Residential building energy use and HVAC system comparison study

by

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ABSTRACT

The objective of this study was to evaluate alternative heating and cooling approaches for a non-typical residence including geothermal and radiant floor heating technology. The analysis included four main components: estimating the design heating and cooling loads of the home, developing alternative approaches for heating and cooling the residence, designing an hourly energy use and heating, ventilating, and air conditioning (HVAC) performance simulation model for the home over a period of one year, and estimating economic factors for each alternative system.

The design heating and cooling loads were estimated using methods recommended by the American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE) organization. These methods are the basis for the Manual J methods recommended by the Air Conditioning Contractors of America (ACCA) which is the current "industry standard" for residential design load calculations. The resulting estimated design heating and cooling loads based on calculations were found to be 6.2 (7.7 tons including the garage) and 4.5 tons for the upper and lower levels of the home, respectively. These estimated design loads were then used in sizing the heating and cooling equipment.

Background information on residential geothermal and radiant floor heating systems was researched; this information is presented within the study. Using this knowledge and considering the design heating and cooling loads, four alternative approaches for conditioning air in the home were developed. These alternatives include systems that utilize either a water-to-air ground-source geothermal heat pump or a liquid-propane gas furnace for the forced air conditioning and either an electric boiler, liquid propane boiler, or a water-to-water ground-source geothermal heat pump for hydronic heating. Subsequently, equipment sizes for each of the approaches were selected.

The hourly simulation model for the home energy demand considers conduction heat transfer through the structure, solar loads, infiltration effects, and internal gain. Typical

Meteorological Year (TMY2) data was used to estimate weather and solar conditions expected at the geographical location (Altoona, Iowa) of the home for each hour over an entire year. Hourly energy demand was estimated for each level and garage of the home. It was found that the home will use approximately 117.1 MMBtu's for heating and 19.9 MMBtu's for cooling per year.

The HVAC model estimates the performance and costs associated with using either a ground-source heat pump or conventional liquid propane furnace and typical air conditioner for the forced air distribution system. In addition, the model estimates the performance and costs associated with using a water-to-water ground-source heat pump, electric boiler, or a liquid propane boiler for the radiant floor heating system for the lower level and garage. The annual operating costs under the current fuel rates for the ground-source heat pump for the forced air heating and cooling were estimated to be \$208 and \$92 dollars respectively. The annual operating costs under the current fuel rates for the water-to-water heat pump, electric boiler, and liquid propane boiler used in combination with the water-to-air heat pump are \$102, \$408, and \$511 dollars, respectively. The annual operating heating and cooling costs for the conventional system, namely, the liquid propane furnace and boiler and a typical air conditioner was found to be approximately \$1,736 dollars in total.

The economics for each alternative approach was evaluated based on a life-cycle-cost analysis. All annual expenses and savings for each approach were estimated over the assumed life of each system. The present-value and payback-period for each system was determined and compared. It was found that the approach utilizing a nominal 5 ton water-to-air ground-source geothermal heat pump and 15 kW electric boiler had the least negative present value of -\$46,645 dollars, and thus, was deemed the most economical. The estimated payback period of this approach was found to be approximately 17.8 years. In addition, for further comparison, many other economic comparisons were considered and include: initial equipment and installation costs, the costs of borrowing money, operation costs and savings, tax savings, and benefit dollars after payback.

NOMENCLATURE

$\dot{m}$	= Mass flow rate of air, ($lb_{m, air}$/sec.)
$\dot{q}$	= Heat transfer rate (Btuh, MBtuh)
GLF	= Glass load factor ($Btu/(hr*ft^2)$)
A	= Area (ft^2)
R	= Thermal resistance ($(hr*ft^2*F)/Btu$)
U	= Thermal transmittance ($Btu/(hr*ft^2*F)$)
SHF	= Sensible heat factor (unitless)
LF	= Latent heat factor (unitless)
Q	= Volumetric flow rate of air, (cfm)
c_p	= Specific heat of air, [$Btu/(lb_m*R)$]
a	= Annual installment amount on loan ($)
Pr	= Principal payment amount on loan ($)
i_T	= Interest rate on loan (%)
N	= Number of years in loan period, where N = 1, 2, 3, ….., 25
B	= Remaining balance of loan ($)
I_P	= Interest payment on loan ($)
K	= Total amount of loan ($)
UA_{eff}	= Effective thermal transmittance value (Btu/(hr*F))
T	= Temperature (F)
HDD	= Heating degree day (F*day)
CDD	= Cooling degree day (F*day)
PV	= Present value ($)
n	= Denotation for year n, where n = 1, 2, 3, ….., 25
CLTD	= Cooling load temperature difference (^oF)
ACH	= Air changes per hour (1/hr)
OAT	= Outdoor air temperature (^oF)
P	= Air pressure (lb_f/in^2)
V	= Air velocity (ft/min)

ρ_{air} = Density of air (lb_m/ft^3)

C = Discharge coefficient (unitless)

WS = Wind speed (ft/min)

$I_{b,T}$ = Hourly beam radiation on a tilted surface (Btu/[ft^2*hr], MBtu/[ft^2*hr])

$I_{b,T}$ = Hourly beam radiation on a horizontal surface (Btu/[ft^2*hr], MBtu/[ft^2*hr])

R_b = Ratio of beam radiation on a tilted surface to that on a horizontal surface (unitless)

θ = Angle of incidence of beam radiation on a surface (degrees, radians)

θ_Z = Zenith angle of beam radiation between vertical and the line to the sun (degrees, radians)

δ = Angular position of the sun at solar noon with respect to the plane of the equator with north positive, $-23.45^o \leq \delta \geq 23.45^o$ – declination (degrees)

φ = Latitude of location, $-90^o \leq \varphi \geq 90^o$ (degrees)

β = Angle between the plane of the surface and the horizontal – slope (degrees)

ω = Angular displacement of the sun east or west of the local meridian at 15 degrees per hour with morning negative and afternoon positive (degrees)

γ = Deviation of the projection on a horizontal plane of the normal to the surface from the local meridian with zero due south and east negative, $-180^o \leq \gamma \geq 180^o$ (degrees)

L_{ST} = Local standard meridian (degrees west)

L_{loc} = Longitude of location, $0^o < L_{loc} > 360^o$ (degrees west)

i = Denotation for the hour of the year, where i = 1, 2, 3, ….., 8,760

$I_{d,T}$ = Hourly diffuse radiation on a tilted surface (Btu/[ft^2*hr], MBtu/[ft^2*hr])

I_d = Hourly diffuse radiation on a horizontal surface (Btu/[ft^2*hr], MBtu/[ft^2*hr])

I = Total hourly irradiation (Btu/[ft^2*hr], MBtu/[ft^2*hr])

ρ_g = Ground reflectance (unitless)

SHGC = Solar heat gain coefficient (unitless)

IAC = Solar attenuation coefficient (unitless)

HC = Heating capacity of ground-source heat pump (Btuh, MBtuh)

CC = Cooling capacity of ground-source heat pump (Btuh, MBtuh)

EWT = Entering water temperature to a ground-source heat pump (^{o}F)

DEWT = Design entering water temperature to a ground-source heat pump (^{o}F)

PLF = Part load factor of a ground-source heat pump (unitless)

C_D = Degradation factor of a ground-source heat pump (unitless)

PD = Power draw of a ground-source heat pump (kW)

AFUE = Annual fuel utilization efficiency (unitless)

COP = Coefficient of performance of a ground-source heat pump (unitless)

EER = Energy efficiency ratio (Btuh/W)

CHAPTER 1 - INTRODUCTION

Overview of Study

Space heating and cooling is the largest single energy expense in most homes, accounting for more than 44 percent of a typical home's utility bill (USDOE, 2004). The type of HVAC system(s) used in a home can significantly impact the overall system efficiency along with monthly and annual operating costs. In addition, the correct sizing of the equipment is critical for ideal operation of the system. While the technology is available for residential applications, often contractors do not perform valid estimates nor do they present the potential cost savings to the consumer, which potentially decreases the use of more efficient systems.

The objectives of this study were to (i) develop a model to simulate the hourly energy performance of a residential home (ii) develop a model for predicting the operating performance and costs of alternative residential HVAC systems and (iii) compare these systems economically. These models can then be used to evaluate a residential home and present the potential cost savings to the consumer. For this study, the models were used to evaluate a home located in Des Moines, Iowa (N latitude 41.5, W longitude 93.7).

The case study home used in this study has a substantially greater amount of living area and glass than would be expected in a typical home, which exacerbates the amount of energy use that will be required for heating and cooling. Thus, selecting a highly efficient means for heating and cooling this home is critical for minimizing the amount of money spent to condition the home.

The sizing of the heating and cooling equipment also has a significant impact on the overall efficiency of the HVAC system, and thus, affects the operating costs. The correct sizing of this equipment is critical to achieve comfortable interior conditions, and saving on initial and operating costs. When the equipment is oversized the system may short-cycle (i.e.

start and stop excessively), which often results in poor control of indoor air humidity levels and excessive wear and tear of the equipment, thus causing premature equipment failure and shortening the life of the system. In addition, the initial costs are higher, and operating efficiency is reduced, and thus, energy costs increase. Conversely, if the system is undersized it again may not be able to maintain comfortable temperatures or humidities in the space and will demonstrate excessive run times due to its inability to meet the load when the structure is subjected to design conditions.

Design heating and cooling loads are determined for a residential home to properly size the heating and cooling equipment. Since the HVAC equipment is sized to these loads, it is vital that these loads are accurately determined. The design heating and cooling loads for this study were determined in accordance to the methods recommended by the American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE). These methods are the basis for the Manual J methods recommended by the Air Conditioning Contractors of America (ACCA), which is an industry standard for residential design load calculations.

A number of methods have been developed to predict the energy use of a residential structure along with the operating performance and costs of the HVAC systems. When residential energy use and/or HVAC operating cost estimations are performed in practice for a homeowner, simplified procedures are generally used due to time constraints imposed on the contractor. Two of the most common simplified methods currently used in practice for estimating residential energy use are the degree-day and bin methods. In addition, contractors may also make rule of thumb estimations based on the size of the home. These methods use many assumptions that may limit the accuracy of the results.

In addition to simplified procedures, commercial software packages are also available to estimate energy use and HVAC performance. However, the use of these packages can be very involved and time consuming. Further, many commercial software packages are written in a fairly general form allowing them to be used for many types of structures. Thus, representing a particular or non-typical residence with great detail can be difficult and/or extremely time consuming due to input constraints imposed by the program. As a result, homeowners and residential contractors can be reluctant to execute these types of analyses.

The home under evaluation possesses numerous characteristics that are not common to a typical residential home. First, the home uses two independent systems for heating, a forced air distribution system for the upper and lower levels, and a hydronic radiant floor system for the lower level and garage. In addition, the lower level is not completely below grade giving some exterior walls exposure to outdoor weather conditions. Also, the home has an additional amount of glass and living space than would typically be expected (average window to wall area is approximately 0.29). Moreover, the garage will be heated in the heating season to a temperature different than that of the living space of the home. As a result of these non-typical characteristics, representing the home and HVAC systems in an existing load simulation model such as Energy Plus or DOE 2 would be difficult due to the input constraints imposed by these programs.

In an effort to improve upon the current methods, an alternative model was developed. This model estimates building energy performance for a residential home and HVAC system performance as well as operating costs for each considered system. The model estimates energy use on an hourly, monthly, and annual basis. It considers transmission, infiltration, variable internal gain, and solar effects which increases the accuracy of the estimates in comparison to the degree-day and bin methods. Increased accuracy in predicting a residence's energy use will allow for a more accurate forecast of HVAC performance and cost predictions. As a result, a more informed decision can be made in less time for selecting the most economical HVAC system to incorporate in the home, which could lead to the use of more efficient technology such as geothermal systems.

Background of Alternative Systems

A major focus of this study is applying alternative energy and innovative technology to a non-typical residence. Therefore, this section presents background information on geothermal and radiant floor systems and on how they may be applied to the case study home. Residential geothermal heat pump systems have become increasingly popular due their ability to reduce the heating and cooling costs for the home. Geothermal systems are energy efficient and environmentally friendly. Geothermal systems can demonstrate increased efficiencies from conventional systems in heating by 50 to 70 percent and cooling by 20 to 40 percent (IGSHPA, 2005). The fundamental principal of residential geothermal systems is that the earth's natural thermal energy, which is a renewable energy, is used to aid in heating and cooling the residence; thus, reducing the amount of energy that would otherwise be self-generated or purchased from the local utility. In addition, many of these systems are used to create hot water, which may supplement or even eliminate the conventional water heater for either hydronic or domestic hot water heating.

There are several types of geothermal systems that could be used for this home. The two main types of geothermal systems used in residential applications are open and closed systems; both systems obtain heat from the ground in the winter and reject heat to the ground in the summer. In an open system, water is pumped from and rejected to a common water source, such as a well or pond. Closed systems circulate the same volume of fluid (e.g., water/glycol mixture) through a series of piping that is functioning as a heat exchanger either laid in the ground or submerged in a pond. Since there is not a pond or sufficient underground water source, a closed system will be used at the case study home.

Closed geothermal systems can be installed in several different ways. There are three types of closed-loop system installations available for residential applications: vertical loop, horizontal loop, and a pond or lake loop. Again, the residence does not have a pond making this option is unfeasible. Choosing between the vertical or horizontal loop system will be mainly dependant upon local contractor availability and cost. Diagrams of horizontal and vertical loop systems can be seen in Figure 1.1.

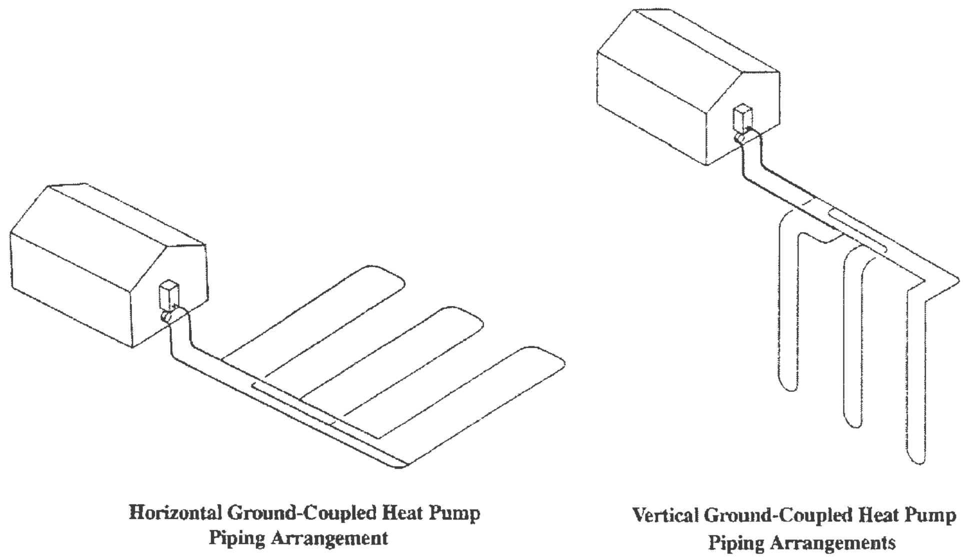

Figure 1.1: Example of a horizontal (left) and a vertical (right) closed loop system
Source: ASHRAE, 1995

Vertical loop systems are in general more expensive; however, these systems place the loop in a more thermally stable zone, resulting in much more consistent and predictable returning water temperatures and overall operation. Moreover, the returning water temperatures will potentially be warmer in the winter and cooler in the summer and thus, increasing the efficiency of the system and saving additional energy costs.

A ground-source heat pump (GSHP) will be the type of equipment used to condition the air to supply the home with both heating and cooling. These systems consist of a reversible vapor compression cycle linked to a closed ground heat exchanger buried in the soil near the home (ASHRAE, 1995). Ground-source heat pumps can come with a wide variety of options and can condition either air or water, while some condition both. Typical closed-loop, ground-source heat pumps demonstrate an EER[1] of 14.1 or more, and a COP[2] of 3.3 or more (USDOE, 2003), making a heat pump an extremely efficient option in comparison to conventional equipment (e.g., gas furnace). A diagram (not to scale) showing typical components of a standard ground-source heat pump used for forced air heating and cooling can be seen in Figure 1.2.

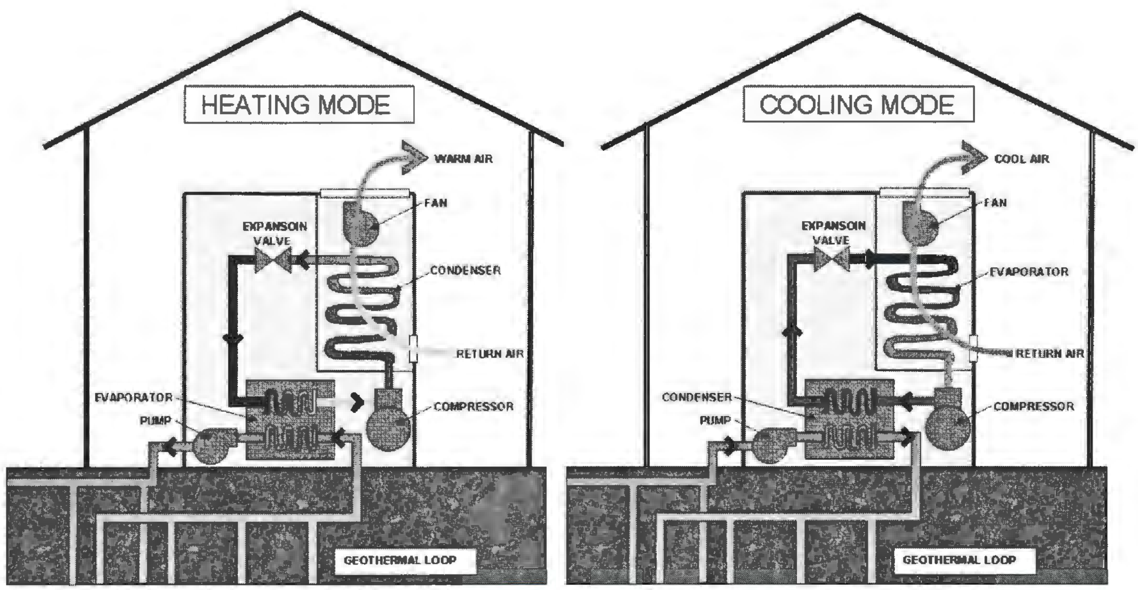

Figure 1.2: Vertical closed-loop ground-coupled forced-air heat pump system

Some additional features available for residential heat pumps include: two speed compressors, variable speed or two speed fans (blowers), desuperheaters, and scroll compressors. Most heat pumps have only single-stage compressors that always operate at full capacity (i.e., one speed) regardless of the load; the efficiency of the system decreases when it is under partial load, which is a majority of the time (USDOE, 2004b). In contrast, a two-speed, or two stage compressor always operates at the capacity that is closest to the

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[1] The EER (Energy Efficiency Rating) is the cooling capacity (in Btu/hr) of the unit divided by the electrical power input (in Watts) to the unit for standard conditions of 77 °F entering water temperature for closed-loop models and includes fan and pumping energy (USDOE, 2003).

[2] The COP (Coefficient of Performance) is the heating capacity (in Btu/hr) of the unit divided by the electrical power input (also in Btu/hr) to the unit for standard conditions of 32 °F entering water temperature for closed-loop models and includes fan and pumping energy (USDOE, 2003).

appropriate capacity to meet the need for heating or cooling at that particular moment; therefore, increasing efficiency and reducing compressor wear (USDOE, 2004b).

Variable speed or two speed fans (blowers) attempt to keep the supply air to the home moving at a comfortable velocity to minimize cooling drafts and maximize efficiency. Additionally, a variable speed blower can be used in conjunction with a two-stage or two speed compressor, which will allow the compressor to operate at low capacity most of the time. Low capacity operation will reduce the compressor on-off cycling as well as temperature fluctuations in the room (USDOE, 2004b). As a result, the efficiency of the system will be further increased.

Desuperheaters are devices that aid a water heater in domestic hot water production. In the heat pump's cooling mode, waste heat from the system is transferred into water entering the home for domestic use; the partially conditioned water is then sent to the water heater. Desuperheaters can heat water 2 or 3 times more efficiently than a conventional water heater (USDOE, 2004b). As a result, the load on the domestic hot water heater is reduced and thus, saving money.

