule PM requests to be carried out at each school. Annually, custodial and maintenance staff are responsible for over 20,000 PM tasks on mechanical equipment throughout the 50 schools within the district (Styskal 1998). For the 20 schools sampled in this study, the number of HVAC-related PM requests totaled 8392. Each database record includes information on equipment type, equipment identification number and location, date of request, sequence of activity, description of work to be performed, and craft responsible for the work. PM requirements are entered into the database at database inception, after installation of equipment, or by request of on-site custodial or maintenance personnel, and are closely related to manufacturers' requirements. PM tasks are removed after decommissioning of equipment or, again, by request of on-site custodial or maintenance personnel.

PM requests are submitted to on-site personnel and completed on a monthly basis; however, the school district does not record actual labor and material expenditures. Therefore, no formal records of labor hours and material costs per PM request were available. In order to estimate annual PM costs for the 20 schools, labor effort and costs and material costs were developed for each request generated by the PM database. Most of the supporting data on required labor effort and material costs was provided by Means' cost guides (Means 1998a, 1998b), with supplemental material cost data collected from the district itself (when available) and from the *Grainger Industrial and Commercial Catalog* (Grainger 1998).

For the majority of schools studied, it was evident that the scope of PM tasks identified by Means (1998a), on an equipment-by-equipment basis, did not consistently match those identified by the school district. This was not unexpected because the sources used by the two are different: Means is based primarily on detailed records from the Navy and the Army Corps of Engineers, while the Lincoln tasks are related to the schools' interpretations of manufacturers' specifications. Lincoln's tasks most often reflect a portion of those identified by Means. In addition, while Means' *Facilities Maintenance and Repair Data* provides itemized estimates for labor effort on a task-by-task basis for each piece of equipment, material costs are annualized for the aggregated tasks. Itemized material costs were not available from any published source. Therefore, these costs were estimated using Means' *Mechanical Cost Guide Data* (Means 1998a), the Grainger catalog, and limited costs obtained from Lincoln.

Labor costs were calculated using itemized estimates of effort for tasks identified on corresponding equipment from Means, labor rates for the craft responsible for the action, and mobilization/demobilization allowances. Labor rates have been normalized to reflect national in-house rates for common laborers and skilled workers, and include base rate, fringe, workers' compensation, and fixed overhead (Means 1998a). Four crafts were identified: custodial, HVAC specialist and technicians, building crafts, and building maintenance. Custodial, building crafts, and building maintenance rates are $21.15 per hour, while the more skilled HVAC trades cost $36.85 per hour. Mobilization and demobilization allowances are included in aggregated PM tasks listed in Means; however, they are not adequately represented when only a few tasks are chosen by the user (personal communication with Mel Mossman, senior editor, R. S. Means Co., Kingston, Mass., 1999). Adjustments have been made to labor efforts to include mobilization and demobilization requirements, and are based on the original PM task identified by the Lincoln School District, as well as equipment type, size, and location.

As noted in Section 6.2.1, Lincoln performs most of its maintenance work using in-house resources. The only contracted HVAC task identified was for cooling tower chemical treatment. The contract is worth $22,000 annually for all cooling towers in the district. PM costs for this task were allocated to schools utilizing cooling towers, based on the capacity of the cooling systems installed in each.

6.3 REPAIR, SERVICE, AND CORRECTIVE ACTION COSTS

Tables 6.3 and 6.4 summarize labor hours, labor costs, material costs, and total unplanned maintenance costs for schools utilizing the four groups of HVAC systems studied. Table 6.3 presents this information on a per request basis, while the data in Table 6.4 are presented on an annual basis. These results represent a 2- to 3-year snapshot of repair, service, and corrective maintenance actions taken during the lifetime of the installed equipment.

Database records indicate that individual repair, service, and corrective maintenance requests for the GHP systems are less costly than those reported for the conventional systems of various ages (Table 6.3). The average labor effort required per GHP request was 2 h, with a labor cost of $47. While GHP labor costs per request are only slightly lower than the average reported for WCC/GHWB systems ($53), material costs per GHP request ($29) are well below those for ACC/GHWB systems ($79). Average per request material costs are highest for WCC/GSB systems, at $122. As a result, GHP schools reported lower average total costs per request at $77, followed by ACC/GHWB schools at $153, WCC/GHWB schools at $157, and WCC/GSB schools at $253.

On an annual cost basis, combined labor and material costs for repair, service, and corrective actions are lower, on average, for GHP systems than for the three other systems studied (Table 6.4). Average annual labor effort and labor costs reported for the GHP systems, at 43 h and $937, are competitive with the average for WCC/GHWB systems, at 52 h and $1142. Average total labor effort and costs are highest for the WCC/GSB schools, at 121 h and $2703 per year. Similar to the per request comparison, average total annual costs are lowest for GHP schools as a result of lower material costs. GHP schools reported lower average total annual costs at $1508, followed by ACC/GHWB schools at $2870, WCC/GHWB schools at $3250, and WCC/GSB schools at $6487.

The commonly recognized method to compare costs from one case to the next uses an area-normalized basis. Annual reported average unplanned maintenance costs per square foot of floor area are lowest for GHP systems. GHP schools reported total average repair, service, and corrective maintenance costs at 2.13¢/ft2-yr, followed by ACC/GHWB schools at 2.88¢/ft^2-yr, WCC/GSB schools at 3.73¢/ft^2-yr, and WCC/GHWB schools at 6.07¢/ft^2-yr. A review of the building characteristics data seems to indicate a linear relationship between these aggregated costs and cooling system age. While no relationship exists between the heating system age and these aggregated costs (R^2 0), a statistically significant relationship does exist between cooling system age and costs ($R^2 = 0.52$, $p < 0.05$). Figure 6.1 illustrates the linear dependence of aggregated repair, service, and corrective maintenance costs on cooling system age.

Table 6.3. Labor hours and costs (in dollars) per call for repair, service, and corrective maintenance (unplanned maintenance)

School	Labor effort per request (hours)	Labor costs per request[a]	Material costs per request	Total costs per unplanned request
Group A: Geothermal heat pumps (vertical-bore)				
Campbell	2	43	47	89
Cavett	2	45	39	84
Maxey	3	58	17	75
Roper	2	45	15	59
Minimum	2	43	15	59
Maximum	3	58	47	89
Average	2	47	29	77
Std. deviation	0	7	16	13
Group B: Air-cooled chiller and gas-fired hot water boiler				
Belmont	4	88	80	168
Humann	3	62	77	139
Minimum	3	62	77	139
Maximum	4	88	80	168
Average	3	75	79	153
Std. deviation	1	19	2	21
Group C: Water-cooled chiller and gas-fired steam boiler				
East	3	76	150	226
West Lincoln	8	185	95	280
Minimum	3	76	95	226
Maximum	8	185	150	280
Average	6	131	122	253
Std. deviation	3	78	39	39
Group D: Water-cooled chiller and gas-fired hot water boiler				
Zeman	2	52	40	92
Everett	2	42	39	81
Fredstrom	2	39	41	80
Goodrich	2	54	48	101
Hill	3	61	45	105
Kahoa	2	50	41	92
Mcphee	1	26	36	62
Morley	4	84	305	389
Park	3	69	73	142
Pyrtle	3	74	392	466
Rousseau	2	34	24	58
Bryan	2	48	168	217
Minimum	1	26	24	58
Maximum	4	84	392	466
Average	2	53	104	157
Std. deviation	1	17	122	134

[a] Labor costs include base wages, worker compensation, and overhead and are normalized using national averages. Costs do not include preventive maintenance or capital renewal actions.

Table 6.4. Annual maintenance requests, labor hours, and costs for repair, service, and corrective maintenance (unplanned maintenance)

School	Requests per year	Labor effort per year (hours)	Costs per year ($) Labor[a]	Material	Total	Total costs per ft^2-year (¢/ft^2-yr)
			Group A: Geothermal heat pumps (vertical-bore)			
Campbell	19	36	793	865	1,658	2.38
Cavett	20	42	919	786	1,705	2.35
Maxey	23	61	1,315	393	1,708	2.45
Roper	16	33	724	238	962	1.33
Minimum	16	33	724	238	962	1.33
Maximum	23	61	1,315	865	1,708	2.45
Average	20	43	937	571	1,508	2.13
Std. deviation	3	13	264	303	365	0.54
			Group B: Air-cooled chiller and gas-fired hot water boiler			
Belmont	24	93	2,072	1,894	3,966	3.79
Humann	13	36	786	988	1,774	1.98
Minimum	13	36	786	988	1,774	1.98
Maximum	24	93	2,072	1,894	3,966	3.79
Average	18	65	1,429	1,441	2,870	2.88
Std. deviation	8	40	909	641	1,550	1.28
			Group C: Water-cooled chiller and gas-fired steam boiler			
East	43	148	3,277	6,481	9,758	2.65
West Lincoln	11	94	2,128	1,088	3,216	4.80
Minimum	11	94	2,128	1,088	3,216	2.65
Maximum	43	148	3,277	6,481	9,758	4.80
Average	27	121	2,703	3,784	6,487	3.73
Std. deviation	22	38	812	3,813	4,626	1.52
			Group D: Water-cooled chiller and gas-fired hot water boiler			
Zeman	35	82	1,810	1,397	3,207	6.09
Everett	8	16	347	324	671	0.74
Fredstrom	7	12	274	290	563	0.93
Goodrich	36	88	1,947	1,721	3,668	3.09
Hill	26	70	1,548	1,138	2,686	4.79
Kahoa	27	63	1,372	1,125	2,497	4.6
Mcphee	18	21	469	640	1,110	2.32
Morley	24	87	1,971	7,205	9,176	16.27
Park	14	44	958	1,014	1,972	1.03
Pyrtle	21	70	1,578	8,332	9,910	22.38
Rousseau	30	46	1,020	711	1,731	2.37
Bryan	8	18	405	1,409	1,814	8.19
Minimum	7	12	274	290	563	0.74
Maximum	36	88	1,971	8,332	9,910	22.8
Average	21	52	1,142	2,109	3,250	6.07
Std. deviation	10	29	648	2,689	3,089	6.71

[a] See note a to Table 6.3.