Radiant floor heating systems have become very popular for residential applications due to their quiet operation, net reduction of energy use, and ability to provide superior comfort in the space. These systems evenly heat the entire floor of the room(s) which in turn reradiates to the objects in the room to evenly heat the space. Radiant floor heat can also eliminate draft and dust problems that are commonly associated with forced air systems.

Hydronic radiant floor systems pump hot water through tubing laid in a pattern beneath or within the floor. The hot water heats up the floor and releases radiant heat to the space. Hydronic radiant floor systems are more popular and cost-effective with increased heating (USDOE, 2004a). A hydronic radiant floor heating system was deemed to be the best choice for the lower level of the home and will be the type installed and evaluated. The main components of hydronic radiant floor systems are the tubing, a heat transfer fluid (typically water), the floor, and a device to heat the fluid.

The tubing that is most commonly used today in these systems is cross-linked polyethylene (PEX) tubing with an oxygen diffusion barrier. This type of tubing reduces corrosion problems that are common with copper or steel tubing when in contact with concrete (USDOE, 2004a). Furthermore, PEX tubing has proven to withstand temperature and pressure fluctuations over the long term. This tubing was laid within the lower level and garage concrete floor and backed with two-inch polystyrene foam-board insulation. The insulation on the bottom of the floor has an R-Value of approximately 10 (ft^2*hr*F)/Btu, and it will be used specifically to direct the heat to the space instead of to the ground or between the floors. Devices used to heat the water for these systems include water heating sources such as a hydronic boiler, water heater, solar collector, or a geothermal heat pump.

Typically, for radiant floor heating systems in Iowa, hydronic boilers are used as the auxiliary power to heat the water to supply to the home. High efficiency hydronic boilers can reach an AFUE[3] of approximately 97 percent. A water-to-water heat pump could also be used which may operate at a COP of 3.5 or higher. The standard installation for a boiler or heat pump is comparable, however, the initial cost of a boiler would be less; therefore, the economic analysis over the operating lifetime of each of these devices will be the deciding factor for determining the most economical system for the home.

[3] The AFUE (Annual Fuel Utilization Efficiency) is a statement of efficiency and is the ratio of heat output of a furnace or boiler to the total energy (excluding fan energy) consumed by the furnace or boiler. (USDOE 2004)

CHAPTER 2 - HEATING AND COOLING DESIGN LOAD CALCULATIONS

The case study home is categorized as a single-family detached residential structure. The load calculations will determine the peak, or maximum load in each room. Then, these loads are summed together to determine the design load of the residence for sizing the heating and cooling equipment. Once the load calculations were estimated, they were compared to the estimates made by the heating and cooling subcontractor.

Design Cooling Load Calculation Analysis

The technique used to estimate the design cooling load for the case study home is the ASHRAE recommended Cooling Load Temperature Difference Method (CLTD) which is a simplification of the Transfer Function Method (TFM) (ASHRAE, 2001). The CLTD method assumes that the home will be occupied 24 hours-per-day for virtually every heating and cooling day throughout the year and an indoor temperature swing of no more that 3 ^{o}F on a design day when the thermostat is set at 75 ^{o}F (ASHRAE, 2001). It further assumes that the exterior walls of the home are a dark color (ASHRAE, 2001).

For the design cooling load calculations, the heat gained into the structure per hour at design conditions was estimated. The cooling loads consist of both sensible and latent loads. The estimation of the sensible loads was performed by using the cooling load temperature differences and glass load factors. The estimation of the latent loads was determined by using a load factor. These estimations are explained in greater detail in the following sections.

Sensible Heat Gain through Envelope Components

The sensible heat gain through the glass of the home was calculated using the glass load factors (GLFs) for single-family residences which have been formulated by ASHRAE. The glass load factors account for both the transmission and solar radiation heat gain during summer conditions. The GLFs are a function of the type of glass, type of interior shading, geographical location, and design outdoor air temperature. The types of windows to be used in the home are double pane windows with a Low-E coating and filled with argon gas. The glass under design conditions was assumed to be shaded with fully drawn draperies or translucent roller shades. The total sensible heat transferred into the space through the glass was found by

$$\dot{q} = A_{Glass}(GLF) \tag{2.1}$$

The GLFs for this type of window, interior shading, geographical location, and design outdoor air temperature are tabulated in Table 2.1.

Table 2.1: Glass load factors (GLFs)

Orientation	GLFs [Btu/(hr*ft^2)]
NE and NW	32
SE and SW	41.2

Source: ASHRAE, 2001

The amount of the glass[4] in the home in square feet was calculated using the dimensions in the blue prints for the home and are tabulated in Appendix A, Tables A1 and A2. The estimated cooling load that is attributed to the glass in the home under design conditions was determined to be 28.85 MBtuh. The individual room sensible heat gain loads as a result of the glass can be seen in the Total Cooling Load Summary section in Table 2.5.

The design sensible heat gain through the walls, window frames, doors, ceilings, and floors was calculated using the CLTD values formulated by ASHRAE. The CLTD values represent the effective temperature difference (delta T) across the construction type (i.e., walls, window frames, ceilings, floors, and doors), which accounts for the effect of radiant heat transfer and conduction heat transfer. Furthermore, the CLTD values are a function of

[4] The glass area includes only the glazing, i.e. does not include the frame

the design outdoor temperature, design daily temperature range, and face orientation[5]. The design outdoor temperature was assumed to be 95°F. Also, the daily temperature range for Des Moines, Iowa is 18.5°F (ASHRAE, 2001).

The heat transfer into the space for these construction types were estimated using

$$\dot{q}=UA(CLTD) \tag{2.2}$$

The cooling load temperature differences for the various construction types recommended by ASHRAE for the assumed design conditions can be found in Table 2.2.

Table 2.2: CLTD values for the various construction types

Area	Orientation	CLTD (F)
Walls and Doors and Window Frames	NE and NW	19
	SE and SW	21
Roof	Horizontal	47
Floors	Horizontal	12
Partitions to Unconditioned Space	Vertical	12

Source: ASHRAE, 2001

The U-Value is the measure of heat transmission through a building part or given thickness of material. To determine the necessary U-Values, the total R-Value for the building part was calculated. An R-Value is a measure of resistance to heat flow through a given thickness of material. The total R-Value for a type of construction is the summation of each R-Value for each material of the construction in series, as seen in Equation 2.3.

$$R = R_1 + R_2 + \ldots\ldots + R_N \tag{2.3}$$

[5] The cooling load temperature differences are not a function of orientation for the floors and ceilings since they are assumed to be horizontal.

For building partitions that are exposed to the outdoor and indoor air, a film resistance for the inside, R_i, and outside, R_o, must be included to obtain the overall thermal resistance, R_T, as seen in Equation 2.4.

$$R_T = R_i + R_o + R \tag{2.4}$$

All R-Values for the construction types and inner and outer films were chosen using ASHRAE recommended values. A wind speed of 7.5 miles-per-hour (mph) in the summer was assumed, therefore, value for the outside film resistance of 0.25 (ft^2*F*hr)/Btu was used (ASHRAE, 2001). Using the total R-values for each type of construction the U-value for each type of construction was determined using

$$U = \frac{1}{R_T} \tag{2.5}$$

A summary of each U-Value used in the design cooling load calculation can be seen in Table 2.4.

Table 2.3: U-Values for each construction type for the design cooling load calculations

U-Values - Summer		
Building Construction Type	U-Value [Btu/(ft^2*F*hr)]	Notes / Assumptions
Window Frames	0.1940	Frames approx. 5" thick solid wood
Upper Lev. Above Grade Exterior Wall	0.0471	Shingle and insulation portion of wall
Upper Lev. Above Grade Exterior Wall	0.1274	Shingle and stud portion of wall
Upper Lev. Above Grade Exterior Wall	0.0463	Stone and Insulation portion of wall
Upper Lev. Above Grade Exterior Wall	0.1214	Stone and Stud portion of wall
Exposed Floors	0.0965	Concrete and 2" rigid board Insulation
Ceiling	0.0260	Ceiling only - neglect roof R-Value
Doors	0.3817	Doors assumed to be Approx. 2" thick oak wood
Garage Partition	0.0486	Insulation portion of wall
Garage Partition	0.1386	Studded portion of wall
Lower Lev. Above Grade Exterior Wall	0.0641	Insulation and stone exposed in lower level wall
Lower Lev. Above Grade Exterior Wall	0.1616	Studs and stone exposed in lower level
Exterior Wall Between Floors	0.0641	Assumed to be similar to stone insulation wall

The total R-Value and U-Value calculations for each construction type for summer conditions can be seen in Appendix B. In addition, the areas of the various construction types were calculated by using the blue prints and are tabulated in Appendix A, in Tables A3 through A14. A summary of the cooling loads attributed to these construction types can be seen in Table 2.3.

Table 2.4: Summary of cooling loads for various construction types

Construction Type	Load (MBtuh)
Walls above grade	5.59
Walls below grade	0.26
Window Frames	0.74
Ceiling	2.47
Doors	0.29

The individual room loads contributed by each of these construction types can be seen in the Total Cooling Load Summary in Table 2.5.

Sensible Heat Gain Due to Internal Loads

The internal loads of the residence will consist mainly of occupancy and appliances that run continuously. The internal heat gain due to lights, bathing, cooking, and laundry were neglected. These loads could in fact be considered; however, the likelihood of each of these loads occurring simultaneously and contributing to the block load is minimal and could lead to over-sizing the equipment.

It was assumed that the occupancy will consist of two adults in the home. Each person was assumed to contribute an estimated 230 Btuh of sensible heat (ASHRAE, 2001). For room loads, one occupant was placed in the master bedroom and one in the kitchen on the upper level.

For the appliances, a refrigerator was included in the kitchen on the upper level and in the bar area on the lower level. Each refrigerator was assumed to produce roughly 900 Btuh assuming that each refrigerator is approximately 30 cubic feet in volume (ASHRAE, 2001). The total loads due to the occupancy and appliances were found to be 0.46 and 1.8 MBtuh respectively. A summary of these loads for each room in the residence can be seen in the Total Cooling Load Summary section in Table 2.5.

Sensible Heat Gain Due to Infiltration and/or Ventilation

In the early design stages of this project it was decided to use an energy recovery unit for the circulation of fresh outdoor-air into the home at all times during the use of the heating and cooling equipment. Further research concluded that an appropriate ventilation rate that is universally recommended for a new tightly sealed residential home is approximately 0.35 air-changes-per-hour (ACH) (Home Ventilating Institute, 2003). Since outdoor air will be intentionally introduced into the home while the heating and cooling equipment is in operation, the home will be put into a more positively pressured state. Therefore, infiltration through the structure during these times will be decreased (depending on the outside wind velocity), however, not eliminated.

For the sensible heat gain due to infiltration and/or ventilation, an ACH value of 0.5 was used as the rate of outdoor air entering the structure. This ACH value is recommended by ASHRAE for the assumed conditions. This estimated value accounts for both the ventilation introduced by the heat recovery unit and additional infiltration that may occur simultaneously on the structure. In addition, this is the recommended infiltration rate given by ASHRAE for summer conditions and medium[6] construction further validating this assumption. It should be noted that the heat transfer within the heat recovery unit has been neglected for these calculations and the resulting load estimation will be conservative. The airflow rate of infiltration and ventilation into the home in units of cubic-feet-per-minute (cfm) was determined by

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[6] "Medium" airtightness construction denotes a residential structure that is a new, two-story frame house or one-story house that is more than 10 years old with average maintenance, a floor area greater than 1500 ft^2, average fit windows and doors, and a fireplace with damper and glass closure (ASHRAE, 2001).

$$Q = ACH\left(\frac{Home\,Volume}{60}\right) \tag{2.6}$$

The sensible heat required to cool this amount of air entering the home was calculated by

$$\dot{q} = \dot{m}\,c_p(\Delta T) \tag{2.7}$$

Using an airflow rate in units of cfm and standard temperature and pressure, $\dot{m}\,c_p$ is reduced to a value of 1.1, resulting in heat transfer in units of Btuh. Thus, for the heat transfer in Btuh equation 2.8 below was used.

$$\dot{q} = 1.1Q(\Delta T) \tag{2.8}$$

The sensible cooling load due to ventilation and infiltration to the home was determined to be 8.04 MBtuh. The load due to ventilation and infiltration for each room in the residence can be seen in the Total Cooling Load Summary section in Table 2.5.

Latent Heat Gain

The latent heat gain into the structure was found using the ASHRAE recommended sensible heat factor (SHF) by

$$\dot{q}_{latent} = (1 - SHF) * Total\ Sensible\ Load \tag{2.9}$$

The sensible heat factor is the ratio of the sensible load to the total load.

$$SHF = \frac{Sensible\ Load}{Total\ Load} \tag{2.10}$$

To determine the total load to the structure accounting for the latent loads, the latent factor (LF) was used which is the reciprocal of the sensible heat factor.

$$LF = \frac{1}{SHF} \tag{2.11}$$

Thus, the total design cooling load of the structure was estimated by

$$\dot{q}_{total} = LF * Total\ Sensible\ Load \tag{2.12}$$

ASHRAE recommends using a latent factor of 1.3, which is derived from a sensible heat factor of 0.77, and estimates the performance of a typical residential vapor compression cooling system (ASHRAE, 2001). However, upon further evaluation, it was determined that a more accurate representation of the sensible heat factor for the case study home is 0.82. This result yields a latent factor of 1.22 and was the latent factor used in determining the total cooling load to the home. The latent load was found to be 9.7 MBtuh.

Total Cooling Load Summary

All contributors to the design cooling load have been evaluated and tabulated in Table 2.5.

Table 2.5: Summary of design cooling load calculations

HEAT MODE	Trans. & Solar	Sensible (Btu/hr)									L & S Load (Btu/hr)
HEAT SOURCE / Upper Level Room	Glass (Btu/hr)	Window Frame (Btu/hr)	Doors (Btu/hr)	A.G. Ext Walls (Btu/hr)	Part. Unc. Space (Btu/hr)	Ceiling (Btu/hr)	Between Floors (Btu/hr)	Infiltration (Btu/h)	People (Btu/hr)	Appliances (Btu/hr)	Total Load (Btu/hr)
Garage	0	0	0	0	0	0	0	0	0	0	0
Bath off Garage	240	17	0	134	42	35	15	53	0	0	654
Entryway off Garage	0	0	0	27	29	56	2	85	0	0	243
Pantry	0	0	0	29	80	83	2	125	0	0	391
Kitchen	1,974	96	121	49	0	320	21	485	230	900	5,118
Sunroom	4,046	130	0	288	0	236	41	358	0	0	6,219
Dining Room	2,278	68	0	83	0	216	17	328	0	0	3,647
Great-room	3,964	61	0	276	0	393	27	830	0	0	6,768
Foyer	1,456	17	97	198	0	325	26	540	0	0	3,242
Master Bedroom	2,650	93	0	281	0	279	43	423	230	0	4,876
Hall off Master Bed.	0	0	0	0	0	43	0	65	0	0	133
Bath off Master Bed.	943	40	77	276	0	213	33	322	0	0	2,322
Master Bed Closet	768	22	0	354	0	180	45	273	0	0	2,003
Laundry Rm.	0	0	0	0	0	85	0	129	0	0	261
TOTALS	18,320	544	295	1,996	151	2,465	273	4,016	460	900	35,877
HEAT SOURCE / Lower Level Room	Glass (Btu/hr)	Window Frame (Btu/hr)	Ceiling (Btu/hr)	A.G. Ext Walls (Btu/hr)	A.G. Ext. Walls B.G. (Btu/hr)	Between Floors (Btu/hr)	Exposed Floors (Btu/hr)	Infiltration (Btu/h)	People (Btu/hr)	Appliances (Btu/hr)	Total Load (Btu/hr)
Storage 1	0	0	180	0	42	0	-225	251	0	0	302
Storage 2	0	0	73	0	25	0	-91	102	0	0	132
Game Room	0	0	366	0	59	0	-458	509	0	0	581
Bar Area	0	0	0	0	26	0	-484	538	0	900	1,196
Den	1,814	53	0	99	37	0	-226	314	0	0	2,552
Sunken Rec. Room	5,619	56	0	267	0	0	-644	1,015	0	0	7,698
Hall/Stairs	0	0	0	0	22	0	-204	227	0	0	54
Mechanical Room	0	0	0	0	12	0	-81	90	0		26
Bed 2	2,185	53	0	380	0	0	-304	423	0	0	3,339
Bed 3	907	33	0	230	39	0	-318	443	0	0	1,626
Bath	0	0	0	0	0	0	-99	111	0	0	14
TOTALS	10,526	195	619	976	262	0	-3,134	4,023	0	900	17,521

Design Heating Load Calculation Analysis

The heating loads were calculated in accordance to the recommended ASHRAE methods. For the heating load calculations, the heat lost through the structure in each room per hour for design conditions in the winter was estimated and then summed to attain the design heating load. Typically, the design heating load is estimated for conditions in the middle of the night during the winter time when outside air humidity levels are low; therefore, the heating loads will only account for sensible heat transmission. It was assumed that the moisture levels in the home during the winter will be maintained through the occupancy, bathing, cooking, and laundry.

Sensible Heat Loss through Envelope Components

The design sensible heat losses through the glass, walls, window frames, doors, ceilings, and floors were estimated by using the overall heat transfer coefficient, its area, and the relevant temperature difference across the construction type. The design outdoor-air and indoor-air temperature for the design heating load calculations were assumed to be -9^{o}F and 68^{o}F respectively. Further, it should be noted that the estimation of the heating load for these construction types only accounts for heat transmission losses and neglects any heat gain due to solar loads. This approach was taken assuming that the maximum heating load will occur on the home in the winter time and during the middle of the night when the sun is down, thus, solar gain is irrelevant.

The estimation of the heat loss through each construction type was determined using

$$\dot{q} = UA\left(\Delta T_{design}\right) \tag{2.13}$$

The approach for the determination of the U-Values for each construction type was calculated in the same manner as for the cooling loads assuming that the thermal conductivity of the building envelope is constant with changing temperature. The outside film resistance under the winter conditions is different from summer conditions, and it is due to a different assumed wind speed. The design wind speed in the winter was assumed to be 15 mph, thus, changing the recommended outside film resistance, R_o, from 0.25 to 0.17 (ft^2*F*hr)/Btu (ASHRAE, 2001). All U-Values under the winter conditions were calculated and can be seen in Table 2.6.

Table 2.6: U-Values for each type of construction for the heating load calculations

U-Values - Winter		
Building Construction Type	U-Value [Btu/(ft^2*F*hr)]	Notes / Assumptions
Window Frames	0.1970	Frames approx. 5" thick solid wood
Upper Lev. Above Grade Exterior Wall	0.0473	Shingle and insulation portion of wall
Upper Lev. Above Grade Exterior Wall	0.1287	Shingle and stud portion of wall
Upper Lev. Above Grade Exterior Wall	0.0465	Stone and Insulation portion of wall
Upper Lev. Above Grade Exterior Wall	0.1226	Stone and Stud portion of wall
Exposed Floors	0.0965	Concrete and 2” rigid board Insulation
Ceiling	0.0260	Ceiling only - neglect roof R-Value
Doors	0.3937	Doors assumed to be Approx. 2" thick oak wood
Garage Partition	0.0486	Insulation portion of wall
Garage Partition	0.1386	Studded portion of wall
Exterior Wall Between Floors	0.0644	Assumed to be similar to stone insulation wall
Glass	0.3200	Double pane, Low-E, Argon filled, interior shading

All R-Values with the exception of the glass values were attained as per ASHRAE recommendations. The U-Value for the glass was obtained directly from the glass manufacturer, and was found to be 0.32 Btu/(ft^2*F*hr). This U-Value was determined knowing that the windows are double pane windows with a Low-E coating and filled with argon gas.

The areas for each type of construction were determined from the blueprints and can be found in Tables A1 through A14 in Appendix A. Finally, all of the design heating loads for the previously mentioned construction types were estimated using Equation 2.13 and a summary of these loads can be seen in Table 2.7.

A summary of the loads contributed by each of these construction types for each room can be seen in the Total Heating Load Summary section in Table 2.8.

Table 2.7: Summary of design heating loads for various construction types

Construction Type	Load (MBtuh)
Glass	18.3
Walls	18.0
Window Frames	2.8
Ceiling	3.5
Floors	5.8
Doors	1.2

Sensible Heat Gain Due to Internal Loads

The internal loads consist mainly of occupancy and appliances that run continuously, similarly to the cooling loads. However, the internal loads for the heating calculations will actually contribute to heating the space and will reduce the load on the heating equipment. The internal heat gain due to lights, bathing, cooking, and laundry were again neglected because the peak heating load was assumed to occur in the middle of the night when occupants are most likely asleep.

It was assumed for the design heating load that the occupancy consists of two adults in the home. It was assumed that each person will emit approximately 230 Btuh of sensible heat into the space (ASHRAE, 2001). For room loads, one occupant was placed in the master bedroom and one in the kitchen on the upper level.

For the appliances, a refrigerator was included in the kitchen on the upper level and in the bar area on the lower level. Each refrigerator was assumed to contribute roughly 900 Btuh, assuming that each refrigerator is approximately 30 cubic feet in volume (ASHRAE, 2001). A summary of these loads for each room in the residence can be seen in the Total Heating Load Summary section in Table 2.8.

Sensible Heat Loss Due to Infiltration and/or Ventilation

Again, the ACH method was used to estimate the heat loss due to infiltration and ventilation for the heating load. The assumed flow rate of air entering the home was again 0.5 air changes per hour. The sensible heat lost due to infiltration was determined using Equation 2.8. The contribution of the ventilation and infiltration to the total design heating load was found to be 26.6 MBtuh. The load due to ventilation and infiltration for each room in the residence can be seen in the Total Heating Load Summary section in Table 2.8.

Total Heating Load Summary

All of the estimated heating loads per room and for the entire home can be seen in Table 2.8.

Table 2.8: Summary of design heating load calculations

HEAT MODE	Trans. & Solar	Sensible									Sensible (Btu/hr)
HEAT SOURCE / Upper Level	Glass (Btu/hr)	Window Frame (Btu/hr)	Doors (Btu/hr)	A.G. Ext Walls (Btu/hr)	Part. Unc. Space (Btu/hr)	Ceiling (Btu/hr)	Exposed Floors (Btu/hr)	Infiltration (Btu/h)	People (Btu/hr)	Appliances (Btu/hr)	Total Sensible Load (Btu/hr)
Bath off Garage	185	68	0	516	270	50	0	225	0	0	1,314
Entryway off Garage	0	0	0	111	184	80	0	357	0	0	732
Pantry	0	0	0	108	516	118	0	527	0	0	1,269
Kitchen	1,520	397	506	201	0	457	0	2,040	-230	-900	3,990
Sunroom	2,723	508	0	1,131	0	337	0	1,504	0	0	6,203
Dining Room	1,361	254	0	308	0	308	0	1,377	0	0	3,609
Great-room	2,369	226	0	1,031	0	560	0	3,488	0	0	7,673
Foyer	1,121	70	404	845	0	463	0	2,270	0	0	5,173
Master Bedroom	1,583	345	0	1,037	0	398	0	1,778	-230	0	4,912
Hall off Master Bed	0	0	0	0	0	62	0	275	0	0	337
Bath off Master Bed.	564	151	291	1,045	0	303	0	1,353	0	0	3,706
Closet off Master Bed	591	91	0	1,626	0	257	0	1,149	0	0	3,714
Laundry Room	0	0	0	0	0	121	0	541	0	0	662
TOTALS	12,017	2,109	1,201	7,959	970	3,514	0	16,885	-460	-900	43,294
HEAT SOURCE / Lower Level	Glass (Btu/hr)	Window Frame (Btu/hr)	Doors (Btu/hr)	A.G. Ext Walls (Btu/hr)	B.G. Ext. Walls (Btu/hr)	Exposed Floors (Btu/hr)	Between Floors (Btu/hr)	Infiltration (Btu/h)	People (Btu/hr)	Appliances (Btu/hr)	Total Sensible Load (Btu/hr)
Storage 1	0	0	0	0	664	330	139	120	0	0	1,254
Storage 2	0	0	0	0	380	134	82	49	0	0	644
Game Room	0	0	0	0	771	671	189	244	0	0	1,874
Bar Area	0	0	0	0	475	709	84	257	0	-900	626
Den	1,084	197	0	366	593	508	203	1,321	0	0	4,274
Sunken Rec. Room	3,357	209	0	994	0	1,449	164	4,268	0	0	10,440
Hall	0	0	0	0	308	299	69	953	0	0	1,629
Mechanical Room	0	0	0	0	190	119	40	43	0		392
Bed 2	1,306	197	0	1,403	0	684	198	1,779	0	0	5,567
Bed 3	542	121	0	871	593	716	248	212	0	0	3,303
Bath	0	0	0	0	0	146	0	465	0	0	611
TOTALS	6,289	724	0	3,633	3,974	5,765	1,416	9,710	0	-900	30,614

CHAPTER 3 - HEATING AND COOLING ALTERNATIVE APPROACHES

Several alternative approaches for heating and cooling the home with the geothermal and radiant floor systems are presented within this section. Subsequently, through evaluation of cost, availability, installation, and control flexibility specifics of each alternative, the best available system was chosen for the home. Using geothermal and radiant floor systems, the following approach for conditioning the home was developed, seen in Table 3.1.

Table 3.1: Heating and cooling approach 1

APPROACH 1		
	HEATING	COOLING
Upper Level	Forced Air - GSHP Air Heating	Forced Air - GSHP Air Cooling
Lower Level	Radiant Floor - GSHP Hydronic Heating	Forced Air - GSHP Air Cooling
Garage	Radiant Floor - GSHP Hydronic Heating	None

The possibility of a hybrid system was also considered. Initial research performed as part of this study along with lengthy discussion with two independent subcontractors suggested that using a geothermal approach for the radiant floor heating may be uneconomical in the long run. This suggestion was based on the requirement of additional wells and the required use of more expensive equipment. Further, by using the same loop field for two heat pumps, additional system design, pumps, valves, and controls would be necessary. Therefore, the following two approaches were developed using conventional methods for the radiant floor in conjunction with geothermal means for the forced air heating and cooling. Approach 2 (Table 3.2) utilizes an electric boiler for hydronic heating to the radiant floor.

Table 3.2: Heating and cooling approach 2

APPROACH 2		
	HEATING	COOLING
Upper Level	Forced Air - GSHP Air Heating	Forced Air - GSHP Air Cooling
Lower Level	Radiant Floor – Electric Boiler	Forced Air - GSHP Air Cooling
Garage	Radiant Floor – Electric Boiler	None

Approach 3, seen in Table 3.3 utilizes a gas boiler for hydronic heating to the radiant floor.

Table 3.3: Heating and cooling approach 3

APPROACH 3		
	HEATING	COOLING
Upper Level	Forced Air - GSHP Air Heating	Forced Air - GSHP Air Cooling
Lower Level	Radiant Floor – Gas Boiler	Forced Air - GSHP Air Cooling
Garage	Radiant Floor – Gas Boiler	None

For the purpose of comparing conventional methods of heating and cooling the home, approach 4 was developed and can be seen in Table 3.4.

Table 3.4: Heating and cooling approach 4

APPROACH 4		
	HEATING	COOLING
Upper Level	Forced Air - Furnace	Forced Air Typical Air Conditioner
Lower Level	Radiant Floor – Gas Boiler	Forced Air Typical Air Conditioner
Garage	Radiant Floor – Gas Boiler	None

It should be noted that the residence is located in the country where natural gas is unavailable; therefore, if a boiler and/or furnace is used, it must be supplied with liquid propane gas.

Equipment Selections for Each Heating and Cooling Approach

Determining tentative equipment options for each heating and cooling approach was the next step.

APPROACH 1:

- Option A:
  - One GSHP for hydronic heating to heat the lower level and garage
  - One GSHP for forced air heating and cooling for the upper level and forced air cooling for the lower level

- Option B:
  - One GSHP for hydronic heating for the lower level and garage and forced air heating for the upper level and forced air cooling to the upper and lower level

APPROACH 2:

- Option A:
  - One GSHP for forced air heating and cooling for the upper level and forced air cooling for the lower level
  - One electric resistance boiler for hydronic heating to the lower level and garage

APPROACH 3:

- Option A:
  - One GSHP for forced air heating and cooling for the upper level and forced air cooling for the lower level
  - One gas boiler for hydronic heating to the lower level and garage

APPROACH 4:

- Option A:
  - One gas furnace for forced air heating to the upper level
  - One standard air conditioner for forced air cooling the upper and lower level
  - One gas boiler for hydronic heating to the lower level and garage

The estimated heating and cooling loads that were previously determined for the upper and lower level and garage will be the demand that the equipment must meet in order to maintain a comfortable environment in the home under the assumed design conditions, and can be seen in Table 3.5.

Table 3.5: Summary of the heating and cooling loads on the home

	Upper Level	Lower Level	Garage	Total Home
Heating Load (MBtuh)	43.3	30.6	18.0	91.9
Cooling Load (MBtuh)	35.9	17.5	N/A	53.4

Each approach and option must be evaluated to determine the approximate heating and cooling load that will be imposed upon each type of equipment under design conditions. This is done so that the correct size for each piece of equipment can be selected as a function of the load it could be subjected to under design conditions, which will vary between each approach. For example, it can be seen that Approach 4 requires three pieces of equipment, namely: a liquid propane gas furnace, an air conditioner, and a liquid propane boiler.

In the case that a water-to-water ground-source geothermal heat pump is not chosen, it was decided that both radiant floor and forced air heating for the lower level will be used simultaneously. For the design, the radiant floor heating will be assumed to supply approximately 18.1 MBtuh (60 percent of design load) and the forced air heating will supply the remaining 12.5 MBtuh (40 percent of design load) under design conditions. The determination of these values will be demonstrated in Chapter 4. The approximate sizing load for each piece of equipment (in bold) of each approach can be seen in Table 3.6.

Table 3.6: Design heating and cooling loads on equipment

App.	Option	Means of Conditioning Space	Design Load		Type of Equipment
			(MBtu/hr)	Tons	
1	A	Forced air heating to the upper level	43.3	3.61	GSHP
		Forced air cooling to the upper and lower levels	**53.4**	**4.45**	
		Hydronic radiant floor heating to the lower level and garage	**36.1**	**3.00**	GSHP
	B	Hydronic heating for lower level and garage and forced air heating to upper and lower levels	**91.9**	7.66	GSHP
		Forced air cooling to upper and lower levels	53.4	4.45	
3	A	Forced air heating to the upper and lower level	**55.8**	**4.65**	GSHP
		Forced air cooling to the upper and lower level	53.4	4.45	
		Hydronic radiant floor heating to the lower level and garage	**36.1**	**3**	L.P. boiler
2	A	Forced air heating to the upper and lower level	**55.8**	**4.65**	GSHP
		Forced air cooling to the upper and lower level	53.4	4.45	
		Hydronic radiant floor heating to the lower level and garage	**36.1**	**3**	Elec. resistance boiler
4	A	Forced air heating to the upper and lower level	**55.8**	**4.65**	L.P. Furnace
		Forced air cooling to the upper and lower level	**53.4**	**4.45**	Typical CAC
		Hydronic radiant floor heating to the lower level and garage	**36.1**	**3**	Elec. Boiler

For Approach 1, Option A1, two ground-source-heat-pumps will be chosen, one water-to-air heat pump and one water-to-water heat pump. To size the heat pumps, several parameters must be initially known or assumed and include: entering water temperature (water temperature from the loop field entering the heat pump) for both heating and cooling modes, air flow rate, and water flow rate through the heat pump. For the initial sizing procedures, average values for water and air flow rates in the selection tables were used for the system. Entering water temperatures of 40 ^{o}F in the heating mode and 70 ^{o}F in the cooling mode were also used. Seen in Table 3.7 is a summary of the heating and cooling capacity of an E-Series water-to-air WaterFurnace heat pump, and an E-Series water-to-water WaterFurnace heat pump (WaterFurnace 2004a, 2004b).

Table 3.7: Approach 1, option A equipment selection and capacity

Unit	Heating Capacity (MBtuh)	Cooling Capacity (MBtuh)
WaterFurnace Heat Pump E060 (nominally 5 tons)	45.6	61.8
WaterFurnace Heat Pump EW042 (nominally 3.5 tons)	44.9	N/A

It can be seen that the total heating capacity of the nominal 5 ton water-to-air and the nominal 3.5 ton water-to-water heat pump is approximately 90.5 MBtuh. The design heat loss to the home is 73.9 MBtuh; using these heat pumps, under design conditions, the heating capacity left for the garage is approximately 16.6 MBtuh. Assuming that the design heat loss to the garage is 18.0 MBtuh (at an indoor air temperature of 50 °F), the system would be able to heat the garage to approximately 45°F, resulting in an acceptable tentative option.

The match of the cooling capacity to the design demand would be acceptable. Using the nominal 5 ton unit on high speed the cooling capacity is about 8.4 MBtuh oversized. A little over-sizing is acceptable in this situation due to the duel speed capability and for pick-up of the space temperature.

For Approach 1, option B, one heat pump can be used. WaterFurnace's Synergy 3 unit has the capability of heating either water or air in the same mode and cooling air. This unit is fairly new on the market and was considered with caution. The heating capacity for both the water and air and cooling capacity for the air can be seen in Table 3.8.

Table 3.8: Approach 1, option B equipment selection and capacity

Unit	Air Heating Capacity (MBtuh)*	Water Heating Capacity (MBtuh)*	Air Cooling Capacity (MBtuh)
WaterFurnace Heat Pump RTV066	60	44.2	64.6

* Unit can heat either air or water in the same mode

The heating capacity of this unit on the air side is approximately 60.0 MBtuh, which falls short of the design heating demand. However, the electric resistance auxiliary heating would be available to make up the remainder of the load. The cooling capacity of this unit in comparison to the design heat gain is acceptable.

Approach 2, option A would essentially replace the water-to-water heat pump with an electric boiler. The specific equipment information for this approach can be seen in Table 3.9.

Table 3.9: Approach 2, option A equipment selection and capacity

Unit	Heating Capacity (MBtuh)	Cooling Capacity (MBtuh)
WaterFurnace Heat Pump E060	45.6	61.8
Thermolec Electric Resistance Boiler	78.43 (15 KW)	N/A

Approach 3, option A, includes a water-to-air heat pump used with a Weil McLain Ultra 105 MBtuh propane fired hydronic boiler. Using the nominal 5 ton heat pump with the boiler will provide adequate capacity for both heating and cooling. The equipment capacity can be seen in Table 3.10.

Table 3.10: Approach 3, option A equipment selection and capacity

Unit	Heating Capacity (MBtuh)	Cooling Capacity (MBtuh)
WaterFurnace Heat Pump E060	45.6	61.8
Weil McLain Ultra 105 Boiler	105	N/A

Finally, approach 4, option A was considered and quoted by the contractor. This option was based entirely on conventional methods of heating and cooling a home and was included in this analysis as a baseline for comparison to the other approaches. This option consists of a liquid propane fired furnace and boiler, and a typical vapor compression refrigeration cycle air conditioner.

Table 3.11: Approach 4, option A equipment selection and capacity

Unit	Heating Capacity (MBtuh)	Cooling Capacity (MBtuh)
Bryant Gas Furnace 92% Model # 340MAV0600120	120.0	N/A
Bryant 5 TON 12 S.E.E.R. Air Conditioner	N/A	60.0
Weil McLain L.P. Ultra 105 Boiler	105.0	N/A

This system will easily meet both the heating and cooling demand of the home under design conditions. However, the heating and capacity of the system is in great excess of the estimated design loads.

CHAPTER 4 - ANNUAL ENERGY USE AND HVAC OPERATING PERFORMANCE MODEL

The model presented herein estimates monthly and annual energy use of the residence by executing an hourly simulation of the relevant energy transfers. This method is based on an energy balance and considers transmission, solar, infiltration, and internal loads on the home. A diagram showing each of the considered energy transfers can be seen in Figure 4.1.

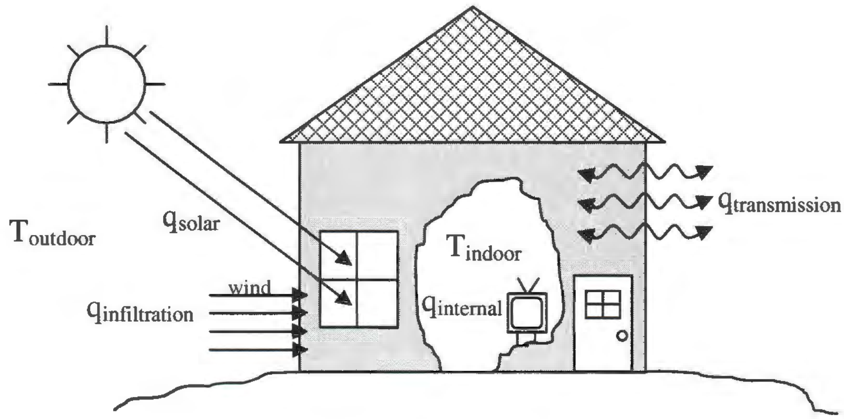

Figure 4.1: Energy flows to and from the home

For obtaining weather data for estimating hourly loads (i.e., transmission, infiltration and solar) to the home, Typical Meteorological Year (TMY2) data was used. TMY2 data is a set of hourly values of solar and meteorological elements for a location over a one-year period. This data is intended for computer simulations of solar energy conversion systems and building systems to facilitate performance comparisons of different system types, configurations, and locations in the United States. Because TMY2 data represents typical rather than extreme conditions, they are not suited for designing systems to meet the worst-case conditions occurring at a location (NREL, 1995).

From the previous sections of the study, it was determined that both a forced air distribution system and a radiant floor system will be used to heat the home. The lower level of the home will be heated by both systems, and it will be necessary to know the heating loads for each level and garage separately to determine the amount of energy demanded by each system. Therefore, the following analysis for heating was performed by estimating the hourly heating loads for the upper level, lower level and garage, separately. With these results, the hourly demand on each system and hourly cost of each system in the heating mode was determined.

The cooling of the home will be supplied by the forced air distribution system exclusively. Therefore, the analysis of the cooling loads was performed by determining the upper and lower level hourly heat gain together and the garage was excluded since it will not be cooled. With these results, the respective hourly costs for cooling the home with the forced air distribution were determined.

Transmission Heat Gain and Loss

Transmission heat transfer to and from each part of the home is driven by conduction and will occur when there is a temperature difference between the indoor and outdoor air. For the hourly simulation of transmission heat transfer, outdoor-air dry-bulb temperatures from the TMY2 data were used. Also, constant indoor-air dry-bulb temperatures were assumed; the indoor air temperature for the upper and lower levels for heating was assumed to be 68°F and 50°F for the garage, respectively. Further, an indoor air temperature of 75°F for the upper and lower levels was assumed for the cooling mode. The hourly transmission heat transfer to and from the home through the walls, windows, etc. are assumed to be one-dimensional conduction heat transfer and were estimated by

$$\text{Heating: } \dot{q}_{transmission,\,i} = UA_{conduction}\left(T_{indoor\ air} - T_{outdoor\ air,i}\right) \tag{4.1}$$

$$\text{Cooling: } \dot{q}_{transmission,\,i} = UA_{conduction}\left(T_{outdoor\ air,i} - T_{indoor\ air}\right) \tag{4.2}$$

where the subscript i represents each hour of the year.

The $UA_{conduction}$ term is the combined thermal transmittance of the home and represents the amount of energy conducted through the structure per unit time per unit of temperature, e.g., Btu's per hour per degree Fahrenheit [Btu/(hr*F)]. The $UA_{conduction}$ value is found by

$$\begin{aligned} UA_{effective} = {} & U_{wall1}A_{wall1} + U_{wall2}A_{wall2} + \dots\dots\dots + U_{wall\,i}A_{wall\,i} \\ & + U_{window1}A_{window1} + U_{window2}A_{window2} + \dots\dots\dots + U_{window\,i}A_{window\,i} \\ & + U_{door1}A_{door1} + U_{door2}A_{door2} + \dots\dots\dots + U_{door\,i}A_{door\,i} \\ & + U_{x1}A_{x1} + U_{x2}A_{x2} + \dots\dots\dots + U_{xi}A_{xi} \end{aligned} \tag{4.3}$$

The subscript x in Equation 4.3 represents any other structural component that would be subjected to conduction energy transfer, e.g., ceiling, exposed floor, etc. This value was determined using the blueprints for the home and for both summer and winter conditions. The $UA_{conduction}$ value calculated for a heating condition was 0.3606 MBtu/(hr*F) for the upper level, 0.3011 MBtu/(hr*F) for the lower level and 0.1695 MBtu/(hr*F) for the garage. The $UA_{conduction}$ value calculated for a cooling condition was 0.703 MBtu/(hr*F) for the entire home.