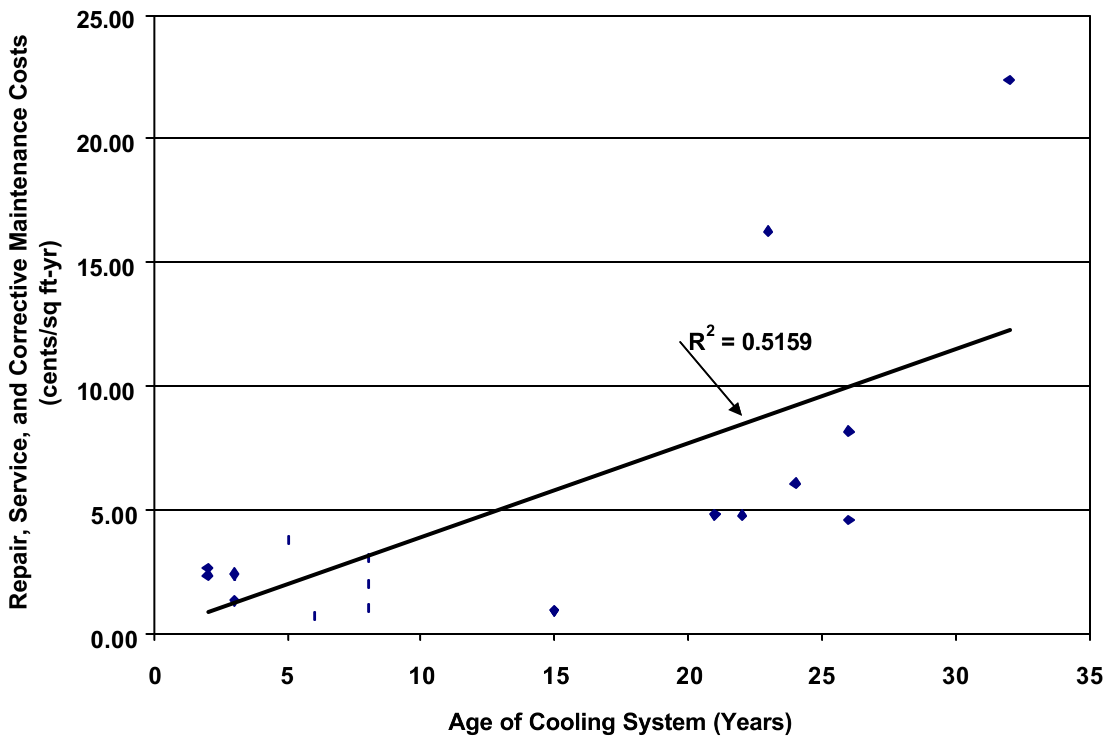


Fig. 6.1. Relationship between aggregated repair, service, and corrective action maintenance costs and cooling system age.

The top five work codes listed for each system type are presented in Table 6.5. Clearly, there is some overlap between work codes as well as definitions that are extremely general. The selection of appropriate work codes is dependent on the requestor, and thus gray areas exist between codes. Nevertheless, there is some value in the generalizations. Requests listed under heating and air-conditioning generally pertain to problems with the plant and air-handling equipment but also include complaints of "room too cold" or "room too hot." EMS and control requests are similar in that both list issues with thermostats or complaints about comfort. Many requests under EMS, however, specifically mention reprogramming, replacement of batteries, and communication problems. The other major work codes—pump repair, boiler repair, and ventilation repair—are more clearly defined.

A closer look at the details of GHP requests indicates several common, recurring equipment, design, and installation problems. The most common problem, which is actually an application flaw, is leakage found in the packing of motorized two-way ball valves located at each heat pump. The two-way valves isolate the unit when it is not in operation and are part of the variable flow design of the water loop. Ball valves with external actuators were selected over solenoids as a cost-saving measure. It is hypothesized that either the external actuators torque the valve stems so that the packing leaks, or that the valves were intended for manual operation and limited cycles. Most of the actuators have been disengaged and the valves left in the open position. Another source of leakage from the GHP units themselves reportedly came from the condensate lines or drip pans. Condensate leaks may be caused by problems such as units that are not mounted level, failure to flush lines with biocide to prevent clogs, damage to the unit during shipping or at the site, poor mounting of the condensate drip pan, or pinched hoses. Several occurrences of water leakage at the loop central pumping station were also reported.

Table 6.5. Top five work codes for repair, service, and corrective maintenance actions (unplanned maintenance) by HVAC system

Work code	Total calls per year	Calls per year per school
Group A (4 schools): Geothermal heat pumps (vertical-bore)		
Heating	20	5
Air-conditioning	10	2.5
Pump repair	9	2.25
Controls	9	2.25
EMS equipment	6	1.5
Group B (2 schools): Air-cooled chiller and gas-fired hot water boiler		
Air-conditioning	6	3
EMS equipment	6	3
Controls	4	4
Boiler repair	5	2.5
Group C (2 schools): Water-cooled chiller and gas-fired steam boiler		
Controls	9	4.5
Air-conditioning	8	4
Ventilation	5	2.5
Heating	4	2
Boiler repair	4	2
Group D (12 schools): Water-cooled chiller and gas-fired hot water boiler		
Controls	60	5
Heating	29	2.4
Boiler repair	27	2.25
Air-conditioning	22	1.8
Ventilation	16	1.3

Like the schools with conventional systems, all GHP schools reported common thermostat, freeze-stat, or other control-related issues, in addition to EMS alarms for GHP unit shutdowns. Filter racks and/or access doors were added to many of the heat pumps to make filter changes and servicing easier for personnel. Finally, a handful of requests were received to repair heat pump vibration or noise.

Many of the requests listed for the GHP systems identify concerns that have been commonly expressed by designers, contractors, and those considering GHPs as an alternative to more conventional heating and cooling equipment. While the existence of these issues in this database supports such concerns, it is obvious that many of the problems could have been avoided with improvements in application and/or installation. In addition, as is evident in the cost summary data, resolution of these problems proved to be inexpensive because maintenance actions were completed quickly with low-cost materials by less-skilled laborers.

Table 6.6 illustrates the average annual costs for these unplanned actions. Average costs for unplanned maintenance actions ranged from 2.13¢/ft^2 to 6.07¢/ft^2. While geothermal systems had the lowest costs, they also were among the youngest HVAC systems in the district, with an average cooling system age of 3 years, versus 7 years for ACC/GHWB, 12 for WCC/GSB, and 16 for WCC/GHWB.

Table 6.6. Annual costs for repair, service, and corrective maintenance, Lincoln Public Schools

HVAC system type	Average age of cooling system (years)	Annual unplanned maintenance costs (¢/ft^2-yr)		
		Average	Minimum	Maximum
Geothermal heat pumps (vertical-bore)	3	2.13	1.33	2.45
Air-cooled chiller and gas-fired hot water boiler	7	2.88	1.98	3.79
Water-cooled chiller and gas-fired steam boiler	12	3.73	2.65	4.80
Water-cooled chiller and gas-fired hot water boiler	16	6.07	0.74	22.8

Note: Labor costs include base wages, workers' compensation, and overhead and are normalized using national averages.

6.4 PREVENTIVE MAINTENANCE COSTS

The number of annual PM requests, labor costs, material costs, and total PM costs for typical units of equipment in each school studied are identified in Table 6.7. Units may be GHPs, pumps, air-handlers, exhaust fans, unit ventilators and/or heaters, chillers, boilers, cooling towers, air-cooled condensers, or air compressors. On a per unit basis, average annual PM requirements or requests, per unit of equipment, are lowest for GHP schools, at 4.8, and highest for WCC/GSB schools, at 7.5. Similarly, average labor costs per GHP component are lowest ($32), followed by ACC/GHWB ($45), WCC/GSB ($57), and WCC/GHWB ($65). Average material costs, however, are lowest for the ACC/GHWB schools ($31 per unit) and highest for the WCC/GHWB schools ($73 per unit). Overall PM costs, on a per unit basis, are lowest for the GHP schools, at $65, and highest for the WCC/GHWB schools, at $138.

Annually, the GHP schools exhibit lower average total PM costs than the schools with conventional HVAC systems (Table 6.8). Itemized components of total PM costs include labor costs, material costs, and contracted PM costs for cooling tower water treatment. The variation in average total PM costs is due to the size of the schools and the number of HVAC units requiring PM, as well as the particular PM needs of the components of the HVAC system. From this perspective, the GHP schools have the lowest average annual PM costs, at $5074, followed by ACC/GHWB at $5808, WCC/GHWB at $8255, and WCC/GSB at $13,075. It is interesting to note that the standard deviation for the GHP schools is relatively small, at $225. The reason for this is that the GHP schools all have the same mechanical design and basic service requirements. The fact that Campbell and Maxey Schools use equipment from a different heat pump manufacturer than Roper and Cavett does not seem to have an impact on this.

Table 6.7. Annual preventive maintenance (PM) costs per unit of equipment by type of HVAC system and school (planned maintenance)

School	Annual PM requests per unit	Costs per unit ($)		
		Labor	Material)	Total
Group A: Geothermal heat pumps (vertical-bore)				
Campbell	4.4	30	35	65
Cavett	5.2	34	29	63
Maxey	4.7	32	38	70
Roper	4.8	32	29	61
Minimum	4.4	30	29	61
Maximum	5.2	34	38	70
Average	4.8	32	33	65
Std. deviation	0.3	1	4	4
Group B: Air-cooled chiller and gas-fired hot water boiler				
Belmont	4.9	34	44	78
Humann	7.8	56	18	75
Minimum	4.9	34	18	75
Maximum	7.8	56	44	78
Average	6.3	45	31	76
Std. deviation	2.1	16	18	3
Group C: Water-cooled chiller and gas-fired steam boiler				
East	6.8	49	67	116
West Lincoln	8.2	65	39	104
Minimum	6.8	49	39	104
Maximum	8.2	65	67	116
Average	7.5	57	53	110
Std. deviation	1.0	12	20	8
Group D: Water-cooled chiller and gas-fired hot water boiler				
Zeman	3.5	27	37	64
Everett	5.2	55	109	165
Fredstrom	6.1	60	100	160
Goodrich	9.8	82	67	150
Hill	4.0	29	15	45
Kahoa	6.4	54	51	105
McPhee	9.3	87	95	182
Morley	9.8	88	79	166
Park	6.3	62	149	211
Pyrtle	9.1	79	50	128
Rousseau	9.7	89	68	157
Bryan	8.9	71	50	122
Minimum	3.5	27	15	45
Maximum	9.8	89	149	211
Average	7.4	65	73	138
Std. deviation	2.4	21	36	48

Table 6.8. Annual preventive maintenance costs by HVAC system and school (planned maintenance)

School	Annual PM requests	Labor effort (hours)	Costs ($)				PM costs per ft^2-year (¢/ft^2-yr)
			Labor	Material	Contracted PM	Total	
Group A: Geothermal heat pumps (vertical-bore)							
Campbell	334	109	2,307	2,651	0	4,958	7.12
Cavett	429	132	2,792	2,447	0	5,239	7.22
Maxey	349	114	2,417	2,867	0	5,285	7.59
Roper	381	120	2,540	2,276	0	4,816	6.64
Minimum	334	109	2,307	2,276	0	4,816	6.64
Maximum	429	132	2,792	2,867	0	5,285	7.59
Average	373	119	2,514	2,560	0	5,074	7.14
Std. deviation	42	10	208	256	0	225	0.39
Group B: Air-cooled chiller and gas-fired hot water boiler							
Belmont	477	156	3,362	4,302	0	7,664	7.32
Humann	413	136	2,991	960	0	3,951	4.41
Minimum	413	136	2,991	960	0	3,951	4.41
Maximum	477	156	3,362	4,302	0	7,664	7.32
Average	445	146	3,177	2,631	0	5,808	5.87
Std. deviation	45	14	263	2,363	0	2,626	2.05
Group C: Water-cooled chiller and gas-fired steam boiler							
East	563	181	4,053	5,590	6,253	15,895	4.32
West Lincoln	757	266	5,991	3,594	670	10,225	15.31
Minimum	563	181	4,053	3,594	670	10,225	4.32
Maximum	757	266	5,991	5,590	6,253	15,895	15.31
Average	660	224	5,022	4,592	3,461	13,075	9.82
Std. deviation	137	60	1,371	1,411	3,948	3,988	7.77
Group D: Water-cooled chiller and gas-fired hot water boiler							
Zeman	139	51	1,084	1,474	1,263	3,822	7.26
Everett	207	88	2,212	4,377	912	7,501	8.23
Fredstrom	171	68	1,672	2,808	1,093	5,573	9.18
Goodrich	1,178	450	9,885	8,080	1,305	19,270	16.24
Hill	152	52	1,118	585	1,456	3,160	5.64
Kahoa	289	102	2,430	2,308	1,520	6,259	11.53
McPhee	373	150	3,462	3,809	669	7,940	16.62
Morley	580	231	5,163	4,656	789	10,609	18.65
Park	291	118	2,850	6,873	2,866	12,590	6.59
Pyrtle	556	211	4,789	3,037	1,018	8,884	19.98
Rousseau	476	193	4,383	3,326	1,534	9,244	12.65
Bryan	277	94	2,207	1,560	487	4,254	19.21
Minimum	139	51	1,084	585	487	3,160	5.64
Maximum	1,178	450	9,885	8,080	2,866	19,270	19.98
Average	391	151	3,438	3,574	1,243	8,255	12.65
Std. deviation	290	112	2,444	2,195	613	4,477	5.30

Table 6.9 provides a list of the most frequent PM work codes cited for each system type. Filter replacement, lubrication of motors and pumps, and belt checks are the PM activities most often implemented for both GHP and conventional systems. In the case of the GHP systems, the manufacturer recommends the following activities: keeping air out of water coils, maintaining positive loop pressure, periodically checking water coils for scaling, inspecting (and replacing) filters every 2–3 months, inspecting (and cleaning) condensate pans and drains twice a year, and inspecting (and cleaning) air coils once a year (Water Furnace 1996). By comparison, the Lincoln Schools' PM database issues requests for filter inspection (and replacement) every 3 months, belt inspection every 3 months, and motor and pump lubrication every 2–6 months. While this practice is less than ideal, it is not uncommon for organizations to address only what they have the capacity to handle on a regular basis. Under these conditions, neglected PM activities eventually surface as repair, service, or corrective actions. Our review of the repair, service and corrective action database showed this to be the case for the Lincoln GHP schools, as it was apparent that the omitted PM task of inspecting condensate pans and drains was actually resulting in leakage problems for some GHPs.

Table 6.9. Most frequent work codes cited for preventive maintenance (planned maintenance) activities

Work code	Total calls per year	Calls per year per school
Group A (4 schools): Geothermal heat pumps (vertical-bore)		
Filter replacements	1,336	334.0
Lubrication of motors and pumps	81	20.3
Filter replacement and belt check	56	14.0
Total	1,473	368.3
Total PM calls	1,493	
% of total PM calls	99%	
Group B (2 schools): Air-cooled chiller and gas-fired hot water boiler		
Filter replacements and coil vacuuming	236	118.0
Filter replacements	200	100.0
Filter replacements and belt check	182	91.0
Belt check	96	48.0
Lubrication of motors and pumps	92	46.0
Total	806	403.0
Total PM calls	890	
% of total PM calls	91%	
Group C (2 schools): Water-cooled chiller and gas-fired steam boiler		
Filter replacements	789	394.5
Filter replacements and lubrication	213	106.5
Lubrication and belt check	76	38.0
Lubrication of motors and pumps	59	29.5
Heating plant burner check	32	16.0
Total	1,169	584.5
Total PM calls	1,247	
% of total PM calls	94%	

Table 6.9 (continued)

Work code	Total calls per year	Calls per year per school
Group D (12 schools): Water-cooled chiller and gas-fired hot water boiler		
Filter replacements	2,779	231.6
Filter replacements and lubrication	483	40.3
Filter replacements, lubrication and belt check	269	22.4
Filter replacements and belt check	130	10.8
Lubrication of motors and pumps	118	9.8
Total	3,779	314.9
Total PM calls	4,689	
% of total PM calls	81%	

Table 6.10 summarizes area-normalized, average annual costs for planned maintenance actions for the four systems studied. Average annual PM costs are lowest for the ACC/GHWB schools, at 5.87¢/yr-ft^2; however, the standard deviation is large (2.05¢/yr-ft^2), as Belmont School's annual PM costs outweigh Humann's by nearly 3¢/yr-ft^2. GHP PM costs are second lowest, at an average of 7.14¢/yr-ft^2, followed by WCC/GSB at 9.82¢/yr-ft^2 and WCC/GHWB at 12.65¢/yr-ft^2. Again, standard deviations are large for the other conventional systems, WCC/GSB and WCC/GHWB, at 7.77¢/yr-ft^2 and 5.30¢/yr-ft^2, respectively.

Table 6.10. Annual costs for preventive maintenance, Lincoln Public Schools

HVAC system type	Average age of cooling system (years)	Annual planned maintenance costs (¢/ft^2-yr)		
		Average	Minimum	Maximum
Geothermal heat pumps (vertical-bore)	3	7.14	6.64	7.59
Air-cooled chiller and gas-fired hot water boiler	7	5.87	4.41	7.32
Water-cooled chiller and gas-fired steam boiler	12	9.82	4.32	15.31
Water-cooled chiller and gas-fired hot water boiler	16	12.65	5.64	19.98

Note: Labor costs include base wages, workers' compensation, and overhead and are normalized using national averages.

6.5 TOTAL MAINTENANCE COSTS

Table 6.11 summarizes planned and unplanned maintenance costs, including area-normalized average annual preventative maintenance costs; repair, service, and corrective action costs; and total annual maintenance costs reported by Lincoln databases. Total annual average maintenance costs were determined by combining the results of the PM analysis with those of the study of repair, service and corrective actions. GHP systems report total average annual maintenance costs of 9.27¢/yr-ft^2, which correspond well to the in-house averages of 9.3¢/yr-ft^2 reported in the recent ASHRAE study (Cane, Morrison, and Ireland 1998). ACC/GHWB systems reported the lowest average annual total maintenance cost, 8.75¢/yr-ft^2, outperforming GHP systems by only 0.52¢/yr-ft^2 with equipment that is just slightly older than the GHP equipment. Average WCC/GHWB costs were highest, at 18.71¢/yr-ft^2.

Our examination of unplanned maintenance actions uncovered a linear relationship (R^2 = 0.51) between annual costs for unplanned maintenance (normalized by total floor area) and the age of the schools' cooling systems. A similar analysis of area-normalized PM costs revealed a statistically significant but weaker linear relationship (R^2 = 0.269, p < 0.05) between annual PM costs and age of cooling system. Again, no linear relationship between PM costs and heating system age was identified. When PM costs are combined with the unplanned costs for repair, service, and corrective actions, a linear relationship with cooling system age is retained (R^2 = 0.464, p < 0.05), as is shown in Fig. 6.2. The relationship between these area-normalized total maintenance costs and cooling system age seems to be dominated by the unplanned maintenance actions.

Since most of the schools do not cool their total floor space, we also developed total annual maintenance costs per unit area of cooled floor space for a more accurate comparison (Table 6.11). The average annual total maintenance costs per cooled square foot are lowest for the GHP schools, at 9.27¢/yr-cooled-ft^2, with a standard deviation of 0.9¢/yr-cooled-ft^2. ACC/GHWB costs are second lowest, at an average of 10.43¢/yr-cooled-ft^2, followed by WCC/GSB at 18.68¢/yr-cooled-ft^2 and WCC/GHWB at 20.71¢/yr-cooled-ft^2. The standard deviations for the three conventional systems are 3.31, 14.81 and 11.83¢/yr-cooled-ft^2 for ACC/GHWB, WCC/GSB, and WCC/GHWB, respectively.