Infiltration Heat Gain and Loss

Infiltration is the uncontrolled movement of outdoor air into the building through cracks, windows, doors and other openings in the building envelope. The amount of infiltration occurring at a particular time can be mainly attributed to the structure tightness, wind speed, wind direction and outdoor air temperature. To estimate infiltration loads for each level and to the garage of the home, correlations developed by ASHRAE were used. These correlations estimate infiltration air changes per hour (ACH) under particular conditions. The air change per hour value represents the number of times the air in the home is completely replaced by outdoor air via infiltration.

Table 4.1: Winter air exchange rates

Class	Outdoor Air Temperature (F)									
	50	40	30	20	10	0	-10	-20	-30	-40
Tight	0.41	0.43	0.45	0.47	0.49	0.51	0.53	0.55	0.57	0.59
Medium	0.69	0.73	0.77	0.81	0.85	0.89	0.93	0.97	1	1.05
Loose	1.11	1.15	1.2	1.23	1.27	1.3	1.35	1.4	1.43	1.47

Source: ASHRAE, 2001

The winter design air exchange rate values assume an outdoor air wind speed of 15 miles per hour (ASHRAE, 2001). The values in Table 4.1 were plotted and lines were fit to the data to obtain the ACH as a function of outdoor air temperature in equation form.

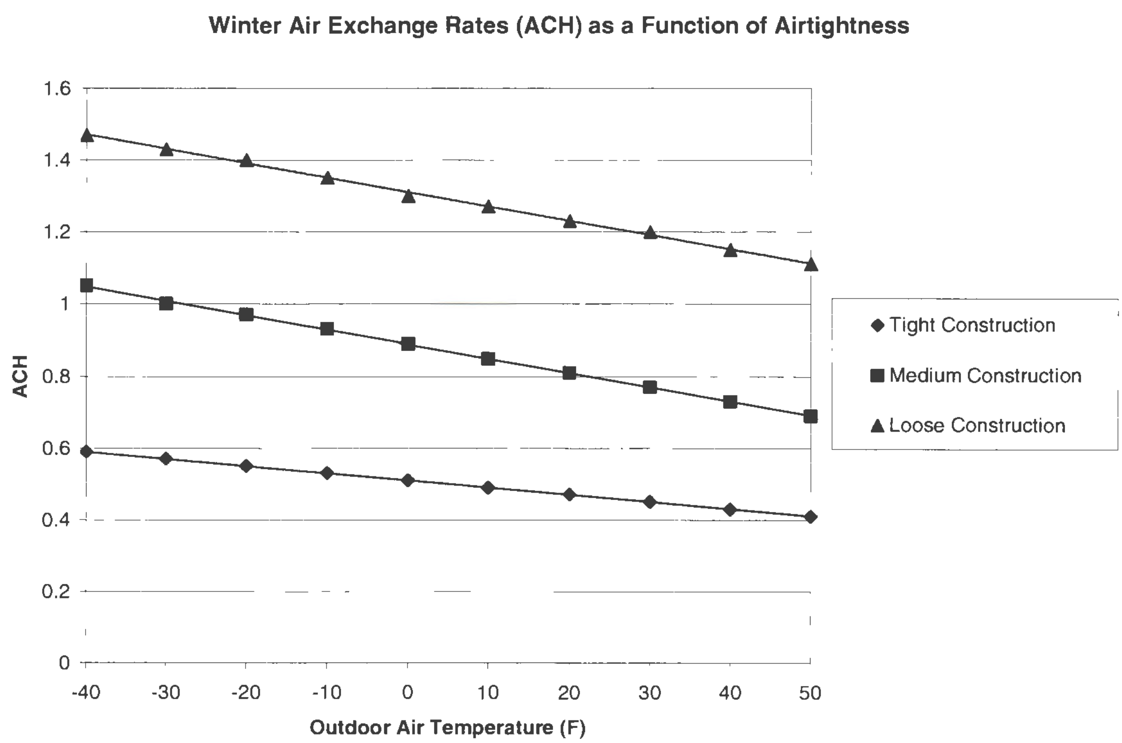

Figure 4.2: Winter design ACH values

The fitted equation for a tight construction home for ACH as a function of outdoor air temperature was found to be

$$ACH_{Winter,i} = -0.002(OAT_i) + 0.51 \tag{4.4}$$

with an R^2 value of 1.0.

Table 4.2: Summer air exchange rates

Class	Outdoor Air Design Temperature (F)					
	85	90	95	100	105	110
Tight	0.33	0.34	0.35	0.36	0.37	0.38
Medium	0.46	0.48	0.5	0.52	0.54	0.56
Loose	0.68	0.7	0.72	0.74	0.76	0.78

Source: ASHRAE, 2001

The summer design exchange rate values assume an outdoor air wind speed of 7.5 miles per hour (AHSRAE, 2001). The values in Table 4.2 were plotted and lines were fit to the data to obtain the ACH as a function of outdoor air temperature in equation form.

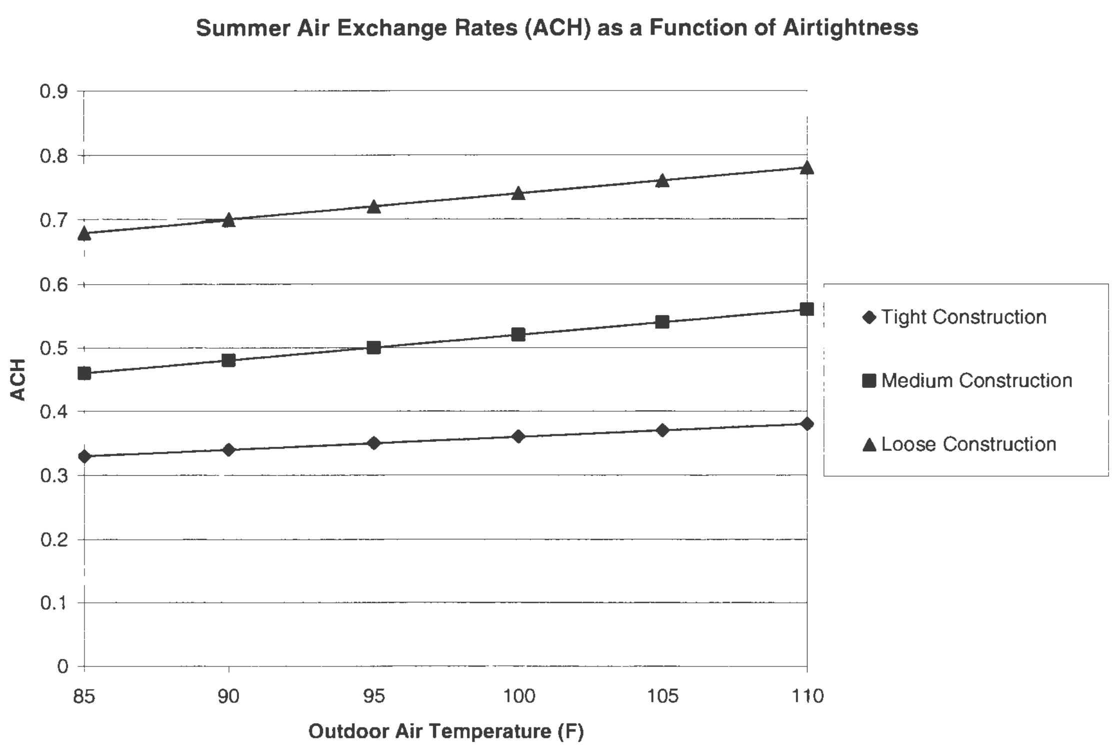

Figure 4.3: Summer ACH values

The home was assumed to be a tight construction. The fitted equation for a tight construction home for ACH as a function of outdoor air temperature was found to be

$$ACH_{Summer,\,i} = 0.002(OAT_i) + 0.16 \tag{4.5}$$

with an R^2 value of 1.0.

For simulating the infiltration for each hour of the year, the ACH values were adjusted according to the wind speed and outdoor air temperature. The phenomenon of infiltration is driven by a pressure difference across the building shell, which is caused by the pressure exerted by the wind on the exterior walls of the home. These pressures arise from the conversion of the kinetic energy of the wind into a pressure rise against the wall as the air is brought to rest, or stagnation (Munson, Young, & Okiishi, 2002). A diagram of the pressure rise on the exterior walls of the home assuming that the indoor air pressure is equal to the atmospheric air pressure can be seen in Figure 4.3.

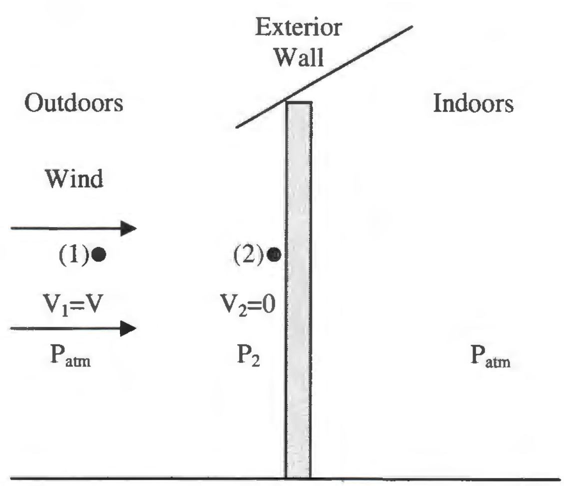

Figure 4.4: Diagram of wind pressure on exterior walls of the home

The pressure rise from atmospheric pressure to P_2 can be estimated by the Bernoulli equation assuming that the air acts as an incompressible fluid at the relatively low velocities that could typically be expected with wind.

$$P_2 = P_{atm} + \frac{V_{air}^2}{2}\rho_{air} \tag{4.6}$$

Rewriting this equation in terms of a change in pressure across the partition assuming that the indoor air pressure is equal to atmospheric pressure yields

$$\Delta P = P_2 - P_{atm} = \frac{V_{air}^2}{2}\rho_{air} \tag{4.7}$$

Thus, the pressure difference across the exterior wall is proportional to the velocity squared assuming that the wind is perpendicularly striking the wall.

$$\Delta P \propto V_{air}^2 \tag{4.8}$$

Assuming that all doors and windows are closed, the air will be entering the home primarily through cracks in the walls. These cracks in the home were modeled as a single orifice, seen in Figure 4.5.

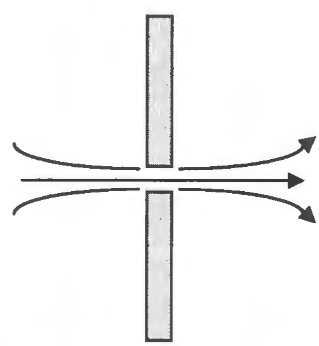

Figure 4.5: Diagram of air infiltrating through a wall crack

The volumetric flow rate of air entering the home through the cracks can be represented as

$$Q = VA \tag{4.9}$$

The pressure difference across the wall or orifice can be represented using the discharge coefficient. In the turbulent regime this non-dimensional parameter is nearly constant and is defined as

$$C = \frac{\Delta P}{\frac{1}{2}\rho_{air} V_{air}^2} \tag{4.10}$$

Solving for the velocity of the air through the crack and substituting into the volumetric flow rate of air relationship, Equation 4.11 is obtained.

$$Q = A\sqrt{\frac{\Delta P}{\frac{1}{2}\rho C}} \tag{4.11}$$

Substituting the Bernoulli equation in for change in pressure, Equation 4.12 is obtained.

$$Q = A\sqrt{\frac{\frac{\rho V_{air}^2}{2}}{\frac{1}{2}\rho\, C}} = \frac{AV_{air}}{\sqrt{C}} \tag{4.12}$$

Thus, under these assumptions, the volumetric flow rate of air infiltrating the structure and thus, ACH value, is directly proportional to the wind velocity.

The hourly infiltration loads to the home were determined for the upper level, lower level, and garage, separately. This is due in part to the fact that less infiltration will occur in the lower level than the upper level and garage because part of the lower level is below grade; consequently, no wind will hit the underground walls.

To estimate the hourly ACH for the upper level and garage, hourly wind values and outdoor-air dry-bulb temperatures were used from the TMY2 data. The ACH values were estimated using Equations 4.4 and 4.5 and prorated according to the wind speed at the particular hour. For example, in the summer or cooling mode the ACH value was estimated to be 1.02 for a "tight" class home and an outdoor-air dry-bulb temperature of 90°F and wind speed of 22.5 miles per hour.

The estimation of the ACH of the lower level was made considering that part of the lower level is below grade. The lower level of the case study residence has exposed walls only on the southeast and southwest sides. Therefore, it was assumed that the ACH of the lower level is zero when the wind was blowing between the directions north of east and west of southwest. During the hours when the wind is blowing at all other directions, the ACH of the lower level was estimated to be the same as for the upper level and garage.

Once the hourly ACH values were estimated for each part of the home, the energy required to heat or cool the air in Btu's per hour was determined by

$$\text{Heating:} \quad \dot{q}_{infiltration,i} = \frac{1.08 * ACH_i * (Zone\ Volume) * (T_{indoor\,air} - T_{outdoor\,air,i})}{60}\left(\frac{WS_i}{15}\right) \tag{4.13}$$

$$\text{Cooling:} \quad \dot{q}_{infiltration,i} = \frac{1.1 * ACH_i * (Zone\ Volume) * (T_{outdoor\,air,i} - T_{indoor\,air})}{60}\left(\frac{WS_i}{7.5}\right) \tag{4.14}$$

where the zone volume represents the volume of the upper level, lower level or garage. It should be noted that the volume is in cubic feet and the temperatures are in degrees Fahrenheit.

Solar Heat Gain

The total solar radiation striking the home can be split into two components, beam and diffuse radiation. The beam radiation is the solar radiation striking the home from the sun without having been scattered by the atmosphere and it is often referred to as direct solar radiation (Duffy & Beckman 1991). The diffuse radiation is the solar radiation striking the home from the sun after its direction has been changed due to scattering by the atmosphere, and it is often referred to as solar sky radiation (Duffy & Beckman 1991). Thus, the total irradiance, which is the rate at which radiant energy is incident on a surface per unit area of surface, is the sum of the beam and diffuse radiation rates per unit area.

The TMY2 data gives solar values for a horizontal surface; therefore, the amount of solar radiation incident on a vertical surface, e.g., the windows, must be found. The hourly beam radiation incident on a tilted surface can be determined by

$$I_{b.T} = I_b R_b \tag{4.15}$$

The parameter R_b represents the ratio of beam radiation on a tilted surface to that on a horizontal surface for a particular hour and was determined by

$$R_b = \frac{I_{b.T}}{I_b} = \frac{\cos(\theta)_i}{\cos(\theta_Z)_i} \tag{4.16}$$

The angle of incidence, θ, is the angle between the beam radiation on a surface and the normal to that surface; whereas, the zenith angle, θ_Z, is the angle between the vertical and the line to the sun. The angle of incidence and the zenith angle were determined by

$$\begin{aligned}\cos(\theta)_i = {} & \sin(\delta)_i \sin(\phi)\cos(\beta) \\ & -\sin(\delta)_i \cos(\phi)\sin(\beta)\cos(\gamma) \\ & +\cos(\delta)_i \cos(\phi)\cos(\beta)\cos(\omega)_i \\ & +\cos(\delta)_i \sin(\phi)\sin(\beta)\cos(\gamma)\cos(\omega)_i \\ & +\cos(\delta)_i \sin(\beta)\sin(\gamma)\sin(\omega)_i\end{aligned} \tag{4.17}$$

$$\cos(\theta_Z)_i = \cos(\phi)\cos(\delta)_i \cos(\omega)_i + \sin(\phi)\sin(\delta)_i \tag{4.18}$$

The Greek term omega, ω, is the angular displacement of the sun east or west of the local meridian, and it is due to the rotation of the earth on its axis at 15 degrees per hour. The angular displacement value is calculated in radians where the morning values are negative and the afternoon values are positive by

$$\omega_i = (Solar\ \ Hour - 12) * 15^o \tag{4.19}$$

The solar time, which is used in all of the sun-angle relationships, is the time based upon the rotation of the earth around the sun. When the sun is the highest in the sky, it is solar noon. The difference between standard time and solar time in minutes can be determined by

$$\text{Solar Time} - \text{Standard Time} = 4\ (L_{ST} - L_{loc}) + E \tag{4.20}$$

where the equation of time, E, is

$$E = 229.2 \begin{pmatrix} 0.000075 \\ +0.001868\cos B \\ -0.032077\sin B \\ -0.014615\cos 2B \\ -0.04089\sin 2B \end{pmatrix} \tag{4.21}$$

and B is

$$B_i = (n-1)\frac{360}{365} \tag{4.22}$$

The standard meridians, L_{ST}, for the continental U.S. time zones are: Eastern, 75°W; Central, 90°W; Mountain, 105°W; and Pacific, 120°W. The declination, δ, is the angular position of the sun at solar noon with respect to the plane of the equator with north being positive.

$$\delta_i = 23.45 * \sin\left(360\frac{284+n}{365}\right) \tag{4.23}$$

The latitude, φ, for the case study home was determined using a global positioning system (GPS) to be 42 degrees. In addition, all windows of the home are oriented vertically; therefore, the slope of each surface, β, is 90 degrees. The surface azimuth angle, γ, is the deviation of the projection on a horizontal plane of the normal to the surface from the local meridian, with zero due south, east negative, and west positive (Duffy & Beckman 1991). This value was also determined using a GPS device. The back of the home faces southwest, resulting in a surface azimuth of 45 degrees.

Any beam insolations that were calculated and corresponded to a negative $\cos(\theta)$ or $\cos(\theta_z)$ were set to zero. This was done because negative values for these parameters indicate that either the sun has gone under the horizon or the sun is behind the surface so that there is no incident beam radiation on the window face.

The hourly diffuse radiation on a tilted surface was determined by

$$I_{d,T} = I_d\left(\frac{1+\cos(\beta)}{2}\right) + I\rho_{g,i}\left(\frac{1-\cos(\beta)}{2}\right) \tag{4.24}$$

The ground reflectance, ρ_g, was estimated assuming two conditions, grass and snow covered. The hourly snow depth values from the TMY2 data were used to determine if the ground was snow covered. If the snow depth for the hour was found to be greater than zero, then the ground reflectance was assumed to be 0.7; otherwise, the ground was assumed to be grass covered with an approximate ground reflectance value of 0.25. All of the hourly diffuse values that corresponded to a negative $\cos(\theta_z)$ value were set to zero.

The total amount of solar energy entering the home through the glass is a function of the solar heat gain coefficient (SHGC) of the glazing and the type of interior shading. The

SHGC is the fraction of incident solar radiation admitted through a window, both directly transmitted and absorbed, and then subsequently released into the home (ASHRAE, 2001). The SHGC for a window is a function of the angle of incidence of the solar radiation. For this study, the program WINDOW 5.1 was used to determine the SHGC of the windows in the home at various angles of incidence. Table 4.3 shows the calculated SHGC at various angles of incidences.

Table 4.3: SHGC for case study residence glass

Angle of Incidence	SHGC
0	0.469
10	0.471
20	0.466
30	0.458
40	0.448
50	0.427
60	0.381
70	0.291
80	0.152
90	0
Hemis.	0.399

These values were then plotted and lines were fit to represent the SHGC at various angles of incidence in equation form.