Table 6.11. Area-normalized average total maintenance costs

School	Costs per year-ft^2 (¢/yr-ft^2)			Total maintenance costs per year-cooling-ft^2 (¢/yr-ft^2)
	Preventive maintenance	Repair, service, corrective action	Total maintenance	
Group A: Geothermal heat pumps (vertical-bore)				
Campbell	7.12	2.38	9.50	9.50
Cavett	7.22	2.35	9.57	9.57
Maxey	7.59	2.45	10.04	10.04
Roper	6.64	1.33	7.97	7.97
Minimum	6.64	1.33	7.97	7.97
Maximum	7.59	2.45	10.04	10.04
Average	7.14	2.13	9.27	9.27
Std. deviation	0.39	0.53	0.90	0.90
Group B: Air-cooled chiller and gas-fired hot water boiler				
Belmont	7.32	3.79	11.11	12.77
Humann	4.41	1.98	6.39	8.09
Minimum	4.41	1.98	6.39	8.09
Maximum	7.32	3.79	11.11	12.77
Average	5.87	2.89	8.75	10.43
Std. deviation	2.05	1.28	3.33	3.31

School	Costs per year-ft² (¢/yr-ft²)			Total maintenance costs per year-cooling-ft² (¢/yr-ft²)
	Preventive maintenance	Repair, service, corrective action	Total maintenance	
Group C: Water-cooled chiller and gas-fired steam boiler				
East	4.32	2.65	6.97	8.20
West Lincoln	15.31	4.80	20.11	29.15
Minimum	4.32	2.65	6.97	8.20
Maximum	15.31	4.80	20.11	29.15
Average	9.82	3.73	13.54	18.68
Std. deviation	7.77	1.52	9.29	14.81
Group D: Water-cooled chiller and gas-fired hot water boiler				
Zeman	7.26	6.09	13.35	13.91
Everett	8.23	0.74	8.97	10.80
Fredstrom	9.18	0.93	10.11	13.85
Goodrich	16.24	3.09	19.33	21.48
Hill	5.64	4.79	10.43	12.13
Kahoa	11.53	4.60	16.13	18.12
Mcphee	16.62	2.32	18.94	18.94
Morley	18.65	16.27	34.92	44.77
Park	6.59	1.03	7.62	8.28
Pyrtle	19.98	22.38	42.36	42.36
Rousseau	12.65	2.37	15.02	16.51
Bryan	19.21	8.19	27.40	27.40
Minimum	5.64	0.74	7.62	8.28
Maximum	19.98	22.38	42.36	44.77
Average	12.65	6.07	18.71	20.71
Std. deviation	5.30	6.71	10.90	11.83

Table 6.11 (continued)

6.6 CONCLUSIONS

Using databases on repair, service, and corrective and preventive maintenance actions from Lincoln Public Schools, estimates of planned, unplanned, and total annual maintenance costs were developed for a sample of 20 schools located within the district (Table 6.12). Each of these 20 schools utilize one of the four following HVAC systems: GHP, ACC/GHWB, WCC/GHWB and WCC/GSB. Based on a 2- to 3-year snapshot of unplanned work orders recorded in the repair, service and corrective maintenance database, 4 schools heated and cooled with vertical-bore geothermal heat pumps were found to have the lowest average annual repair, service, and corrective maintenance costs, per square foot, when compared to 16 other schools utilizing three other types of conventional HVAC systems. A relationship does exist between these unplanned costs and the age of the cooling system; and at an average age of 3 years, the four GHP systems studied are among the youngest in the district. Preventive maintenance costs, as reflected in a database of preventive maintenance work orders maintained by the school district, indicated that annual PM costs, normalized to total floor area, were least for ACC/GHWB, followed by GHP schools. The same result was obtained for total annual maintenance costs—these were least for ACC/GHWB schools, with GHP schools in second place. However, in total maintenance costs, ACC/GHWB systems outperformed GHP systems only slightly (by about 0.52¢/yr-ft^2), with equipment that is four years older than the GHP equipment; and when total annual maintenance costs were compared on the basis of total *cooled* floor space rather than total floor space, GHP systems had the lowest total maintenance costs.

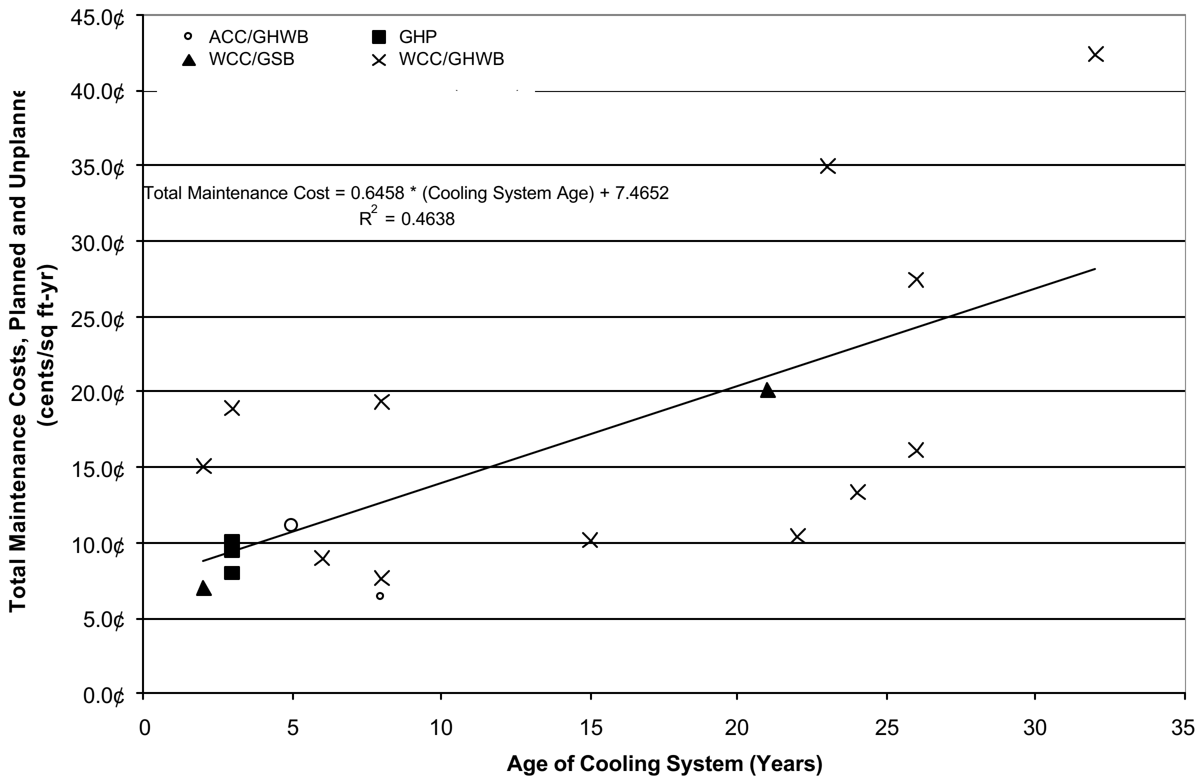


Fig. 6.2. Relationship between total maintenance costs (planned and unplanned) and cooling system age.

Table 6.12. Averages for all maintenance costs by HVAC system

HVAC system	Costs per year-ft^2 (¢/yr-ft^2)			Total maintenance costs per year-cooling-ft^2 (¢/yr-ft^2)
	Preventive maintenance	Repair, service, corrective action	Total maintenance	
Geothermal heat pumps (vertical-bore)	7.14	2.13	9.27	9.27
Air-cooled chiller and gas-fired hot water boiler	5.87	2.88	8.75	10.43
Water-cooled chiller and gas-fired steam boiler	9.82	3.73	13.54	18.68
Water-cooled chiller and gas-fired hot water boiler	12.65	6.07	18.71	20.71

7. TOTAL LIFE CYCLE COSTS

7.1 INTRODUCTION

The extensive operating data from the Lincoln schools' energy management systems allowed us to develop a calibrated simulation model that accurately predicts the energy use of all major building subsystems, including the GHPs, for one of the four schools. This model, combined with independent estimates of installed costs and actual cost information from the school district's maintenance databases, allowed a far more rigorous comparison of the life cycle costs of GHPs and three other space conditioning options than was feasible during the original decision process to select an HVAC system. This chapter presents the results of this more rigorous comparison and shows that when capital, operating, and maintenance costs are considered for the Lincoln application, GHPs have the lowest life cycle cost—about 15% lower than the next most attractive option. For newly constructed schools in Lincoln, Nebraska, GHPs were also found to have a lower first cost than any of the other HVAC systems still commonly used in new schools, as well as the lowest source energy consumption and the lowest total pollutant emissions of any of the HVAC systems considered.

The design of the mechanical systems for Lincoln's four GHP schools was the result of a collaborative effort between the engineer, the school district, and the local electrical utility (Bantam and Benson 1995) in which life cycle costs for five alternative conceptual designs were analyzed using energy consumption inputs from simulations performed with a commercially available software package. Even the preconstruction analysis of estimated capital, operating, and maintenance costs showed the GHPs to have the lowest life cycle cost. However, preconstruction estimates of energy consumption by space-conditioning equipment can vary by 20% or more from the actual energy consumption of an occupied building. Maintenance costs are also difficult to estimate, especially for GHPs, which do not have a long history of application in schools, compared with other space-conditioning technologies. For these reasons, we decided to repeat the comparison of life cycle costs using actual energy consumption data from the schools and maintenance costs from the school district's maintenance database to verify that the preconstruction decision-making process actually led to a good decision.

The energy consumption data allowed us to develop a calibrated simulation model that accurately predicts hourly heating and cooling loads for one of the schools based on occupancy and ambient weather conditions. These loads were used to design three alternative conventional space-conditioning systems for the schools, and independent construction cost estimates were prepared for the GHP system and for the three alternatives. The calibrated simulation was then used to predict the annual energy consumption of the school in a typical year with space-conditioning loads served by each of the four system types. Actual maintenance costs for the installed GHPs were used, and the maintenance cost of the conventional alternatives were determined from the cost of maintaining similar equipment at other schools in the district. This information was used to calculate the life cycle cost of each system.

7.2 ALTERNATIVE SPACE-CONDITIONING SYSTEMS USED IN THE LIFE CYCLE ANALYSIS

The GHP system in Maxey Elementary School was chosen as the baseline system for the life cycle analysis, primarily because of the completeness of the energy consumption data available. A description of the space-conditioning system in this school is provided in Chapter 2.

Three alternative conventional space-conditioning systems were chosen for comparison with the baseline GHP system. An important concern was to maintain consistency with the systems installed in other schools in the Lincoln School District so that the analysis of conventional systems could be based on actual experience. At present, there is some uncertainty in the engineering community about maintenance costs for HVAC equipment in general, and GHPs in particular. A number of publications (e.g., Cane et al. 1997; Shonder and Hughes 1997; Martin, Durfee, and Hughes 1999) indicate that maintenance costs for GHPs may be considerably lower than the values given by ASHRAE (1999) for water-source heat pumps. ASHRAE is known to be reexamining the issue. To minimize the uncertainty in the life cycle cost analysis, we decided to use actual maintenance cost data from the school district's maintenance databases for GHP and conventional HVAC equipment where possible. This influenced the choice of the alternative systems, which are described below.

7.2.1 Option 1: Air-Cooled Chiller with Variable-Air-Volume Air-Handling System (ACC/VAV)

The ACC/VAV system consists of a central air-handling unit with a filter, cooling and heating coils, and a supply air fan. A duct system distributes supply air to variable-air-volume (VAV) terminal units located in the zones, which are identical to the zones supplied by GHPs in the baseline system. An electric reheat coil is included in each zone. An air-cooled centrifugal chiller with capacity of 150 tons and a full-load efficiency of 1.0 kW/ton (COP = 3.57) produces 44°F water for the cooling coils. Heating coils are supplied by a 2800-MBH hot water boiler with 82.5% efficiency. Terminal units in the vestibules and other areas, identical to those installed in the baseline design, are also supplied by the boiler. As in the baseline GHP design, humidification is provided by electric spray humidifiers and domestic hot water, by gas-fired water heaters.

7.2.2 Option 2: Water-Cooled Chiller with Constant-Volume Air-Handling System (WCC/CV)

In the WCC/CV system, constant-volume (CV) forced-flow heating and cooling is provided to individually controlled zones (which are, again, identical to the zones supplied by GHPs in the baseline system installed in the school) from an air handler consisting of a filter, heating and cooling coils, and a draw-through supply fan. A reheat coil is installed in the supply air distribution duct serving each zone. Chilled water at 44°F is provided to the cooling coils by a 150-ton centrifugal chiller with a full-load efficiency of 0.6 kW/ton (COP = 5.86). The cooling tower includes two-speed fan control. A 2800-MBH hot water boiler with 82.5% efficiency provides hot water for heating coils, vestibule units, and outdoor air preheating by way of variable-speed circulation pumps. As in the baseline GHP design, humidification is provided by electric spray humidifiers and domestic hot water, by gas-fired water heaters.

Although systems like Option 2 are no longer commonly installed in schools, the WCC/CV system is actually the most common system in the Lincoln School District because of the age of the building stock. The extensive maintenance cost data for this type of equipment allowed us to calculate the annual rate of increase in maintenance costs, and the system was therefore included in the comparison to maintain consistency with the database.

7.2.3 Option 3: Water-Cooled Chiller with VAV Air-Handling System (WCC/VAV)

The WCC/VAV system is identical to Option 1, except that the 150-ton chiller is water-cooled and has a full load efficiency of 0.6 kW/ton (COP = 5.86). Two-speed fan control is provided for the cooling tower.

7.3 DEVELOPMENT OF THE SIMULATION MODEL

Chapter 3 presented the development of a calibrated simulation model using the TRNSYS software. The TRNSYS model was used to develop consistent inputs for four borefield-sizing packages. For the life cycle cost comparison presented here, we decided to use the DOE-2 software. The main reason for developing a second calibrated model is that DOE-2 is more widely used by engineers than TRNSYS to estimate the energy use of buildings. Thus, we were interested in seeing how well DOE-2 was able to model geothermal systems. A representation of GHP systems was introduced into the software in 1995 in the version known as DOE-2.1E (Gates and Hirsch 1996), and later incorporated without change into version DOE-2.2, the engine that runs PowerDOE (Gates and Hirsch 1997). Version 2.1E was used for this study.

Modeling of the school began with as-built construction plans, from which we obtained construction details and material specifications for floors, walls, ceilings, and windows. The estimates of internal loads were based on occupancy schedules and an on-site survey of light fixtures, computers, and other heat-generating equipment.

In parallel with this effort, 15-min-interval data on water flow rate and inlet and outlet temperatures from the borefield, collected from the school's energy management system, were used to perform an independent calibration of the borefield model in DOE-2. The actual dimensions of the installed borefield (diameter, depth, spacing, and configuration) were entered into the model, and the operation of the borefield was simulated for one year using the fluid flow rate and the inlet fluid temperature to the borefield as inputs. Soil formation thermal properties (far-field temperature, thermal conductivity, and volumetric heat capacity) were adjusted until the predicted outlet temperatures matched the measured outlet temperatures in a least-squares sense for the entire year. Details of this calibration have been presented elsewhere (McLain and Martin 1999).

Results from an independent 50-h soil formation thermal property test performed at Maxey Elementary School (Shonder and Beck 2000) indicated that the thermal conductivity was 1.35 Btu/hr-ft-°F, with a deep earth temperature of 55.7°F. The soil thermal conductivity that resulted in the best fit to the outlet water temperature data using the DOE-2 borefield model was 1.60 Btu/hr-ft-°F, some 18% higher than the measured value. Calibrations of the borefield model in TRNSYS using the same data did in fact converge to soil properties very close to the measured values, as indicated in Chapter 3. One possible reason that the DOE-2 borefield model seems to require a higher conductivity is that the software does not allow the user to specify the borehole backfill material; in effect, the backfill is assumed to have the same thermal properties as the soil formation. Since the thermal conductivity of the fine gravel used to backfill the bores at the site is somewhat higher than the conductivity of the soil, the algorithm in the DOE-2 model may be compensating by requiring a higher soil thermal conductivity in order to replicate the field data.

Although further calibration exercises at other sites would be required to determine whether this is a general problem with the software, engineers using DOE-2 to design GHP systems should be aware of the situation.

The building model, the calibrated borefield model, and representations of the heat pumps and associated equipment were combined to form the overall simulation model. The schedule for the 1995–96 school year was used to model occupancy, and a year of operation was simulated using actual weather conditions for the period. After the initial run, minor adjustments were made to outdoor air infiltration rates and other parameters such as window shading until the annual energy use matched the monitored energy use at the site.

Data from the utility and the EMS separated out the electrical use of the space-conditioning system from the total electrical use. The actual electrical use by the space-conditioning systems from August 1995 to July 1996 was 322,111 kWh; the calibrated DOE-2 model predicted 316,412 kWh, an error of less than 2%. The accuracy of the simulation is illustrated in Fig. 7.1, which presents measured and predicted daily electrical energy use by the space-conditioning systems versus daily average temperature. (Note that the TRNSYS simulation model was calibrated to calendar year 1996 data, while the DOE-2 model was calibrated to data from the 1996 school year.)

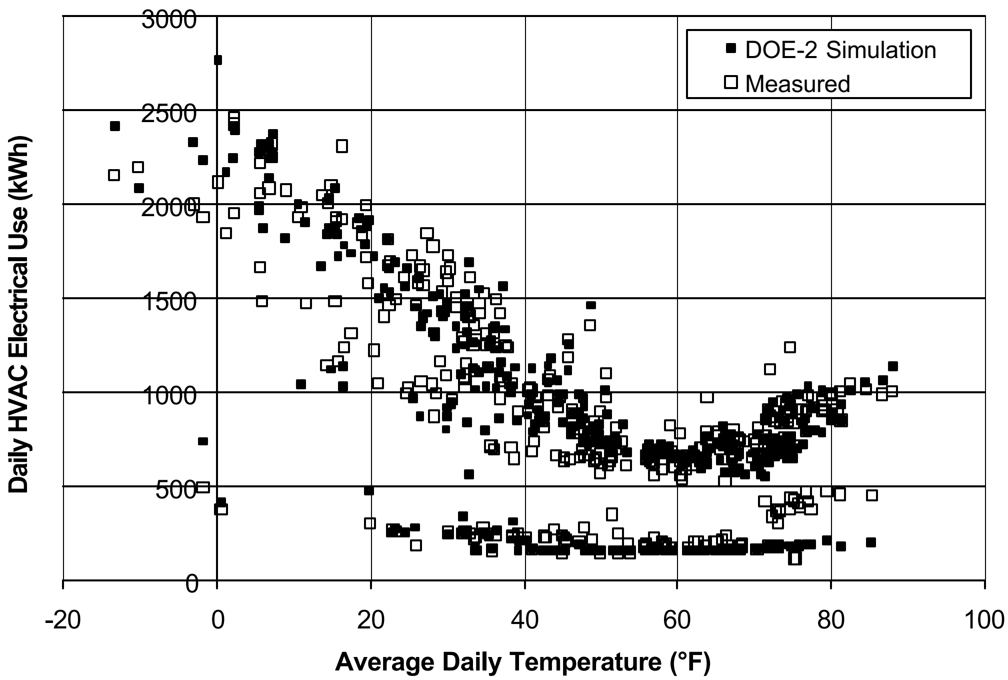


Fig. 7.1. Daily energy use by GHP system vs daily average temperature, August 1995–July 1996: simulation results and site-monitored data.

Figure 7.2 presents a comparison of measured and simulated natural gas use per month for the study period. Because the utility data did not separate out the use of gas specifically for space conditioning, this figure includes the use for domestic water heating. Good agreement is shown. Measured gas consumption for the monitored period was 14,258 therms, while the predicted consumption was 14,274 therms, an error of less than 1%.