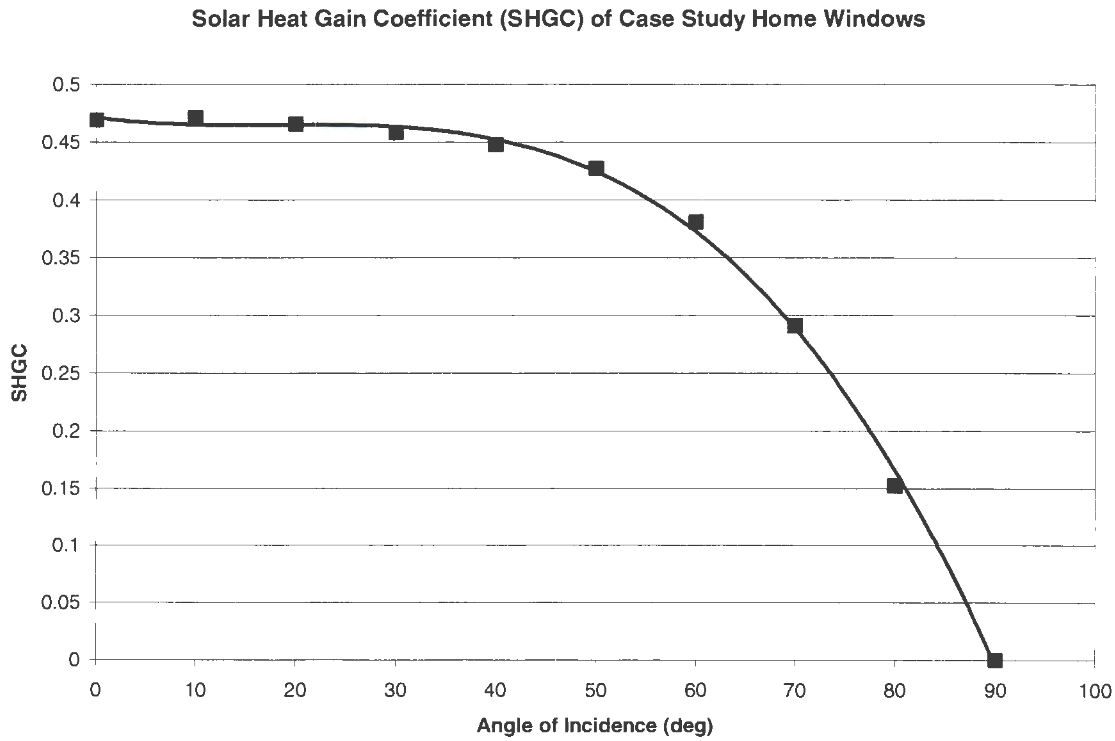

Figure 4.6: SHGC as a function of angle of incidence

The SHGC as a function of angle of incidence was found to be

$$SHGC_i = -0.00000126(\theta)_i^3 + 0.00006681(\theta)_i^2 - 0.00113432(\theta)_i + 0.47135 \qquad (4.25)$$

with an R^2 value of 0.9984.

Interior shading devices used in a home provide privacy, aesthetic effects and solar heat gain control (ASHRAE, 2001). The effectiveness of the interior shading device in controlling the solar heat gain depends on its ability to reflect incoming solar radiation back through the fenestration before it can be absorbed and converted into heat within the building (ASHRAE, 2001). This effectiveness can be estimated using the solar attenuation coefficient (IAC). The IAC represents the fraction of heat flow that enters the room through the shading device (ASHRAE, 2001). These values are a function of the type of window, type of shade, and color of the shade. It was assumed that light colored Venetian blinds will be used in the residence, thus, resulting in an estimated IAC of 0.66 (ASHRAE, 2001).

Once the total irradiance, solar heat gain coefficient, and interior shading was determined for the glass, the amount of heat entering the home and contributing to the heating and cooling loads was estimated. The hourly amount of solar heat entering the building through the glass and shading device can therefore be estimated by

$$\dot{q}_{solar,i} = \left[I_{b,T}\left(SHGC_{\theta}\right)_i + I_d\left(SHGC_{Hemis}\right)\right]IAC \tag{4.26}$$

Internal Heat Gain

The internal gain to the home was estimated considering occupancy, lighting, appliances and laundering machines. The internal gain was estimated on an hourly basis for the upper and lower levels of the home throughout all hours of the year. Any internal gain that may occur in the garage was assumed to be negligible.

People will create an internal load in the home any time they occupy the home. To estimate this hourly load, the following variables were considered: number of occupants, times the occupants are in the home, where the occupants are located in the home, and how much heat will be created by each occupant. The hourly internal load due to the occupants will be the product of the number of occupants and the heat rate per occupant when they occupy the particular zone of the home. It was assumed that there will be two adult occupants at 230 Btu's per hour of sensible heat in the home (ASHRAE, 2001). Also, it was assumed that the occupants will be in the home from 6:00 pm to 8:00 am and in the lower level between 8:00 pm and 10:00 pm each day.

The appliances included in the model are two refrigerators and one large freezer and were assumed to operate all hours of the year. One refrigerator was included in the upper level and one refrigerator and large freezer was included in the lower level. The refrigerators were assumed to emit 900 Btu's per hour each and the freezer was assumed to emit 2,760 Btu's per hour (ASHRAE, 2001).

Also, the heat generated by doing laundry was considered. It was assumed that the laundry machines will operate on each Saturday of the year for two hours, 9:00 am to 11:00 am. The heat emitted into the home by the washer and dryer were estimated to be approximately 4,100 Btu's per hour (ASHRAE, 2001).

Lastly, the home lighting was considered. Heat gained to the home from lights was assumed to occur during times when the occupants are home and awake and also when there is no solar gain. The number of lights used was assumed to be eight, each at one-hundred watts. A summary of all inputs to the internal gain model can be seen in Table 4.4.

Table 4.4: Internal gain model inputs

HOMEOWNER PROFILE - INTERNAL GAINS	
Heat load per occupant (Btu/hr)	230 (SH)
Number of occupants	2
Typical time occupants go to lower level of home	20
Typical time occupants return to upper level of home	22
Home occupied from (PM TIME)	18
Home occupied to (AM TIME)	8
Constant load from upper level appliances (Btu/hr)	900
Constant load from lower level appliances (Btu/hr)	3660
Load from lights on upper level when no solar (Btu/hr)	2,730
Load from lights on lower level when no solar (Btu/hr)	2,730
Load from clothes washer and dryer (Btu/hr)	4,094
Day laundry is done	Sat
Hour of laundering	9
Hour of laundering	10
Typical time occupants go to sleep	22
Typical time occupants awake from sleep	6
If solar gain is greater than X Btuh then no lights	0

The hourly internal gain to the home was estimated by

$$\dot{q}_{internal,\,i} = \dot{q}_{occupancy,\,i} + \dot{q}_{lights,\,i} + \dot{q}_{appliances,\,i} + \dot{q}_{laundry,\,i} \qquad (4.27)$$

Monthly and Annual Heating Loads

The hourly heat losses from the home were estimated for the upper level, lower level, and garage separately by

$$\dot{q}_{heating,i} = \left[\dot{q}_{transmission,i} + \dot{q}_{infiltration,i} - \dot{q}_{solar,i} - \dot{q}_{internal,i}\right]^{+} \tag{4.28}$$

where the + sign represents only positive hourly values were used in the monthly and annual energy estimations. A negative value for an hourly heating load would represent a net heat gain to the home for that hour and heating from the HVAC system would not need condition the indoor air.

The total monthly heat loss of the home is then the sum of the hourly upper level, lower level, and garage heat losses for each month. Similarly, the annual total heat loss of the home is the sum of the hourly upper level, lower level, and garage heat losses over the entire year.

Table 4.5: Monthly and annual estimated total home heat loss

Period	Upper Level Ht. Loss (MMBtuh)	Lower Level Ht. Loss (MMBtuh)	Garage Ht. Loss (MMBtuh)	Total Home Ht. Loss (MMBtuh)
January	16.19	11.16	6.31	33.66
February	10.81	7.31	3.97	22.09
March	7.35	4.58	2.31	14.23
April	3.76	1.94	0.91	6.61
May	1.08	0.31	0.07	1.46
June	0.32	0.04	0.00	0.37
July	0.12	0.01	0.00	0.14
August	0.30	0.03	0.00	0.34
September	1.09	0.30	0.08	1.48
October	3.53	1.77	0.70	6.00
November	7.29	4.45	1.99	13.73
December	12.50	8.17	4.32	25.00
Annual	64.35	40.08	20.67	125.11

Individual homeowners may turn on the heating system to their home at different times; for example, homeowner *x* may turn on the heating system when it is 50°F outdoors, and homeowner *y* may turn prefer to turn on the heat when it is 60°F outside. The hourly heating load was set to zero if the outdoor temperature was above the particular outdoor air temperature specified by the owners. Through conversations with the owners of the case study home it was determined that the heating system would most likely not be turned on if the outdoor air temperature was greater than 50°F. Table 4.6 shows the prorated energy uses of the home with the minimum outdoor air temperature criteria.

Table 4.6: Monthly and annual estimated total home heat loss with outdoor air temperature constraint

Period	Upper Level Ht. Loss (MMBtuh)	Lower Level Ht. Loss (MMBtuh)	Garage Ht. Loss (MMBtuh)	Total Home Ht. Loss (MMBtuh)
January	16.19	11.16	5.93	33.28
February	10.68	7.30	3.68	21.66
March	7.20	4.55	1.69	13.44
April	3.14	1.84	0.38	5.35
May	0.35	0.19	0.00	0.54
June	0.02	0.01	0.00	0.03
July	0.00	0.00	0.00	0.00
August	0.00	0.00	0.00	0.00
September	0.49	0.21	0.00	0.70
October	3.02	1.68	0.15	4.86
November	7.10	4.41	1.28	12.79
December	12.50	8.17	3.84	24.51
Annual	60.68	39.52	16.94	117.14

It can be seen that by including the minimum outdoor air temperature constraint the annual estimated heating load to the home on the HVAC system decreased by 6.37 percent.

Monthly and Annual Cooling Loads

The hourly heat gains to the upper and lower levels of the home were estimated by

$$\dot{q}_{cooling,i} = \left[\dot{q}_{transmission,i} + \dot{q}_{infiltration,i} + \dot{q}_{solar,i} + \dot{q}_{internal,i} \right]^{+} \tag{4.29}$$

where the + sign represents only positive hourly values were used in the monthly and annual energy estimations. A negative value for hourly heat gain would represent a net heat loss from the home for that hour conditioning the indoor air would be unnecessary. The hourly heat gains were then summed up for each month and annually and are tabulated in Table 4.7.

Table 4.7: Monthly and annual estimated total home heat gain

Period	Total Home Ht. Gain (MMBtuh)
January	0.24
February	0.30
March	1.57
April	3.19
May	5.61
June	8.93
July	12.51
August	10.09
September	5.70
October	3.35
November	0.87
December	0.27
Annual	52.65

Table 4.7 represents the heat gain to the home with no ventilation. To more accurately predict the amount of heat gain seen by the HVAC system, all heat gain values that occur at or below a particular outdoor air temperature are assumed to be zero, and the home will be assumed to be open to free ventilation. The owners expressed that they would most likely turn on the air conditioning when the outdoor air temperature reached 80^{o}F. Table 4.8 shows the prorated energy uses of the home with the maximum outdoor air temperature criteria.

It can be seen that by including the maximum outdoor air temperature constraint the annual estimated cooling load to the home and supplied by the HVAC system decreased by approximately a factor of three (285 percent).

Table 4.8: Monthly and annual estimated total home heat gain with outdoor air temperature constraint

Period	Total Home Ht. Gain (MMBtuh)
January	0.00
February	0.00
March	0.00
April	0.25
May	0.76
June	3.47
July	8.05
August	5.54
September	1.38
October	0.40
November	0.00
December	0.00
Annual	19.86

Forced Air System Operating Performance and Costs in the Heating Mode

The hourly demand on the forced air distribution system and radiant floor system must be determined separately so that the operating performance and costs of each can be found. To estimate the amount of heating supplied by each system on an hourly basis, the control scheme for each system must be determined. It was decided that the forced air heating supplied by the heat pump or furnace will be controlled by one thermostat located on the upper level. The radiant floor heating will be controlled by two thermostats, one on the lower level and one in the garage. The forced air heating will turn on when the upper level temperature drops below 68°F, and the radiant floor heating will turn on when the heating supply from the forced air system to the lower level is insufficient to maintain 68°F and/or when the garage temperature drops below 50°F.

To compromise between efficient heating with the heat pump and extra comfortable heating with the radiant floor to the lower level, it was decided that the radiant floor heating system would not be turned on until the outdoor air temperature has dropped to a low value. The owners of the case study home were consulted, and they determined that they would like the radiant floor heating in the lower level to turn on when the outdoor air temperature is near or below 25 degrees Fahrenheit. As a result, the owners will be able to heat the lower level efficiently with the heat pump most of the time, and enjoy the comfort of the radiant floor heat when the heat loss of the lower level is high causing uneven space temperatures.

Knowing that the owners would like the radiant floor heating to turn on in the lower level at about 25°F, the amount of energy required by the forced air system to the lower level can be estimated. The average load of the lower level at the specified outdoor air temperature was found by the ratio of the sum of each load for the lower level that corresponded to an outdoor air temperature between 24.8°F and 25.2°F to the number of times the outdoor air temperature fell within that range. Thus, the result is the amount of heat that must be supplied to the lower level so that no radiant floor heating takes place; for temperatures below this range, the radiant floor heating will be required.

Ground-Source Heat Pump

This section will describe the methods used to predict the hourly performance of the ground-source heat pump and costs. This evaluation will begin with estimating the operating characteristics of the heat pump as an ultimate function of the outdoor air temperature. Then, considering the hourly loads on the home, the hourly performance of the heat pump used in conjunction with the radiant floor will be estimated.

The capacity of the heat pump can be found if the following parameters are known: the air flow rate through the coil, entering water temperature, compressor speed, whether or not the heat pump is creating domestic hot water, and water flow rate. It was assumed that the air and water flow rates through the unit in high speed are 14 gpm and 2,000 cfm; the low speed conditions were assumed to be 11 gpm and 1,100 cfm. In addition, for this analysis it was assumed that the heat pump will not be creating hot water.

To determine the heat pump capacity in both high and low speeds as a function of entering water temperature, the manufacturer's specifications were used. The capacity information from the specifications was plotted and lines were fit to obtain an equation relating entering water temperature to heating capacity for the entire range of entering water temperatures.

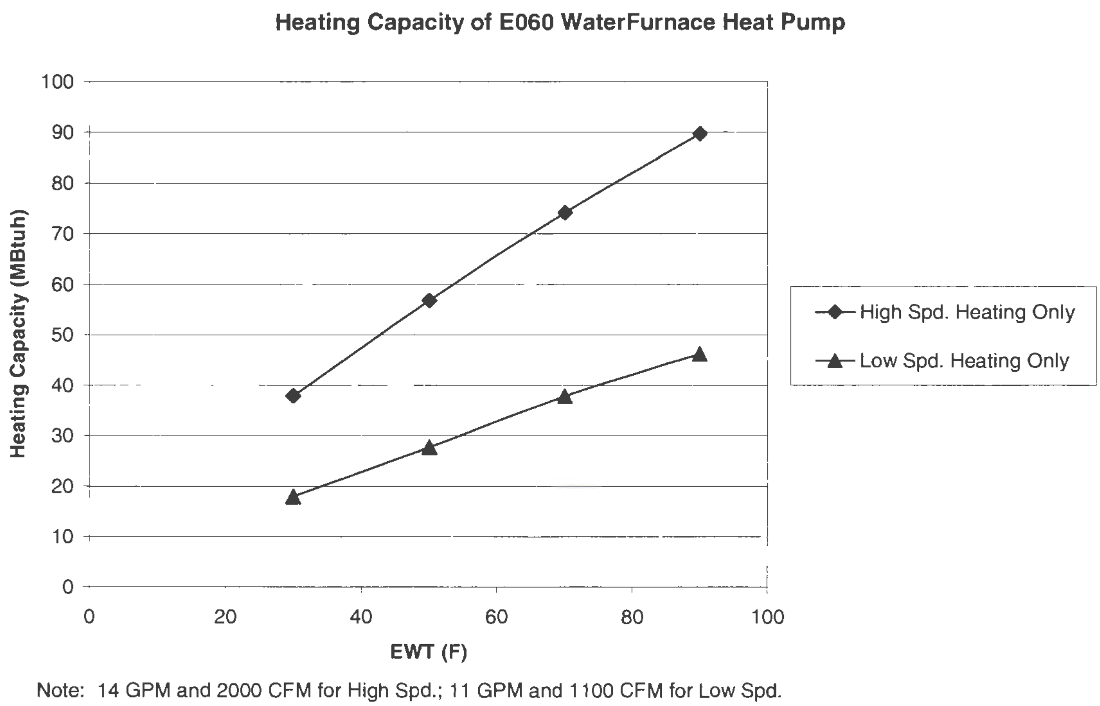

Figure 4.7: Heating capacity of WaterFurnace EO60 heat pump

The heating capacity of the heat pump in high speed was found to be

$$HC = -0.002125(EWT)^2 + 1.119(EWT) + 6.2225 \tag{4.30}$$

with an R^2 value of 0.999995.

The heating capacity of the heat pump in low speed was found to be

$$HC = -0.0008(EWT)^2 + 0.571(EWT) + 1.4963 \tag{4.31}$$

with an R^2 value of 0.9995.

The hourly entering water temperature can be roughly estimated using methods developed by the International Ground Source Heat Pump Association (IGSHPA). This method relates the minimum, maximum and mean outdoor air temperature, undisturbed deep ground temperature, and design entering water temperatures in the heating and cooling mode and is stated for heating as

$$EWT_{H,i} = DEWT_{MIN} + \left[\frac{EWT_{MEAN} - DEWT_{MIN}}{OAT_{MEAN} - OAT_{MIN}}\right]\left[OAT_i - OAT_{MIN}\right] \tag{4.32}$$

and for cooling as

$$EWT_{C,i} = EWT_{MEAN} + \left[\frac{DEWT_{MAX} - EWT_{MEAN}}{OAT_{MAX} - OAT_{MEAN}}\right]\left[OAT_i - OAT_{MEAN}\right] \tag{4.33}$$

where the EWT_{MEAN} is equal to the undisturbed deep ground temperature or the average annual outdoor air temperature plus two degrees Fahrenheit. A general plot of Equations 4.32 and 4.33 can be seen in Figure 4.8.

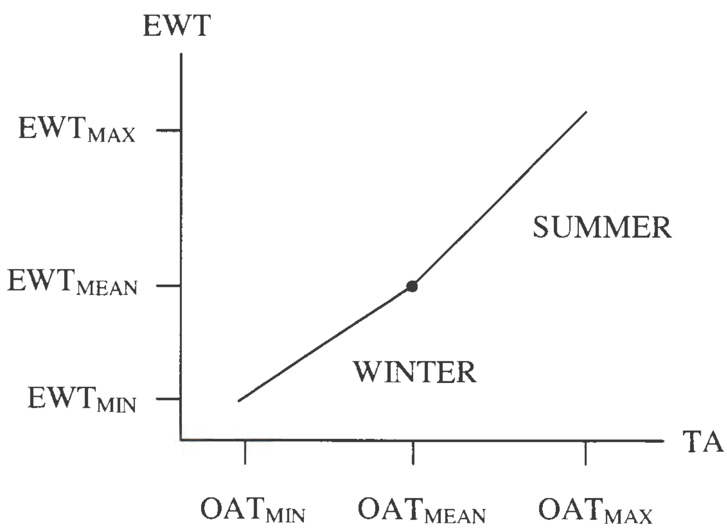

Figure 4.8: Plot of entering water temperature as a function of outdoor air temperature

It can be seen from Figure 4.8 that the entering water temperatures vary throughout the year between the design temperatures, which were chosen to be 35 and 75 degrees Fahrenheit. The estimated hourly entering water temperatures to the heat pump over the entire year were plotted and can be seen in Figure 4.9.