Once the simulation was calibrated to the actual year, the model was run again using TMY data for Lincoln, Nebraska, and occupancy based on the school calendar for the 1995–96 school year. The GHPs were then replaced with each of the conventional space-conditioning systems in turn, and the simulation was repeated for the TMY. The annual gas and electric use as calculated by the DOE-2 simulations for the baseline GHP system and the three conventional alternative systems is shown in Table 7.1. The annual source energy use of each option (assuming 70% losses in conversion, transmission, and distribution of electricity) is presented graphically in Fig. 7.3. The baseline GHP system installed at the school is the lowest user of energy, consuming 17% less source energy than the next lowest system, a water-cooled chiller/VAV/gas boiler combination.

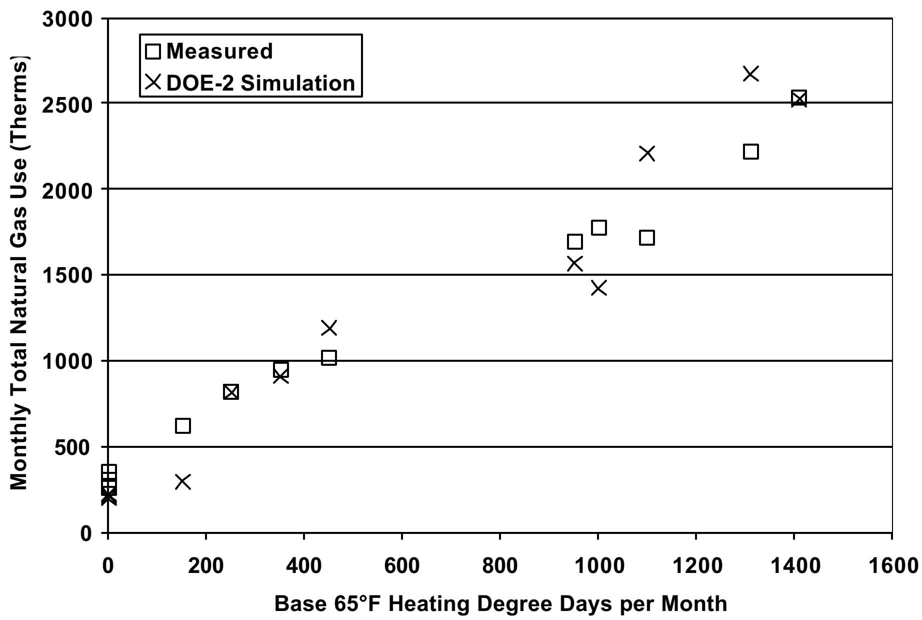


Fig. 7.2. Total monthly gas use vs base-65°F degree days, August 1995–July 1996: simulation results and site-monitored data.

Table 7.1. Annual site energy use for the baseline GHP system and three conventional alternatives as calculated by DOE-2 simulations

Energy use	Baseline: GHP	Option 1: ACC/VAV	Option 2: WCC/CV	Option 3: WCC/VAV
		Electrical use (kWh)		
Non-HVAC	255,807	255,807	255,807	255,807
HVAC systems	288,197	306,855	613,392	280,006
Total electrical use	544,004	562,662	869,199	535,813
		Natural gas use (therms)		
HVAC systems	7,535	22,648	49,021	22,781
Domestic hot water	5,547	5,547	5,547	5,547
Total gas use	13,082	28,195	54,568	28,328
Source energy use, kBtu/ft^2	99.1	123.5	206.6	119.7

The DOE-2 component for the cooling towers does not calculate water loss due to evaporation. Therefore, makeup and blowdown losses for Options 2 and 3 were estimated to be 2% of annual water flow. The losses are approximately 260,000 gal/year for each system. In the life cycle cost analysis, the cost of water and water treatment is included as an annual recurring cost for these two systems.

Comparison with Table 4.1 shows that the calibrated TRNSYS model predicts an HVAC electrical energy use of 314,901 kWh in a TMY, while DOE-2 predicts an electrical use of 288,197 kWh for the same TMY. The difference is approximately 8.5%. The difference in predicted natural gas use is somewhat smaller: for a typical year, DOE-2 predicts consumption of 13,082 therms and TRNSYS predicts 12,424 therms, a difference of about 5%. This result highlights the uncertainty inherent in the use of any simulation model. Since site-monitored data are available only for actual periods of time, there is no way to tell which simulation package makes more accurate predictions of typical year performance.

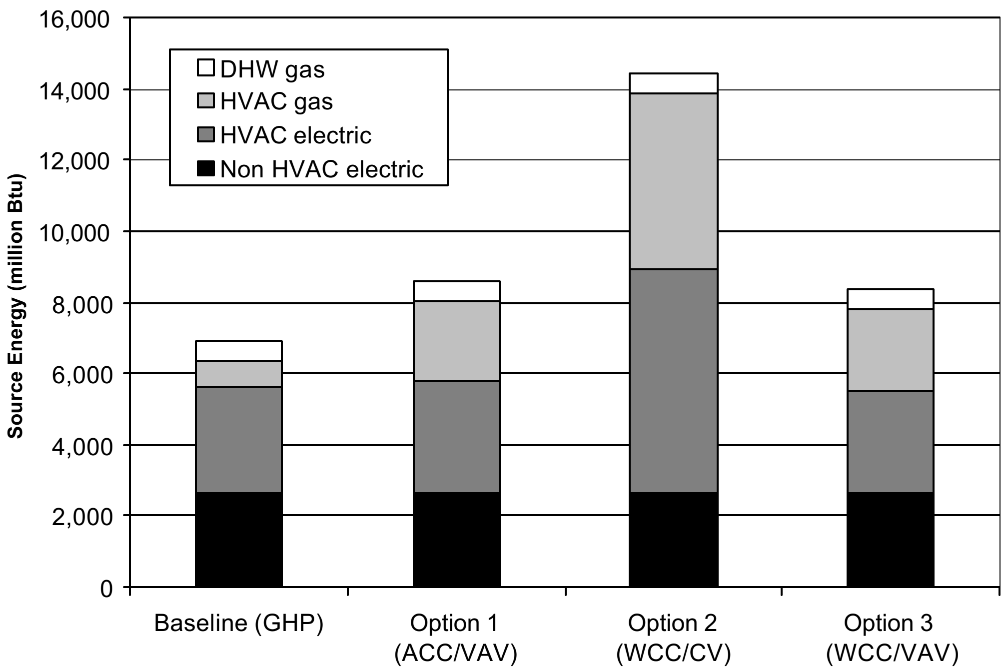


Fig. 7.3. Total source energy use for the baseline GHP and the three alternative HVAC systems, by end use.

7.4 LIFE CYCLE COSTS

7.4.1 Installed System Costs

The installed costs of the baseline system and the three alternatives were estimated by using designs produced to meet the loads seen in the calibrated DOE-2 simulation. In general, total costs were built up from the individual costs of all system components—including chillers, boilers, piping, valves, ductwork, air-handling units, controls and electrical equipment—plus installation costs, management costs, and overhead. In most cases, quoted prices from suppliers and manufacturers were used as the basis for the estimates. Based on current labor rates in

Lincoln, Nebraska, an average rate of $20 per hour was used. General contractor total overhead was assumed to be 25% on labor and 10% on materials, and construction management fees were assumed to be 3% of the construction cost. Because the schools are government-owned, the estimates did not include sales taxes. For the geothermal systems, the price for installation of the ground heat exchangers was assumed to be the actual installation cost of the original installation, $5.93 per bore foot. Costs for all other components of the geothermal systems were independently estimated, without reference to actual installed costs.

The cost estimates were examined by the principal of a Knoxville, Tennessee, engineering firm with extensive experience in design and cost estimating for GHPs and other HVAC systems for schools. The principal made a number of suggestions to improve both the designs and the estimates (J. Regen, personal communication to J. Shonder, November 1999). We then arrived at revised cost estimates on the basis of these suggestions. The final cost estimates for the baseline geothermal system and the three conventional HVAC alternatives are presented in Table 7.2. Except for the constant volume system, which is no longer commonly installed in school applications, the baseline GHP system had the lowest installed cost, about 9.5% lower than the next most attractive option.

The per unit area cost estimates for the three alternative systems compare well with data presented by R. S. Means (1998b), which lists a range of $9.55 to $30.55/ft^2 ($0.89 to $2.84/m^2) for HVAC systems installed in schools, with a national average of $21.65/ft^2 ($2.01/m^2). However, Kavanaugh and Rafferty (1998) indicate that the cost to install the GHPs in the Lincoln schools was approximately $9.44/ft^2 ($1.02/m^2). This figure is based on costs in three broad categories, designated as “well-field contractor,” “HVAC contractor,” and “other fees.” We were unable to determine whether the information presented by Kavanaugh and Rafferty included such components as design fees, power and control electrical wiring, and system controls, as did our estimate.

Table 7.2. Installed cost estimates for the baseline GHP system and the three conventional alternatives

Installation item	Baseline: GHP	Option 1: ACC/VAV	Option 2: WCC/CV	Option 3: WCC/VAV
Engineering				
LES I & II	$41,774	$44,664	$33,270	$45,922
Title III	$20,887	$22,332	$16,635	$22,961
Total engineering	$62,661	$66,996	$49,905	$68,883
Construction Phase				
Loop system	$170,910	—	—	—
HVAC system	$756,356	$1,028,792	$761,059	$1,060,944
Construction management	$31,330	$33,498	$24,952	$34,441
Total estimated cost	$1,021,257	$1,129,286	$ 835,916	$1,164,268
Cost per ft^2	$14.66	$16.21	$12.00	$16.71

7.4.2 Energy, Water, and Water Treatment Costs

Current gas and electric costs were obtained from the local Lincoln utility (Lincoln Electric System 1998). For electricity, the school district is charged 2.5¢/kWh and a demand charge of \$8.50/kW. There is also a customer charge of \$17 per month. Gas rates are \$0.493 per therm, with a \$12 monthly customer charge. The life cycle cost analysis considers only the cost of operating the space-conditioning systems, since the cost of operating other equipment is the same for each option. The customer charges are not included in the analysis.

Water rates charged by the Lincoln Water System are \$1.20 per thousand gallons for the first 120,000 gallons and \$1.48 per thousand gallons thereafter. In the life cycle cost analysis, makeup water for the cooling tower was assumed to be charged at the higher rate. Since the school district's maintenance database includes the costs of boiler water treatment and cooling tower water treatment as maintenance items, these costs are accounted for as necessary in the annual maintenance cost used for each system type.

7.4.3 Maintenance Costs

Using the Lincoln School District's database, Martin and her collaborators (Martin, Durfee, and Hughes 1999; Martin, Madgett, and Hughes 2000) analyzed 2 to 3 years of maintenance records for 20 schools to isolate the annual costs associated with maintaining HVAC systems of various ages and types. For our life cycle cost analysis we disregarded the data for 4 of these schools because they use gas-fired steam boilers, which were not considered here. Data for the remaining 16 schools, which use one of the four system types considered, are presented in Table 7.3 Of these schools, four schools use GHPs (the baseline system), two use air-cooled chillers and gas-fired hot water boilers with VAV air handlers (Option 1), nine use water-cooled chillers and gas-fired hot water boilers with CV air handlers (Option 2), and three use water-cooled chillers and gas-fired hot water boilers with VAV air handlers (Option 3).

The nine cases of systems similar to Option 2 provided sufficient data to develop a correlation between maintenance cost and system age. The data are plotted in Fig. 7.4 and fit to a curve of the form $c = c_0(1+r)^{n-1}$, where c is the system maintenance cost in year n, c_0 is the first-year maintenance cost, and r is the annual rate of increase in maintenance cost. Although the data show a great deal of variation from this curve, both the first-year maintenance costs of 16.6¢ per cooled square foot and the 1.5% annual rate of maintenance cost increase appear reasonable.

Since insufficient data were available to perform similar correlations for the other three system types, the 1.5% annual rate of increase was assumed to apply for the GHPs and for the other alternatives. For the two schools with systems like Option 1 and the three schools with systems like Option 3, the data were correlated to curves of the form $c = c_0(1.015)^{n-1}$ to determine first-year maintenance costs. For the GHP schools, the maintenance cost in year 3 was assumed to be the average of the maintenance costs presented for the four GHP schools; the first-year maintenance cost was determined by bringing this average back to year 1, assuming a 1.5% rate of annual increase. Based on these calculations the first-year maintenance costs are as follows:

Baseline system (GHP):	9.0 ¢/ft^2
Option 1 (ACC/VAV):	9.5 ¢/ft^2
Option 2 (WCC/CV):	16.6 ¢/ft^2
Option 3 (WCC/VAV):	9.7 ¢/ft^2

Table 7.3. Total preventive and corrective maintenance costs per unit area for 16 schools from the Lincoln School District maintenance database

HVAC system and school	Cooling system age (years)	Total HVAC maintenance cost (¢/yr-cooled ft^2)
	GHP/VAV	
Campbell	3	9.50
Cavett	3	9.57
Maxey	3	10.04
Roper	3	7.97
	ACC/VAV	
Belmont	5	12.77
Humann	8	8.09
	WCC/CV	
Zeman	24	13.91
Goodrich	8	21.48
Hill	22	12.13
Kahoa	26	18.12
McPhee	3	18.94
Morley	23	44.77
Pyrtle	32	42.36
Rousseau	2	16.51
Bryan	26	27.40
	WCC/VAV	
Everett	6	10.80
Fredstrom	15	13.85
Park	8	8.28

These are less than half the costs reported by ASHRAE (1999) for similar system types. Data from a 1994 survey of commercial buildings performed by the Building Owners and Managers Association (BOMA 1995) are also higher than our estimates, with an average annual HVAC maintenance cost of 29¢/ft^2 for federal, state, and local government buildings. However, in a report prepared for the Kentucky Utilities Company (1995), several mechanical contractors provide estimates of installation, maintenance, and operating costs of geothermal and other HVAC systems to support life cycle cost comparisons. In this report, first-year maintenance costs are estimated to be 10.0–10.8¢/ft^2 for GHPs and 13.0–45.0¢/ft^2 for other system types. Cane et al. (1997) report a range of 0.94 to 22.66¢/ft^2 and an average of 4.13¢/ft^2 for annual maintenance costs for a sample of 15 schools using GHPs. The wide variation in maintenance costs reported in the literature may be due to a number of factors, including whether scheduled maintenance is performed as opposed to a "fix it when it breaks" policy, the use of contract versus in-house labor, differences in labor and material costs across regions, and differences in accounting practices that sometimes make it difficult to separate HVAC maintenance costs from other maintenance costs. Because the Lincoln database was analyzed in detail, we are confident that these figures represent the best estimate of the costs to maintain the various system types in that particular school district.

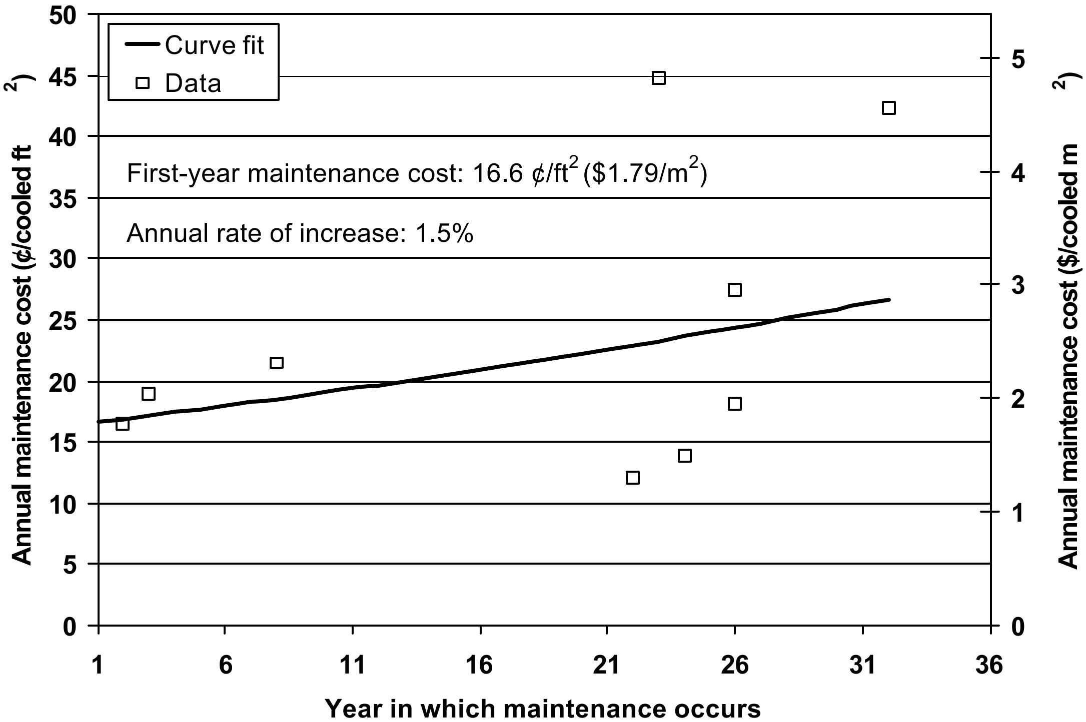


Fig. 7.4. Maintenance costs per unit cooled area vs cooling system age for schools with WCC/GHWB/CV air distribution systems.

7.5 LIFE CYCLE COST ANALYSIS

Once all initial and recurring costs were determined for the baseline GHP system and the three conventional alternatives, the life cycle cost of each one was calculated for a 20-year system life, using the Building Life Cycle Cost (BLCC) software (NIST 1995a). In accordance with NIST guidelines for life cycle cost analysis (NIST 1995b), BLCC calculates present value in constant dollars, using the current (1999) DOE real discount rate of 3.1%, based on the yield of long-term Treasury bonds (net the inflation rate, which currently stands at approximately 2.5%) averaged over the 12-month period from April 1998 to March 1999.

The life cycle cost analysis considered only the gas and electricity (both demand and energy) used by the space-conditioning systems. Since the energy use of all non-HVAC loads is identical across the four simulations, this does not affect the comparison. To determine future gas and electric rates, BLCC uses energy price escalation rate projections from the Energy Information Administration. The current projections were published in April 1999 (NIST 1999).

Maintenance costs for all systems were assumed to increase at an annual rate of 1.5% above the inflation rate, as determined from the database. Water and water treatment costs for the two systems that include cooling towers were assumed to rise only at the general rate of inflation.

The systems were assumed to have no salvage value at the end of their 20-year life. All calculations begin with a base date of January 2000, when the systems are assumed to be installed and begin operation. End-of-year discounting was used throughout the analysis.

Table 7.4 summarizes the inputs for the BLCC analysis of the baseline GHP system and the three conventional systems, as well as their life cycle costs. Of the four systems analyzed for this application, the GHP system has the lowest life cycle cost—$1,498,835 over the 20-year life of the system. The next most attractive option, a VAV system with an air-cooled chiller/gas hot water boiler combination, has a life cycle cost that is about $230,000—or about 15%—higher than the life cycle cost of the GHP system. There is only a small difference (on the order of less than 1%) in life cycle cost between the two VAV systems that use water-cooled and air-cooled chillers. The system with the lowest first cost, the CV system, has the highest life cycle cost of all the options examined, about 26% higher than the life cycle cost of the GHP system. This is why CV systems are no longer commonly used.