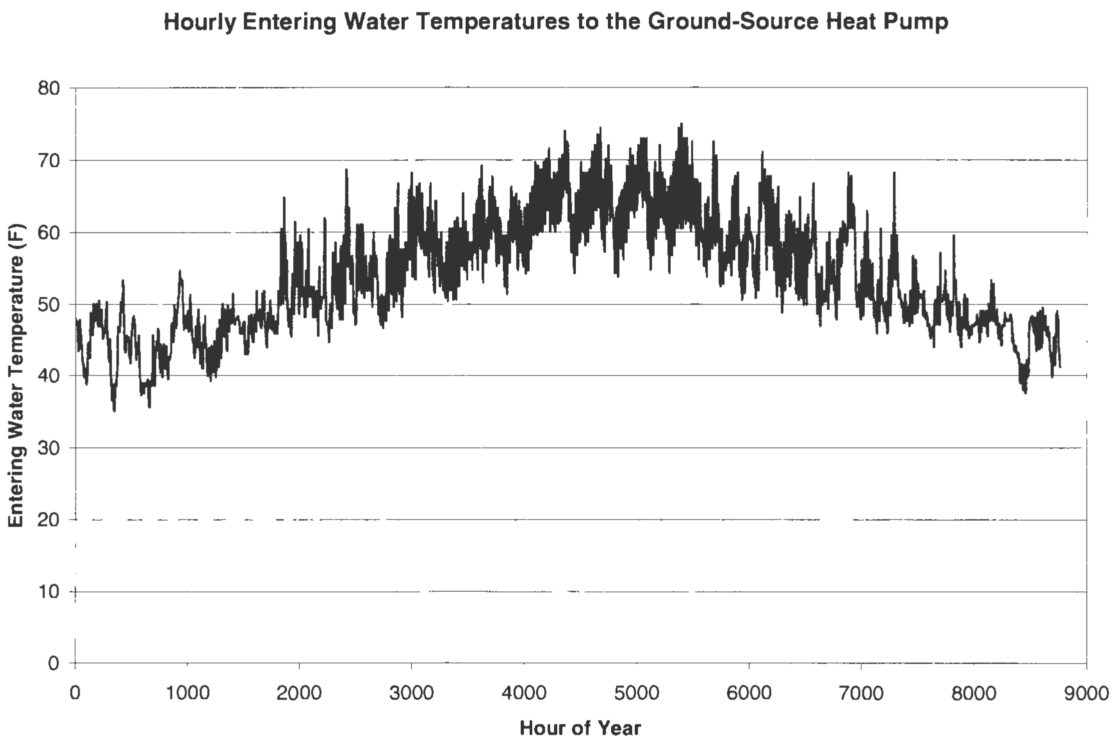

Figure 4.9: Hourly entering water temperature to heat pump

The system can now be simulated over the entire year using the hourly loads to the home determined in the prior sections. The heating supply from the forced air system to the lower level was chosen to be approximately 12.5 MBtuh for the heat pump on high speed. The volume flow rate of air to the lower level when the heat pump is on high speed can be determined by

$$Q = \frac{\dot{q}}{1.1\left(T_{LAT} - T_{indoor\,air}\right)} \tag{4.34}$$

where the load is in Btu's per hour.

The average leaving air temperature of the heat pump in heating mode on high speed was estimated to be approximately 92°F by examining the specifications of the heat pump at entering water temperatures between 50°F and 30 °F and at 2000 CFM. As a result, the volumetric flow rate of air to achieve 12.5 MBtu's per hour in the lower level was calculated to be approximately 470 CFM. This flow rate can be achieved by closing the diffusers in the basement until the total specified volume flow rate of air is obtained and can be measured by a flow hood.

The capacity of the heat pump in low speed is approximately half of the high speed capacity. Thus, the energy supplied to the lower level will also decrease by approximately half since the positions of the diffusers will remain constant. Consequently, it was assumed that 6.25 MBtu's per hour will be supplied to the lower level when the heat pump is on low speed.

The speed of the heat pump for each hour of the year was determined by evaluating the difference between the upper level load and heat pump capacity available to the upper level in both speeds. When the low speed capacity minus 6.25 MBtu's per hour was larger than the upper level load, the unit was assumed to be in low speed, otherwise, the heat pump was assumed to be operating in high speed. With the hourly speed and capacity of the heat pump known, the hourly theoretical run time fraction of the heat pump could be determined by

$$Theoretical\ Run\ Time\ Fraction_i = \frac{Upper\ Level\ Demand_i}{Heat\ Pump\ Capacity_i - Lower\ Level\ Supply_i} \qquad (4.35)$$

If the upper level load was greater than the heat pump capacity minus the lower level supply for the particular hour, the theoretical run time was set to equal unity, i.e., the heat pump will run the entire hour.

To estimate the actual run time of the heat pump for the hour, the partial load factor (PLF) was determined. The PLF is an adjustment factor that accounts for the loss in efficiency of the unit due to on-off cycling, which can be compensated for by increasing the run time. The hourly PLF was determined by

$$PLF_i = 1 - C_D\left[1 - \frac{Upper\ \ Level\ \ Demand_i}{Heat\ \ Pump\ \ Capacity_i - Lower\ \ Level\ \ Supply_i}\right] \tag{4.36}$$

The actual run time is the ratio of the theoretical run time to the partial load factor and is stated as

$$Actual\,Run\,Time_i = \frac{Theoretical Run\,Time_i}{Partial\,Load\,Factor_i} \tag{4.37}$$

The hourly heating supply of the heat pump for the upper and lower levels was determined by

$$\begin{aligned} Heat\ \ Pump\ \ Supply_i = {} & Upper\ \ Level\ \ Demand_i \\ & + (Theo.\ Run\ \ Fraction_i) * Lower\ \ Level\ \ Supply_i \end{aligned} \tag{4.38}$$

If the heat pump capacity on high speed for a particular hour minus the lower level supply for that hour is less than the upper level demand, the heat pump will meet the demand by using the electric resistance auxiliary heat. The amount of electric resistance auxiliary heat in kWh that will be required of the heat pump for each hour can be determined by

$$Ele.\ Resis.\ Aux._i = \frac{Upper\ \ Lev.\ Dmnd_i - (Heat\ \ Pump.\ Cap._i - Lower\ \ Lev.\ Supply_i)}{3.412} \tag{4.39}$$

where the upper level demand, heat pump capacity, and lower level supply are in MBtu's per hour.

The total electrical input to the heat pump can be found by

$$Heat\ Pump\ Tot.\ Ele.\ Input_i = Actual\ Run\ Fract._i * (Unit\ Input_i + Ele.\ Resis.\ Aux._i) \quad (4.40)$$

The electrical input of the heat pump in heating mode was determined in a similar manner as for the capacity and can be seen in Figure 4.10.

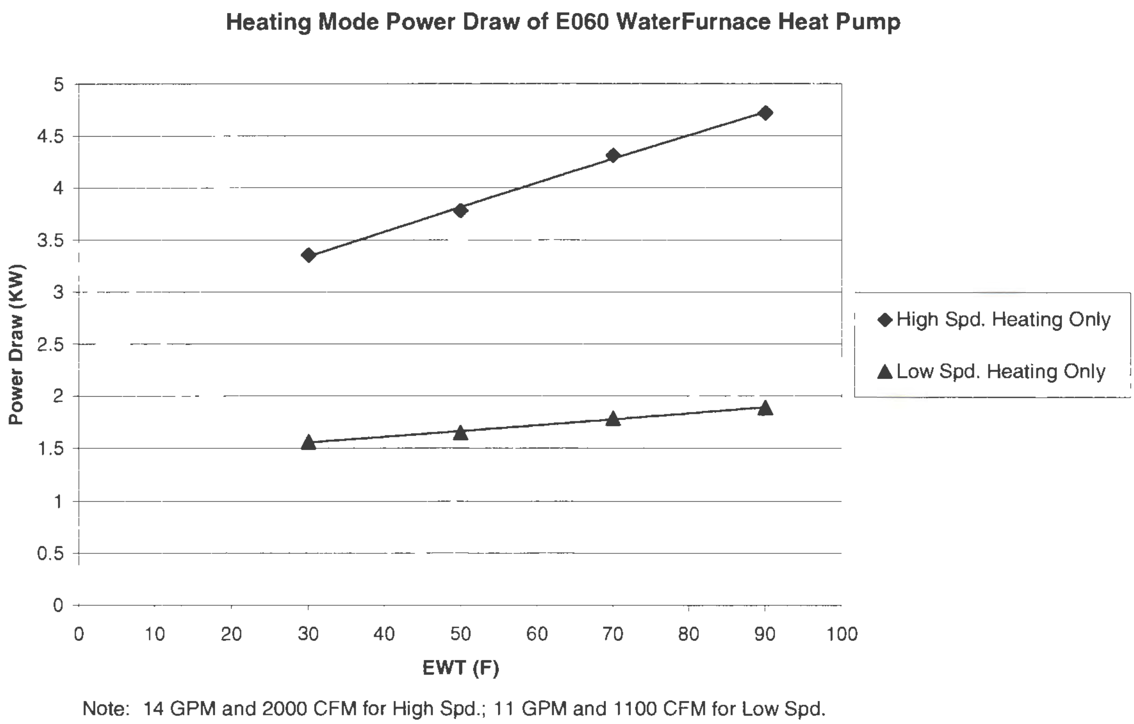

Figure 4.10: Power draw of heat pump in heating mode

The power draw of the heat pump in the heating mode on high speed was found to be

$$PD = -0.000012(EWT)^2 + 0.024700(EWT) + 2.60925 \quad (4.41)$$

with an R^2 value of 0.9978.

The power draw of the heat pump in the heating mode on low speed was found to be

$$PD = 0.000006(EWT)^2 + 0.0049(EWT) + 1.402875 \quad (4.42)$$

with an R^2 value of 0.9937.

Since the heat pump is controlled by only one thermostat, there may be times where the lower level is being supplied too much air. This energy was determined as the difference of the total supply and the total home demand and can be seen in Table 4.9. A summary of the estimated monthly and annual heating supply of the forced air system, radiant floor system, and both can be seen in Table 4.9.

Table 4.9: Monthly and annual heating supply of HVAC system using ground-source heat pump for forced air distribution system

Time Period	Air Side Upper and Lower Level Supply (MMBtuh)	Water Side Garage Supply (MMBtuh)	Water Side Lower Lev. Supply (MMBtuh)	Total Air and Water Supply (MMBtuh)	Total Extra Heat to LL (MMBtuh)
January	21.42	5.93	6.02	33.37	0.092
February	14.18	3.68	3.87	21.72	0.070
March	9.39	1.69	2.43	13.50	0.067
April	4.06	0.38	0.96	5.40	0.051
May	0.45	0	0.09	0.54	0.002
June	0.03	0	0	0.03	0
July	0	0	0	0	0
August	0	0	0	0	0
September	0.63	0	0.08	0.70	0.003
October	3.91	0.15	0.84	4.90	0.048
November	9.23	1.28	2.34	12.86	0.068
December	16.52	3.84	4.21	24.57	0.059
Annual	79.82	16.94	20.84	117.60	0.461

Using an electric rate of $0.035 dollars per kWh for heating, the hourly operating costs were determined by

$$Heating\ Cost_i = \left[(Actual\ Run\ Time\ Fraction)_i * PD_i + Aux.\ Heat_i\right] * Electric\ Rate \quad (4.43)$$

A summary of the monthly and annual heating costs of the heat pump can be seen in Table 4.10.

Table 4.10: Monthly and annual heating costs of ground-source heat pump

Time Period	Total Heat Pump Cost ($)
January	$62
February	$37
March	$23
April	$10
May	$1
June	$0
July	$0
August	$0
September	$2
October	$9
November	$22
December	$42
Annual	$208

Gas-Fired Furnace

It was determined that 12.5 MBtuh will be supplied to the lower level by the forced air system in order to restrict the use of the radiant floor heating. Often the furnaces used for residential applications are single speed (i.e., on or off). Thus, the heating supply to the lower level by the furnace would be approximately 12.5 MBtuh whenever the upper level calls for heating. Again, Equation 4.34 was used to determine the volumetric flow rate of air to the lower level to achieve 12.5 MBtuh. The leaving air temperature of the furnace was assumed to be 105^{o}F, resulting in approximately 310 cubic feet per minute of air.

The capacity of the furnace that was recommended by the contractor is 120 MBtu's per hour as seen in the quotation in Appendix D. When the furnace is operating it will supply 108 MBtu's per hour to the upper level after accounting for the unit's efficiency. The theoretical run time fraction of the furnace was determined using Equation 4.35. The hourly heating supply of the furnace for the upper and lower levels was determined by

$$\begin{aligned} Furnace\ Supply_i &= Upper\ Level\ Demand_i \\ &+ (Theo.\ Run\ Fraction_i) * Lower\ Level\ Supply_i \end{aligned} \tag{4.44}$$

The hourly heating demand of the radiant floor system for the lower level was determined. The hourly heating demand of the radiant floor system for the garage is equal to the hourly heat loss in the garage.

A summary of the estimated monthly and annual heating supply of the furnace and radiant floor and both can be seen in Table 4.11.

Table 4.11: Monthly and annual heating supply of HVAC system using liquid propane furnace for the forced air distribution system

Time Period	Air Side Demand (MMBtuh)	Water Side Garage Demand (MMBtuh)	Water Side Lower Lev. Demand (MMBtuh)	Total Air and Water Demand (MMBtuh)	Total Extra Heat to LL (MMBtuh)
January	17.88	5.93	9.49	33.29	0.0114
February	11.79	3.68	6.20	21.67	0.0123
March	7.95	1.69	3.82	13.45	0.0140
April	3.46	0.38	1.52	5.36	0.0116
May	0.39	0.00	0.15	0.54	0.0004
June	0.02	0.00	0.01	0.03	0.0000
July	0.00	0.00	0.00	0.00	0.0000
August	0.00	0.00	0.00	0.00	0.0000
September	0.54	0.00	0.16	0.70	0.0010
October	3.34	0.15	1.37	4.87	0.0128
November	7.83	1.28	3.69	12.80	0.0124
December	13.80	3.84	6.88	24.51	0.0067
Annual	67.00	16.94	33.28	117.22	0.0826

To estimate the monthly and annual cost of operating a gas-fired furnace, the efficiency of the unit was assumed to be constant. Thus, the cost per unit energy can be determined for the furnace by

$$\left(\frac{\$}{Gallon}\right)\left(\frac{Gallon}{91{,}600\,Btu}\right)\left(\frac{1{,}000{,}000\,Btu}{MMBtu_{Heating}}\right)\left(\frac{1}{A.F.U.E.}\right)=\frac{\$}{MMBtu_{Heating}} \qquad (4.45)$$

The manufacturer of the furnace states that the A.F.U.E. of the unit is 92 percent. Using the monthly and annual energy demand and cost per unit energy, the monthly and annual costs were estimated.

Table 4.12: Monthly and annual heating cost of liquid propane furnace

Time Period	Total Furnace Cost ($)
January	$243
February	$161
March	$108
April	$47
May	$5
June	$0
July	$0
August	$0
September	$7
October	$45
November	$107
December	$188
Annual	$912

Radiant Floor Operating Performance and Costs in the Heating Mode

Within this section, the annual operating costs for the radiant floor heating system will be estimated. The options for supplying hydronic heating for the radiant floor system consists of either a water-to-water heat pump, electric boiler, or a liquid propane boiler. Each alternative system will be assumed to operate at a constant efficiency. First, the cost per MMBtu's for each type of equipment will be estimated; the electric rate assumed was $0.035 dollars per kilowatt hour. Then, using the total demanded energy for the radiant floor system, the monthly and annual costs were estimated. The demand on the radiant floor system used in conjunction with a ground-source heat pump or gas fired furnace can be seen in Table 4.13.

Table 4.13: Monthly and annual demand on radiant floor system using either a ground-source heat pump or liquid propane furnace for the forced air distribution system

Time Period	Water Side Garage Demand (MMBtu)	Water Side Lower Lev. Demand (MMBtu)	
		GSHP	Gas Fired Furnace
January	5.93	6.02	9.49
February	3.68	3.87	6.20
March	1.69	2.43	3.82
April	0.38	0.96	1.52
May	0.00	0.09	0.15
June	0.00	0.00	0.01
July	0.00	0.00	0.00
August	0.00	0.00	0.00
September	0.00	0.08	0.16
October	0.15	0.84	1.37
November	1.28	2.34	3.69
December	3.84	4.21	6.88
Annual	16.94	20.84	33.28

The cost per MMBtu's for the heat pump and electric boiler were determined by

$$\left(\frac{\$}{kWh}\right)\left(\frac{kW}{1000\,W}\right)\left(\frac{W\,h}{3.412\,Btu}\right)\left(\frac{1{,}000{,}000\,Btu}{MMBtu_{Heating}}\right)\left(\frac{1}{COP}\right)=\frac{\$}{MMBtu_{Heating}} \tag{4.46}$$

The COP of the water-to-water heat pump and electric boiler was assumed to be a constant value of 3.8 and 0.95, respectively. As a result, the costs per MMBtu's using the heat pump and electric boiler were found to be \$2.70 and \$10.80 dollars, respectively. The cost per MMBtu's using the liquid propane boiler was determined to be \$13.53 dollars using Equation 4.46 and assuming \$1.14 dollars per gallon.

The estimated monthly and annual heating costs for each system were calculated by

$$\left(\frac{\$}{MMBtu_{Heating}}\right)\left(\frac{MMBtu_{Heating}}{Period}\right)=\frac{\$_{Heating}}{Period} \tag{4.47}$$

where period represents the time period of either a month or year.

The estimated cost of using an electric boiler for the radiant floor heating with either the heat pump or furnace can be seen in Table 4.14.

Table 4.14: Monthly and annual cost for electric boiler using either a ground-source heat pump or liquid propane furnace for the forced air distribution system

Time Period	Total Electric Boiler Cost ($)	
	GSHP	Gas Fired Furnace
January	$129	$165
February	$82	$106
March	$44	$59
April	$14	$20
May	$1	$2
June	$0	$0
July	$0	$0
August	$0	$0
September	$1	$2
October	$11	$16
November	$39	$53
December	$87	$115
Annual	$408	$538

The estimated cost of using a water-to-water heat pump for the radiant floor heating with either the heat pump or furnace can be seen in Table 4.15.

Table 4.15: Monthly and annual cost for water-to-water heat pump using either a ground-source heat pump or liquid propane furnace for the forced air distribution system

Time Period	Total Water-to-Water Heat Pump Cost ($)	
	GSHP	Gas Fired Furnace
January	$32	$41
February	$20	$26
March	$11	$15
April	$4	$5
May	$0	$0
June	$0	$0
July	$0	$0
August	$0	$0
September	$0	$0
October	$3	$4
November	$10	$13
December	$22	$29
Annual	$102	$134

The estimated cost of using a liquid propane boiler for the radiant floor heating with either the heat pump or furnace can be seen in Table 4.16.

Table 4.16: Monthly and annual cost for liquid propane boiler using either a ground-source heat pump or liquid propane furnace for the forced air distribution system

Time Period	Total L.P. Boiler Cost ($)	
	GSHP	Gas Fired Furnace
January	$162	$207
February	$102	$133
March	$56	$74
April	$18	$25
May	$1	$2
June	$0	$0
July	$0	$0
August	$0	$0
September	$1	$2
October	$13	$20
November	$49	$66
December	$109	$144
Annual	$511	$673

Forced Air System Operating Performance and Costs in the Cooling Mode

Ground-Source Heat Pump

The method used in determining heat pump performance in the cooling mode was performed in a similar manner as with the heating load. First, the capacity of the heat pump in cooling mode was plotted and equations were obtained for the cooling capacity as a function of entering water temperatures.

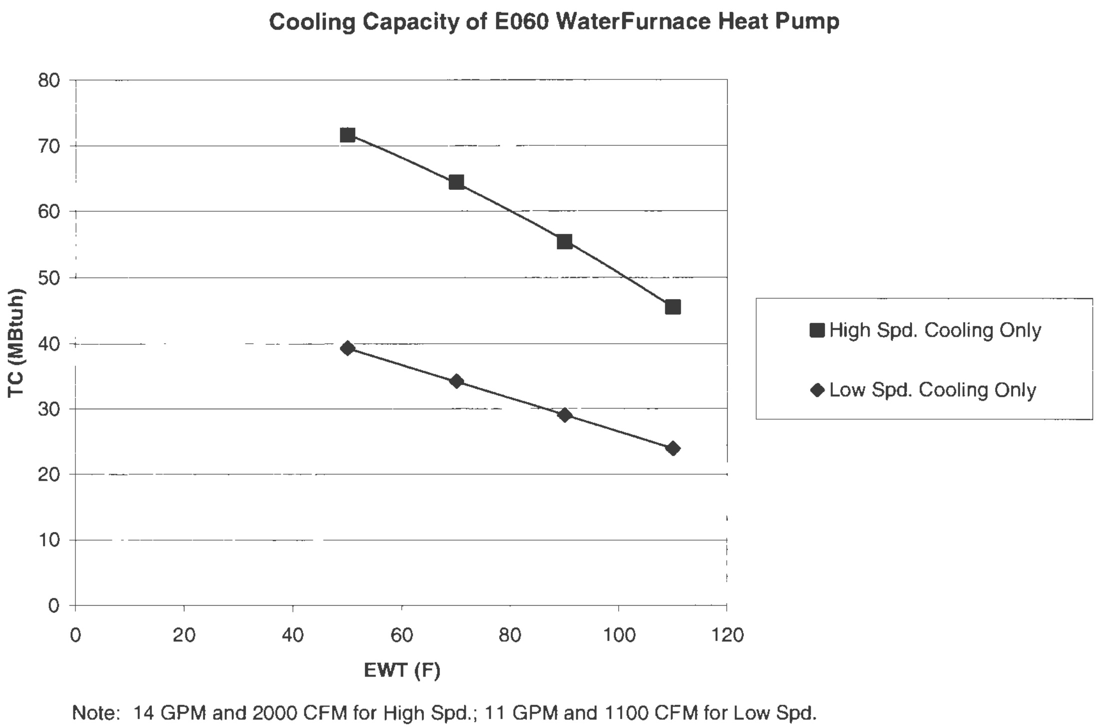

Figure 4.11: Cooling capacity of WaterFurnace EO60 heat pump

The cooling capacity of the heat pump in high speed was found to be

$$CC = -0.001625(EWT)^2 - 0.177(EWT) + 84.5725 \tag{4.48}$$

with an R^2 value of 0.9998.