Table 7.4. Summary of inputs and outputs from BLCC for the four HVAC systems

Costs	Baseline: GHP	Option 1: ACC/VAV	Option 2: WCC/CV	Option 3: WCC/VAV
Initial cost	$1,021,257	$1,129,286	$835,916	$1,164,268
First year maintenance cost	$7,383	$7,824	$13,651	$7,928
First year electric cost	$22,138	$23,037	$34,152	$19,448
rst year gas cost	$3,533	$10,963	$23,944	$11,034
Water cost	—	—	$385	$385
Total annual O&M costs	$33,054	$41,824	$73,826	$38,795
Life cycle cost	$1,498,835	$1,734,327	$1,912,297	$1,728,736

7.6 POLLUTANT EMISSIONS

Life cycle pollutant emissions for the four technologies, as calculated by BLCC using U.S. average emission factors, are presented in Table 7.5. In terms of carbon emissions, GHPs are the lowest producer of pollutants. Over the 20-year life of the equipment, GHPs produce 437 equivalent tons of carbon less than the next lowest producer, Option 3. GHPs also produce about 1700 pounds less NO_x than Option 3 over the lifetime of the equipment. However, the analysis indicates that GHPs produce somewhat more SO_2 than the equipment used in Option 3. This is due to the fact that while the GHPs use 17% less source energy than the water cooled chiller/gas-fired boiler combination, they do use slightly more electricity (about 1.5%).

7.7 DISCUSSION AND CONCLUSIONS

Our analysis of site-monitored energy use data, actual maintenance cost data, and independent, consistent installed cost estimates shows that the GHP systems installed in the four Lincoln schools have a lower life cycle cost than other space-conditioning systems commonly used in the school district. According to this analysis, a GHP system at Maxey Elementary School would cost

Table 7.5. Life cycle emissions (in kilograms) from the baseline GHP and three conventional alternatives using U.S. average emission factors

Pollutant	Baseline: GHP	Option 1: ACC/VAV	Option 2: WCC/CV	Option 3: WCC/VAV
		Electricity		
CO_2	5,587,321	5,949,045	11,891,931	5,428,520
SO_2	17,911	19,070	38,121	17,402
NO_x	16,831	17,921	35,824	16,353
		Natural Gas		
CO_2	758,505	2,353,678	5,140,425	2,368,783
SO_2	3	9	20	9
NO_x	591	1,834	4,005	1,846
		Total		
CO_2	6,345,826	8,302,723	17,032,356	7,797,303
SO_2	17,914	19,079	38,141	17,411
NO_x	17,422	9,755	39,829	18,199

$230,000, or 15%, less than the next most attractive option over a 20-year system life. Except for one system no longer commonly used in schools (WCC/CV), GHPs had the lowest first cost as well. The GHPs also have the lowest annual operating and maintenance costs of all the systems analyzed, and the lowest consumption of source energy.

Maintenance costs for all four system types were determined from the school's maintenance database. From a sample of nine systems, the annual rate of increase in maintenance costs was determined to be 1.5%. As Fig. 7.4 indicates, there was a great deal of variation in this small sample. To determine the effect of a higher rate of cost increase, the life cycle cost analysis was repeated using a rate of 3%. Life cycle costs of the baseline GHP system and Options 1, 2, and 3 increased by 1.4%, 1.2%, 2.2% and 1.5%, respectively, but the ranking of the options did not change. Thus, the rate of maintenance cost increase has only a small effect on life cycle cost for this analysis.

A discount rate of 3.1% was used in the study to maintain consistency with DOE guidelines. This rate is based on the long-term yield of U.S. Treasury bonds, but school construction projects are commonly financed with tax-free municipal bonds. A survey of 20-year school bonds in the state of Nebraska (Bonds Online 1999) shows an average rate of return of 5.65%, which corresponds to a real discount rate of 3.15%. Thus, the use of the DOE discount rate appears warranted.

Because the costs for system renovation involve even greater uncertainty than first cost, we did not attempt to carry the analysis past 20 years. No salvage value was assumed for any of the systems at the end of this period, but it is likely that the GHP system would have some salvage value if the decision was made to install similar equipment. The pipe in the vertical-bore ground heat exchanger has a 50-year warranty, and thus, the borefield would not have to be replaced. The conventional systems would have some salvage value as well; centrifugal chillers generally have a useful life longer than 20 years. Further analysis would be required to determine renovation costs, but we feel that they would not significantly affect the life cycle cost comparison.

Table 7.4 clearly shows that the installed cost of the space-conditioning systems is the most important parameter in the life cycle cost analysis. Although every effort was made to produce accurate, complete, and consistent cost estimates for the four systems, we recognize that different estimators might produce estimates that differ from ours, perhaps by as much as 5%. Nevertheless, we are confident that the figures presented here provide an accurate comparison of the four systems. These figures show that for school districts such as the one in Lincoln, Nebraska, GHPs offer clear advantages in first cost, operating and maintenance cost, and life cycle cost. Also, given their lower source-energy consumption and reduced pollutant emissions, the technology offers further advantages for the nation as a whole.

8. CONCLUSIONS

The decision by the Lincoln, Nebraska, school district to use geothermal heat pumps in four elementary schools—Campbell, Cavett, Maxey, and Roper—was based on a comparison of the estimated life cycle costs of five alternative system designs: air-cooled variable air volume, water-cooled variable air volume, gas-fired absorption cooling, water-source heat pumps, and ground-source heat pumps. Since the four schools were still in the planning phase at the time, the estimates were based on a model of an existing school in the district, a 151,000-ft^2 middle school. Based on estimates of capital costs and likely operating and maintenance costs, GHPs were determined to be the best alternative. One objective of this report was to determine whether that decision was the correct one, given information that has become available since the schools were constructed. An additional objective was to identify all the factors that make it difficult for owners and engineers to consider GHPs in their projects, so that ongoing programs can address these problems, making GHPs no more difficult to consider than any other HVAC system type in the future.

8.1 ENERGY USE OF THE GHP SCHOOLS

We began by comparing the energy use of the four Lincoln GHP schools with the energy use of other schools in the district. We collected energy consumption and cost data for all schools—37 elementary schools, 11 middle schools, and 5 high schools. We also collected data on the physical characteristics on each school, such as floor area, age and number of expansions, and HVAC system types and ages. Using multiple sources of information (utility account data and 1996 and 1997 billing records, facility reports, and equipment inventories from the Lincoln Public Schools) to perform a rigorous verification of the energy and building characteristics information, we derived a qualified data set of 50 schools. The data indicate that the GHP schools are exceptionally low energy users.

Only seven Lincoln schools (including the four GHP schools) were built in the 1990s and have comparable delivery of ventilation air as well as comparable percentages of floor space cooled. On average, the GHP schools use 26% less source energy per square foot per year than the non-GHP new schools. The GHP schools cool 100% of their floor area and meet the ASHRAE 62-89 ventilation standard. Although 12% of the schools in the district use less energy per square foot than the GHP schools, most of these cool less than 15% of their total floor area and deliver less ventilation air.

8.2 MAINTENANCE COSTS OF GHPs AND OTHER SYSTEM TYPES

Using databases on repair, service, and corrective and preventive maintenance actions from Lincoln Public Schools, we developed estimates of planned, unplanned, and total annual maintenance costs for 20 schools located within the district (Table 6.11). Each of these 20 schools utilize one of the four following HVAC systems: GHP, ACC/GHWB, WCC/GHWB and WCC/GSB.

The 2- to 3-year snapshot of work orders for unplanned maintenance recorded in these databases revealed that the four schools heated and cooled with vertical-bore GHPs had the lowest average annual repair, service, and corrective maintenance costs, per square foot, when compared to the other 16 schools using three other types of conventional HVAC systems. A relationship does exist between these unplanned costs and the age of the cooling system; and at an average age of 3 years, the four GHP systems studied are among the youngest in the district. Preventive maintenance costs, as reflected in a database of preventive maintenance work orders maintained by the school district, indicated that annual preventive maintenance costs, normalized to total floor area, were least for schools with ACC/GHWB systems, followed by GHP schools. The same result was obtained for total annual maintenance costs—these were least for ACC/GHWB schools, with GHP schools in second place. However, in total maintenance costs, ACC/GHWB systems outperformed GHP systems only slightly (by about 0.52¢/yr-ft^2), with equipment that is 4 years older than the GHP equipment; and when total annual maintenance costs were compared on the basis of total *cooled* floor space rather than total floor space, GHP systems had the lowest total maintenance costs per square foot.

8.3 LIFE CYCLE COST OF GHPs

Using as-built construction plans, actual operating schedules from the schools and data from the EMSs, we developed a calibrated DOE-2 simulation model that accurately models hourly heating and cooling loads for one of the schools. We ran this model with four different HVAC system models (GHPs; ACC/GHWB with variable-air-volume air handlers; WCC/GHWB with variable-air-volume air handlers; and WCC/GHWB with constant-volume air handlers) to determine the energy use and annual energy cost of each system type. We obtained independent estimates of installation costs for each system and determined maintenance costs from a detailed analysis of per-square-foot maintenance costs on similar equipment in the school district. The installation, maintenance, and energy cost estimates were used to estimate the life cycle costs of each system. Based on these calculations, GHPs are indeed the most cost-effective option for providing space conditioning for the school. Their life cycle cost is some 15% lower than the life cycle costs of the next most economical option, the WCC/GHWB combination with variable-air-volume air handlers.

8.4 DESIGN OF VERTICAL BOREHOLE HEAT EXCHANGERS

Another objective of this work was to use the site-monitored data to determine the accuracy and consistency of commercially available software for the design of vertical borehole heat exchangers. Although the DOE-2 simulation was found to accurately model the building loads and the performance of the geothermal heat pumps at Maxey School, some shortcomings were found in its vertical ground heat exchanger model. (In order to match the site-monitored data, it was necessary to use a higher soil thermal conductivity than the value we measured at the site.) For this reason, we used the TRNSYS simulation software to develop inputs for the vertical-bore heat exchanger software. This required the development of another simulation, based on as-built construction plans and operating schedules, calibrated to site-monitored data.

Outputs from the calibrated TRNSYS program were used to develop consistent inputs for four ground heat exchanger design programs. On a one-year basis, there was a difference of ±16% between the designs from the four programs and the TRNSYS benchmark design. Over 10 years, the four programs differed by only ±12%. There is therefore reasonable consistency between the available methods for designing vertical bore heat exchangers for GHP systems in schools.

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