The cooling capacity of the heat pump in low speed was found to be

$$CC = -0.257(EWT) + 52.16 \tag{4.49}$$

with an R^2 value of 1.000.

Using the estimated hourly entering water temperatures, the speed of the heat pump for each hour of the year was determined similarly as it was for heating. First, the high and low speed capacities were calculated for each hour using the fitted equations with respect to the corresponding entering water temperature. Next, the capacities were compared to the total hourly load. It was assumed that the lower level will be supplied the correct amount of cooling; as a result, the control for the heat pump was not determined by the upper level load alone. If the capacity of the heat pump in low speed was sufficient to meet the total load, then the unit was assumed to be operating in low speed for that hour; otherwise, the unit was assumed to be in high speed.

The hourly theoretical run time fraction of the heat pump is the ratio of the total home cooling load to the heat pump capacity at each respective hour.

$$Theoretical\ Run\ Time\ Fraction_i = \frac{Total\ Home\ Demand_i}{Heat\ Pump\ Capacity_i} \tag{4.50}$$

If the heat pump capacity was determined to be less than the load, then the theoretical run time fraction was assumed to be unity, i.e., the heat pump is assumed to run for the entire hour. The hourly part load factor that considers loss of efficiency due to cycling was determined by

$$PLF_i = 1 - C_D\left[1 - \frac{Total\ Home\ Demand_i}{Heat\ Pump\ Capacity_i}\right] \tag{4.51}$$

The actual run time fraction was then determined by equation 4.37.

The power draw of the heat pump was determined as a function of the entering water temperature as before from the manufacturer's specifications, seen in Figure 4.12.

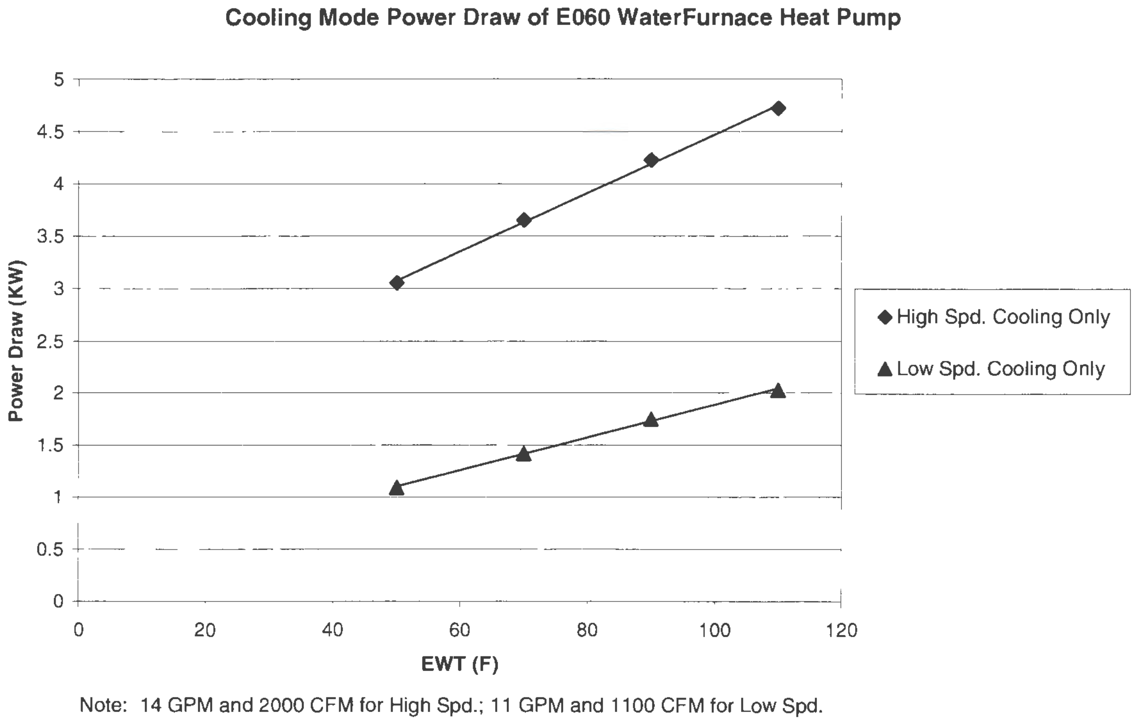

Figure 4.12: Power draw of heat pump in cooling mode

The power draw of the heat pump in the cooling mode on high speed was found to be

$$PD = 0.02795(EWT) + 1.6765 \tag{4.52}$$

with an R^2 value of 0.9979.

The power draw of the heat pump in the cooling mode on low speed was found to be

$$PD = 0.01555(EWT) + 0.3135 \tag{4.53}$$

with an R^2 value of 0.9986.

Finally, using an electric rate of \$0.091 dollars per kWh for cooling, the operating costs were estimated by

$$Cooling\ Cost_i = PD_i * (Actual\ Run\ Time\ Fraction)_i * Electric\ Rate \quad (4.54)$$

A summary of the monthly and annual cooling costs using the heat pump can be seen in Table 4.17.

Table 4.17: Monthly and annual cooling cost of ground-source heat pump

Period	Total Cooling Cost ($)
January	$0
February	$0
March	$0
April	$2
May	$4
June	$21
July	$56
August	$35
September	$8
October	$2
November	$0
December	$0
Annual	$128

Air Conditioner

Assuming that the efficiency of the typical air conditioner is constant, the cost per unit energy for cooling was determined by

$$\left(\frac{\$}{kWh}\right)\left(\frac{kW}{1000\,W}\right)\left(\frac{1{,}000{,}000\,Btu}{MMBtu_{Cooling}}\right)\left(\frac{W\,h}{Btu}\right) = \frac{\$}{MMBtu_{Cooling}} \quad (4.55)$$

The last product in units of watt-hours per Btu (Wh/Btu) is the reciprocal of the air conditioner's EER and will be assumed to be 12, which is typical. Using an electric cost in the summer of $0.091 dollars per kilowatt hour the cost per MMBtu's was found to be $7.58 dollars.

The estimation for the monthly and annual cooling costs for the home using each system were determined by

$$\left(\frac{\$}{MMBtu_{Cooling}}\right)\left(\frac{MMBtu_{Cooling}}{Period}\right) = \frac{\$_{Cooling}}{Period} \tag{4.56}$$

From Table 4.8 the annual cooling energy required was estimated to be 19.86 MMBtu's per year resulting in an annual estimated cost of $151 dollars. All monthly and annual estimated cooling costs can be seen in Table 4.18.

Table 4.18: Monthly and annual cooling cost of air conditioner

Period	Total Cooling Cost ($)
January	$0
February	$0
March	$0
April	$2
May	$6
June	$26
July	$61
August	$42
September	$10
October	$3
November	$0
December	$0
Annual	$151

CHAPTER 5 - ECONOMICS OF ALTERNATIVE APPROACHES

In this section, each of the three main alternative approaches will be compared economically using a life-cycle-cost analysis. First, all of the annual expenses for each approach were predicted over the life of the system. Then, the annual savings for each approach was determined in comparison to the conventional system. Finally, the present value for each approach was determined assuming an estimated interest rate on an alternative investment. Once this analysis was completed, the present values for each of the three alternative approaches were compared, which revealed the most economical system for the owners.

Expenses

The expenses for each heating and cooling approach will comprise the following: annual operation costs, equipment and installation costs, and interest payments on a loan. The maintenance costs for each approach were assumed to be the same for all systems and were not considered in the analysis. Each expense will initially be analyzed individually on an annual basis. Then, each annual expense will be summed together to obtain the total annual cost for implementing each approach.

The three main factors that most significantly contribute to the annual cost of operation for each approach are: the cost per unit of energy, volatility of future energy cost, and the overall system efficiency. Examining the alternative approaches, the residence will be dependant upon electricity for their heating and cooling needs in some way or another. The cost for electricity will vary depending on which utility is used, the benefits offered by that utility, the season, and the geographical location of the home. The utility that will be used at the residence is MidAmerican, which offers stable rates including an all-electric heating rate. The all-electric heating rate is a reduced electric rate that can be obtained in the heating season if the home is dominantly heated with an electrical driven device, such as a heat pump or electrical resistance heater.

The three alternative approaches will all be expected to utilize the all-electric heating rate, where approach 4 will not due to the dominant heating source being liquid propane gas. In addition, this reduced rate of electricity only applies to energy used in excess of 1,000 kilowatt hours (i.e. the first 1,000 kilowatt hours cost the regular rate and energy used in addition to 1,000 kilowatt hours costs the all-electric heating rate). The reasoning for offering this benefit after the first 1,000 kilowatt hours is based on the assumption that the home will use this amount of energy for everyday uses (lights, appliances, etc.) and the remainder is assumed to be used for heating the home. This is a benefit offered by the utility to promote the use of electricity for means of heating opposed to an alternative source of energy. The regular residence rate currently offered by the local utility is \$0.091 dollars per kilowatt hour. The all electric heating rate is currently \$0.035 dollars per kilowatt hour after all additional fees are applied.

To estimate the future increase in electricity rates, predictions formulated by the National Institute of Standards and Technology (NIST) were used (Schultz, 2004). The values in Table 5.1 represent the multiplier used to estimate the cost of electric for the residential market for 25 years in the future at various general price inflation rates. The utility that will be supplying electricity to the case study home has set a freeze on the cost of electricity until the year 2010. As a result, the price of electricity at the residence is guaranteed not to change for at least five years. Consequently, the price of electric until 2010 is known to be \$0.091 dollars per kilowatt hour at the regular rate and \$0.035 dollars per kilowatt hour at the all electric rate. The electric price indices were adjusted to account for the price freeze.

Table 5.1: Projected electric price indices

Year	Year	Electricity General Price Inflation Rate (%)			
		2%	3%	4%	5%
0	2004	1	1	1	1
1	2005	1	1	1	1
2	2006	1	1	1	1
3	2007	1	1	1	1
4	2008	1	1	1	1
5	2009	1	1	1	1
6	2010	1.07	1.14	1.21	1.28
7	2011	1.09	1.17	1.25	1.34
8	2012	1.12	1.21	1.31	1.42
9	2013	1.16	1.26	1.38	1.5
10	2014	1.19	1.31	1.44	1.59
11	2015	1.22	1.35	1.51	1.67
12	2016	1.25	1.4	1.57	1.76
13	2017	1.27	1.44	1.64	1.85
14	2018	1.3	1.49	1.7	1.95
15	2019	1.32	1.53	1.76	2.04
16	2020	1.35	1.57	1.84	2.14
17	2021	1.38	1.63	1.92	2.26
18	2022	1.41	1.68	2	2.37
19	2023	1.43	1.72	2.07	2.48
20	2024	1.47	1.78	2.17	2.62
21	2025	1.5	1.84	2.26	2.76
22	2026	1.53	1.9	2.35	2.9
23	2027	1.57	1.96	2.45	3.05
24	2028	1.6	2.02	2.55	3.2
25	2029	1.63	2.08	2.65	3.37

Source: NIST, 2004

The cost of liquid propane gas is fairly unregulated and is currently volatile. The current price of liquid propane gas for the heating season is approximately \$1.14 dollars per gallon. Again, to estimate the future cost of liquid propane gas, predictions from NIST were used.

Table 5.2: Projected liquid propane price indices

Year	Year	L.P. Gas General Price Inflation Rate (%) 2%	3%	4%	5%
0	2004	1	1	1	1
1	2005	0.89	0.89	0.9	0.91
2	2006	0.87	0.88	0.9	0.92
3	2007	0.89	0.92	0.94	0.97
4	2008	0.92	0.95	0.99	1.03
5	2009	0.94	0.99	1.04	1.09
6	2010	0.96	1.02	1.08	1.15
7	2011	1	1.07	1.14	1.22
8	2012	1.02	1.11	1.2	1.29
9	2013	1.05	1.15	1.25	1.37
10	2014	1.08	1.19	1.31	1.44
11	2015	1.11	1.24	1.38	1.53
12	2016	1.15	1.29	1.45	1.62
13	2017	1.18	1.34	1.52	1.72
14	2018	1.21	1.38	1.58	1.81
15	2019	1.24	1.43	1.66	1.91
16	2020	1.27	1.48	1.73	2.02
17	2021	1.3	1.54	1.81	2.13
18	2022	1.33	1.59	1.89	2.25
19	2023	1.37	1.64	1.97	2.37
20	2024	1.4	1.7	2.06	2.5
21	2025	1.44	1.76	2.16	2.64
22	2026	1.47	1.82	2.25	2.78
23	2027	1.5	1.88	2.35	2.93
24	2028	1.54	1.94	2.45	3.08
25	2029	1.57	2.01	2.55	3.24

Source: NIST, 2004

The use of liquid propane gas at a residence has several other disadvantages in addition to volatile rates. This fuel is not available for remote supply via gas lines to the home from a utility. Therefore, the consumer will be required to maintain sufficient levels of this fuel on-site year-round in a rented or purchased tank. Further, the consumer will benefit if they commit to the gas price in the summer season by accepting to enter into a contract with the supplier; thus, the consumer must monitor the current rate of the fuel in order to obtain the best price throughout the year.

For this study, a long-term average inflation rate of three percent was assumed. Using the corresponding price indices and annual operating cost estimated at current energy costs, the future annual operation costs were found.

Table 5.3: Future annual operating costs using heat pump for forced air distribution system

Year	Forced Air		Radiant Floor		
	E060 Heat Pump Heating	E060 Heat Pump Cooling	EW040 Heat Pump	Electric Boiler	Liquid Propane Boiler
2004	$208	$128	$102	$408	$511
2005	$208	$128	$102	$408	$455
2006	$208	$128	$102	$408	$450
2007	$208	$128	$102	$408	$470
2008	$208	$128	$102	$408	$486
2009	$208	$128	$102	$408	$506
2010	$237	$145	$116	$465	$521
2011	$244	$149	$119	$477	$547
2012	$252	$154	$123	$494	$567
2013	$262	$161	$129	$514	$588
2014	$273	$167	$134	$534	$608
2015	$281	$172	$138	$551	$634
2016	$291	$179	$143	$571	$659
2017	$300	$184	$147	$588	$685
2018	$310	$190	$152	$608	$705
2019	$319	$195	$156	$624	$731
2020	$327	$200	$160	$641	$756
2021	$339	$208	$166	$665	$787
2022	$350	$214	$171	$685	$813
2023	$358	$219	$175	$702	$838
2024	$371	$227	$182	$726	$869
2025	$383	$235	$188	$751	$900
2026	$396	$242	$194	$775	$930
2027	$408	$250	$200	$800	$961
2028	$421	$258	$206	$824	$992
2029	$433	$265	$212	$849	$1,027

The annual operating costs were also estimated for the conventional system and can be seen in Table 5.4.

Table 5.4: Future annual operating costs for conventional system

Year	Forced Air Heating.	Forced Air Cooling	Radiant Floor Heating	Total Annual Heating and Cooling Costs
	Gas Fired Furnace	Air Conditioner	Liquid Propane Boiler	
2004	$912	$151	$673	$1,736
2005	$812	$151	$599	$1,562
2006	$803	$151	$593	$1,546
2007	$839	$151	$619	$1,610
2008	$867	$151	$640	$1,657
2009	$903	$151	$667	$1,721
2010	$931	$172	$687	$1,789
2011	$976	$176	$720	$1,873
2012	$1,013	$182	$747	$1,943
2013	$1,049	$190	$774	$2,013
2014	$1,086	$197	$801	$2,084
2015	$1,131	$203	$835	$2,170
2016	$1,177	$211	$869	$2,257
2017	$1,223	$217	$902	$2,342
2018	$1,259	$224	$929	$2,413
2019	$1,305	$230	$963	$2,498
2020	$1,350	$236	$997	$2,583
2021	$1,405	$245	$1,037	$2,688
2022	$1,451	$253	$1,071	$2,775
2023	$1,496	$259	$1,104	$2,860
2024	$1,551	$268	$1,145	$2,964
2025	$1,606	$277	$1,185	$3,068
2026	$1,661	$286	$1,226	$3,172
2027	$1,715	$295	$1,266	$3,277
2028	$1,770	$304	$1,306	$3,381
2029	$1,834	$313	$1,353	$3,501

Finally, the total cost for heating and cooling the home annually with each individual approach was determined. The resulting annual operating costs for each approach can be seen in Appendix F, Tables F1 through F7. A summary of these results can be seen in Table 5.5.

Table 5.5: Heating and cooling lifetime costs and average annual costs for each approach

Approach	Lifetime Heating Costs	Average Annual Heating Costs	Lifetime Cooling Costs	Average Annual Cooling Costs
1) E060 and EW040	$11,625	$465	$4,783	$191
2) E060 and Electric Boiler	$23,094	$924	$4,783	$191
3) E060 and L.P. Boiler	$25,799	$1,032	$4,783	$191
4) L.P. Boiler & Furnace and CAC	$55,838	$2,234	$5,645	$226

The initial equipment and installation costs for the same project can vary depending on the contractor. Therefore, obtaining competitive bids for the project may be beneficial to the consumer. Residentially, most contractors submit a "lump-sum" quote to the consumer that does not break down costs for each individual item. As a result, determining cost differences between alternative systems can become difficult. For the study, bids were obtained for each alternative approach. It should be noted that approach 1 (i.e., all geothermal) was initially bid using a nominal 6 ton water-to-water heat pump instead of a nominal 2 ton heat pump. The bid price seen in Table 5.6 is a different estimated bid price for replacing this unit.

Table 5.6: Initial cost for the equipment and installation of each alternative

Approach	System	Bid Price
1	E060 and EW040	$50,000
2	E060 and Electric Boiler	$35,900
3	E060 and L.P. Gas Boiler	$37,000
4	L.P. Gas Furnace and Boiler and CAC	$23,700

The annual interest costs on the loan can make a significant impact on the total cost of the project. The residence under consideration will be purchased with an amortized loan from a bank or other financial source for a term of 10 years. An amortized loan is a loan in which the amount of interest owed for a specified period is calculated based on the remaining balance of the loan at the beginning of the period (Park, 2002). The amount paid per period on the loan is a constant amount, thus, the amount of interest paid and amount on the principal paid will vary with each payment.

Initially, the amount of money to be paid per year on the loan was determined. The size of payment is equal to the net present value (cost) of the system separated into annual installments at the particular interest rate over the specified period of time. The average percentage rate (APR) of the loan will be assumed to be 5.6 percent. The amount of each annual installment can be calculated by

$$a = K\left[\frac{i_T(1+i_T)^N}{(1+i_T)^N - 1}\right] \tag{5.1}$$

The amount of interest paid on the first payment can be found using

$$I_{P.1} = K\, i_T \tag{5.2}$$

Knowing that the annual payment is equal to the interest amount plus the principal amount of the payment, the amount of money paid on the principal for the first payment was found by

$$\mathrm{Pr}_1 = \mathrm{a} - \mathrm{I}_{\mathrm{P.1}} \tag{5.3}$$

The remainder of the balance can be determined at the end of the first year after the first payment using Equation 5.4.

$$\mathrm{B}_1 = \mathrm{K} - \mathrm{Pr}_1 \tag{5.4}$$

Once the first values have been found, the remaining values for the term of the loan can be found using Equations 5.5, 5.6, and 5.7.

$$B_n = K - (\mathrm{Pr}_1 + \mathrm{Pr}_2 + ... + \mathrm{Pr}_n) \tag{5.5}$$

$$I_{P.n} = (B_{n-1})i_T \tag{5.6}$$

$$\mathrm{Pr}_n = a - I_{P.n} \tag{5.7}$$

The annual interest paid, annual principal paid, and loan balance for each year for each approach can be seen in Appendix E in Tables E1 through E4. A summary of the total loan payments for the term of the loan can be seen in Table 5.7.

Table 5.7: Summary of loan payments and project costs of each approach

Approach	Total Payments to Bank	Total Interest Paid	Total Cost Difference from Conventional System
1	$66,652	$16,652	$35,059
2	$47,856	$11,956	$16,263
3	$49,323	$12,323	$17,730
4	$31,593	$7,893	$0

Savings

The estimated savings for the three main approaches were considered by separating the savings compared to the conventional system into several different categories, namely, available rebates for installing a geothermal system, tax savings or "tax write-offs," and operation savings.

The rebates that are available for installing a geothermal system are based on the installed heating capacity of the system at the rate of $300.00 dollars per ton. This rebate is given to the owner upon initial installation.

Table 5.8: Available rebate for each approach

Approach	Installed Heating Capacity (Tons)	Available Rebate ($)
E060 and EW040	8	$2,400
E060 and Electric Boiler	5	$1,500
E060 and L.P. Boiler	5	$1,500
L.P. Boiler & Furnace and CAC	0	$0

The tax savings or "tax write-offs" will also contribute to the overall savings of the project. The amount of tax savings to the consumer will be dependant on the tax bracket of the consumer and the APR of the loan. The amount of interest paid on the loan will be tax deductible. The consumer is in the 45 percent tax bracket; as a result, 45 percent of the interest paid on the loan at a particular year will be saved in taxes. The amount of annual tax savings for each approach can be determined by

$$Tax\ Savings_n = Tax\ Rate * \left(I_{P,n}\right) \tag{5.8}$$

The total tax savings for the project is the summation of the annual tax savings over the life of the loan.

$$Total\,Tax\;Savings = \sum_{n=1}^{10} Tax\;Savings_n \qquad (5.9)$$

The annual tax savings for each approach was determined and compared to the conventional system. The annual tax savings for each approach can be seen in Appendix G Tables G1 through G4. Seen in Table 5.9 is a summary of the results.

Table 5.9: Summary of tax savings for each approach

Approach	Total Tax Savings ($)	Total Tax Savings Compared to Conventional ($)
1	$7,494	$3,942
2	$5,380	$1,828
3	$5,545	$1,993
4	$3,552	$0

Finally, the annual and total operation savings was determined. The operating costs for each approach are seen in Appendix F and in Table F.1. The savings for each of the three approaches is the difference in operating costs between each approach and the conventional approach. A summary of the total operation savings for each approach over the assumed lifetime of the systems is shown in Table 5.10.

Table 5.10: Summary of total operation savings for each approach

Approach	Lifetime Heating Savings to Approach 4	Lifetime Cooling Savings to Approach 4	Lifetime Total Savings to Approach 4
1	$44,212	$862	$45,074
2	$32,744	$862	$33,606
3	$30,038	$862	$30,900
4	$0	$0	$0

The annual operation savings for each approach can be seen in Appendix H.

Life-Cycle-Cost Analysis

A life-cycle-cost analysis is a method of analyzing the annual expenses and savings for a system over the life of the equipment; this method normalizes the annual cash flow to an overall present value when given an assumed discount rate. The present value or worth of an alternative represents a measure of future cash flow for an alternative relative to the time point "now" with provisions that account for earning opportunities (Park, 2002). In other words, the present worth of an approach would be the amount of money needed today in a bank account earning an assumed interest rate to purchase and operate the system over the life of the project. Thus, the approach showing the smallest cost in today's dollars or the largest present value will be the most economical option for the consumer over the lifetime of the system.

The total annual expenses were determined by

$$(Loan\ Payment)_n + (Operation\ Cost)_n = (Total\ Expense)_n \tag{5.10}$$

While, the total annual savings were determined by

$$(\mathrm{Re}bate\ Savings)_n + (Tax\ Savings)_n = (Total\ Savings)_n \tag{5.11}$$

Next, the net cash flow for each year was determined. The annual net cash flow is the total flow of money occurring at each year (i.e., dollars saved for the year minus the dollars spent for the year).

$$(Total\ Savings)_n - (Total\ Expenses)_n = (Net\ Cash\ Flow)_n \tag{5.12}$$

The present value of the net cash flow for each year was determined by

$$PW_n = (Net\ Cash\ Flow)_n (1 + i_T)^{-n} \tag{5.13}$$

An interest rate of 4.5 percent was assumed for an alternative investment.

Once the present worth of the net cash flow for each year was found, they were summed together to obtain the net present worth as follows

$$Net\,PW = \sum_{n=1}^{25} (PW)_n \tag{5.14}$$

The annual present values for the cash flows for each approach were determined and are tabulated in Tables I.1 through I.4 in Appendix I. A summary of the overall net present values for each approach can be seen in Table 5.11.

Table 5.11: Overall net present values for each approach over the life of the systems

Approach	Overall Net Present Value ($)
1) E060 and EW040	-$52,854
2) E060 and Electric Boiler	-$46,373
3) E060 and L.P. Boiler	-$48,753
4) L.P. Boiler & Furnace and CAC	-$53,876

It can be seen that approach 2 which uses a nominal 5 ton water-to-air ground-coupled geothermal heat pump and an electric boiler has the least negative net present value of -$46,373 dollars and is thus, the most economical approach based on the life-cycle-cost analysis.

Payback Period

Often the consumer is interested in knowing how long the payback period will be for their investment. The payback period is the amount of time it takes to recover the cost of the project. Consequently, the payback period will be the amount of time it takes to recover the additional amount of money that would be spent to install the custom system, namely approaches 1, 2, or 3. For example, if approach 1 costs $1,500.00 dollars after everything was considered and approach 4 costs $1,000.00 after everything was considered, the additional amount of money for approach 1 would be $500.00 dollars. And further, if approach 1 would save $100.00 dollars per year, the payback period would be 5 years.

For this analysis, the cost of each approach was determined considering the additional cost of borrowing money from the bank. Thus, the cost of each approach was determined by

$$\textit{Total Cost of Approach} = \textit{Initial Equip. and Install Costs} + \textit{Total Interest Costs} \quad (5.15)$$

Seen in Table 5.12 is a summary of the total cost for each approach.

Table 5.12: Total cost for each approach

Approach	Total Cost ($)
E060 and EW040	$66,652
E060 and Electric Boiler	$47,856
E060 and L.P. Boiler	$49,323
L.P. Boiler & Furnace and CAC	$31,593

The additional cost for approaches 1, 2, and 3 can be determined by taking the difference from the conventional system. Seen in Table 5.13 is the additional amount of money required to purchase the custom systems.

Table 5.13: Additional cost required for each approach

Approach	Additional Cost for Each Approach ($)
E060 and EW040	$32,659
E060 and Electric Boiler	$14,763
E060 and L.P. Boiler	$16,230
L.P. Boiler & Furnace and CAC	$0

The payback periods for the custom systems will be the amount of time until the additional costs are recovered by the savings. The savings for each approach will be the summation of the annual rebate savings, operating savings, and tax savings in comparison to the conventional system.

Table 5.14: Payback period for each custom approach

Approach	Payback Period (years)
E060 and EW040	18
E060 and Electric Boiler	11.8
E060 and L.P. Boiler	13.9

Another way to look at the savings for each approach is to determine the total savings that each approach will accrue after the system has paid itself back. These values were determined and can be seen in Table 5.15.

Table 5.15: Benefit dollars after payback

Approach	Benefit $'s After Payback ($)
E060 and EW040	$15,694
E060 and Electric Boiler	$20,217
E060 and L.P. Boiler	$16,389

Seen in Table 5.15, approach 2 will accrue the most money after the system has paid itself back.

Final Selection of the Ideal Heating and Cooling Approach

The final selection of the ideal heating and cooling approach will encompass several considerations. These considerations will include the economics of each approach, the requirements and vulnerabilities for operating each approach, and the owners overall comfort level in investing in each approach.

First, the economics of each approach will be evaluated. After all of the analysis was completed, the owner was presented with the following table that summarizes all of the economic evaluations for each approach.

Table 5.16: Summary of all economic results for each approach

System Alternatives	E060 & EW040	E060 & Electric Boiler	E060 & Weil McLain Boiler (L.P. fired)	Bryant Furnace & Weil McLain Ultra L.P. Boiler & CAC
Utilization Method	All Geothermal	Geo. & Elect. Resist.	Geo. & L.P.	All Conventional
Equipment and Installation Costs				
Bid Price ($)	$50,000	$35,900	$37,000	$23,700
Total Loan Cost ($)	$66,652	$47,856	$49,323	$31,593
Cost Diff. From Conventional ($)	$35,059	$16,263	$17,730	$0
Estimated System Operation Costs Over the Life Time of the Equipment				
Lifetime Heating Cost ($)	$11,315	$21,812	$24,339	$52,650
Ave. Annual Heating Cost ($)	$453	$872	$974	$2,106
Lifetime Cooling Cost ($)	$4,655	$4,655	$4,655	$5,331
Ave. Annual Cool. Cost ($)	$186	$186	$186	$213
Estimated Savings In Heating and Cooling Costs (Utility Bill) Compared to the Conventional System				
Lifetime Heating Savings ($)	$41,335	$30,838	$28,311	$0
Ave. Ann. Heating Savings ($)	$1,653	$1,234	$1,132	$0
Lifetime Cooling Savings ($)	$676	$676	$676	$0
Ave. Ann. Cooling Savings ($)	$27	$27	$27	$0
Estimated Payback Period and Benefit Dollars After Payback				
Payback Period (Yrs.)	>25	17.8	21.5	N/A
Benefit $'s After Payback ($)	$15,694	$20,079	$16,251	N/A
Estimated Present Worth of Each System Assuming Interest Rate of Alternative Investment				
PW Heating & Cooling Operation ($)	-$8,792	-$14,585	-$15,944	-$31,852
PW Cost of System ($)	-$44,063	-$31,860	-$32,882	-$22,023
PW Entire System ($)	-$52,854	-$46,445	-$48,826	-$53,876
Estimated Expenses and Savings				
Total Lifetime Expenses ($)	$82,623	$74,324	$78,317	$89,575
Total Lifetime Savings ($)	$48,353	$34,843	$32,481	$0
Difference ($)	$34,270	$39,481	$45,836	$89,575

As seen in Table 5.16, there are many ways that each system can be compared economically. However, the one economic factor that considers all costs and savings while adjusting these numbers to a present worth, is the overall present value of the system. It can be seen that approach 2, has the greatest overall present value and is thus, the most economical system. Therefore, approach 2 will require the least amount of money in "present dollars" to purchase, install, and operate over the assumed life of the system.

The operation requirements and vulnerabilities for each approach is an additional consideration. All of the approaches are susceptible to a power outage. However, in this day in age, almost all means of heating and cooling are dependent on electricity; this vulnerability will be assumed to be inevitable. Approach 1, being all geothermal could present some vulnerability if something were to happen to the ground heat exchanger. In this circumstance, the system would be solely dependent on the 20 kW electric resistance auxiliary heater installed with the heat pump, and further liquid propane, the home would be without any radiant floor heating. Approach 3, which uses a liquid propane boiler, would be able to maintain the radiant floor heating if the loop were to fail. However, the owner would be required to maintain sufficient levels of fuel on-site in a rented tank and thus, would be subjected to the volatile rates of this fuel. Approach 2 offers versatility from a loop failure and avoids dealing with liquid propane fuel; this approach was deemed to be the least bothersome and most flexible.

The owners' feelings towards each approach are also a concern in that it is important that the owner feels comfortable with the system that will be installed and that they understand the specifics of the system. Consumers tend to be hesitant when dealing with new or non-typical systems and technology. Thus, the basics of each approach were discussed with the owners until they felt satisfies with their understanding of the systems. Further, it was noted that the owner liked the idea of a back-up means for heating the home if something were to happen to the ground loop, which discouraged approach 1. In conclusion, due to the economics, requirements and vulnerabilities, and the owners feeling for each alternative, approach 2 was deemed to be the ideal system.

CHAPTER 6 - DISCUSSION AND RESULTS

This research showed that accurate load calculations are critical for correctly sizing the heating and cooling equipment and thus, achieving comfortable conditions in the home, avoiding excessive initial equipment costs, and obtaining optimal performance from the system. The resulting heating and cooling loads that were calculated using the ASHRAE methods outlined in the analysis yielded reasonable values in comparison to the estimates of the contractor; specifically, the estimated heating and cooling loads were found to be 6.2 (7.7 tons with garage) and 4.5 tons respectively under the assumed design conditions. The resulting heating and cooling loads were within 10 percent of the contractors estimates. It was seen that the greatest effect on the design cooling load was the large amount of glass in the home which contributed approximately 54 percent of the total design load.

Using the owner's request of implementing geothermal and radiant floor systems, three alternative approaches for conditioning the home were devised. Also, conventional heating and cooling systems were considered for comparison purposes. Each of the first three approaches included a water-to-air ground-source geothermal heat pump for the forced air distribution system. The alternatives considered for the hydronic radiant floor system included either a water-to-water ground-source geothermal heat pump, electric boiler, or a liquid propane boiler. Furthermore, tentative equipment selections were made for each approach based on the design heating load. These approaches were ultimately compared on an economic basis.

For calculating monthly and annual energy use for residential homes, there are simplified procedures such as the degree-day and bin methods available. However, these methods make the assumption that the building load is a linear function of outdoor air temperature. In reality, infiltration, solar effects, and internal gain have little or no relation to outdoor air temperature. In addition, these methods utilize a balance point or reference temperature, which indirectly assumes a constant rate of internal gain. And lastly, these methods do not allow the user to represent the homeowners' particular lifestyle and preferences; for example, the indoor air temperature that is kept in the home during the winter, or when the homeowner may turn on the air conditioner in the cooling season. When considering all the discrepancies associated with using these simplified methods, the actual building load and annual energy use estimations can be significantly compromised.

The transmission model assumes one-dimensional, steady-state conduction heat transfer through the walls, doors, floors, etc. The transmission heating and cooling loads are specific to the geographical location of the home since it is a function of the hourly outdoor air temperature. In the heating and cooling modes the indoor air temperatures were assumed to be constant at 68 ^{o}F (50 ^{o}F for garage) and 74^{o}F respectively. For the $UA_{conduction}$ values, the film coefficients were not adjusted according to the hourly wind speed values and were assumed to be constant for the corresponding season. The film coefficient used in the heating season was found for a wind speed of 15 mph, and the cooling season coefficient was found for a wind speed of 7.5 mph.

The hourly solar gain model considers geographical location, orientation of the glazing, ground cover, amount of glazing in the home, type of windows, and type of interior shading. The TMY2 data is tailored towards solar energy systems and represents the typical solar conditions that can be expected at the location of the home. However, some estimations were made in regarding how much of the incident solar energy penetrates the structure and contributes to the heating and cooling loads. For interior shading, this model assumes that the shades in the home will always be drawn. Further, the solar model does not consider exterior shading of any kind. The case study home does have overhangs all around the house and a relatively close bank of trees on the northeast side. Overall, the intent of assuming the interior shading devices will always be drawn was to offset some of the long-term error associated with neglecting the exterior shading (i.e., the overhangs and trees). A future improvement of the model could include estimating the effects of the exterior shading and allow for the variation of interior shading use of the homeowner.

It was assumed that for the hours the TMY2 data had nonzero values for snow depth the ground reflectance was 0.7, which is typical for white snow; otherwise, green grass was assumed to be the ground cover and a ground reflectance value of 0.25 was used. The amount of snowfall in Des Moines, Iowa can significantly vary from year to year which could notably change the amount of diffuse radiation striking the home. To better illustrate this point, Figure 6.1 was created, which shows the diffuse radiation incident on the home in January and February and the assumed ground reflectance.

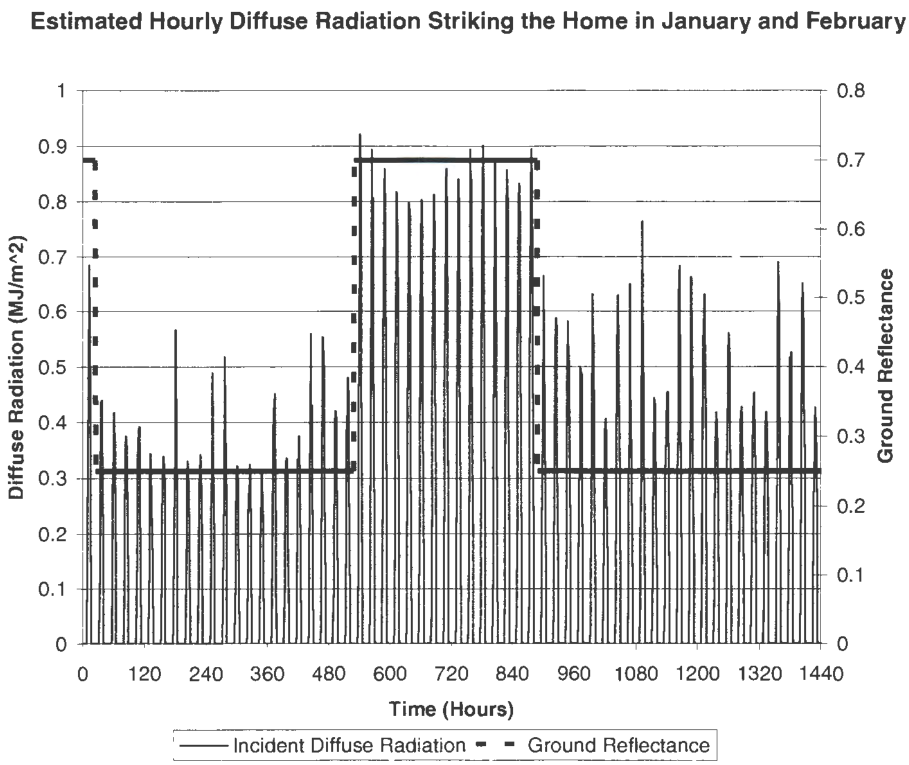

Figure 6.1: Incident diffuse radiation to ground reflectance in January and February

Figure 6.1 shows that the estimated ground reflectance has a significant effect on the incident diffuse radiation on the home. The average nonzero values of diffuse radiation during times when the ground reflectance is 0.25 and 0.7 are approximately 0.3 MJ/m^2 and 0.5 MJ/m^2 respectively. Thus, on average, the magnitude of the diffuse radiation increases by approximately 66 percent from a ground cover of grass to snow. For the southeast face of the home, the diffuse radiation contributes an estimated 41 percent of the total incident radiation annually. Therefore, the amount of time that snow covers the ground surrounding the home annually could make a significant impact on the solar load to the home and it is important that this parameter is represented correctly in the model.

The infiltration model considers the volume of the home, outdoor air temperature, wind speed, wind direction, and the exposure (above or below grade) of each wall of the home. The ACH values were assumed to be directly proportional to wind speed (e.g., if wind speed doubles, then the ACH value doubles). This assumption was made assuming that the cracks in the structure can be modeled as an orifice and that the pressure coefficient associated with the air flow is constant at all air velocities occurring through the cracks. Also, the assumption was made that the wind always strikes the home perpendicularly to the exterior walls of the home. Future improvements of this model could include adjusting infiltration rates for the wind direction variance from perpendicular.

The internal gain portion of the model was particularly tailored to the lifestyle of the individual homeowners. This model allows inputs for times of occupancy, times when the occupants are asleep, number of occupants, appliances, and lighting demands. This approach allows for an estimation of variable internal gain opposed to assuming a balance point temperature or constant internal load. The internal load to the home was difficult to estimate given that the occupants in the home have a fairly active lifestyle. The types of activities that will occur in the home such as cooking, laundering and entertaining guests, etc., can significantly vary from day to day. However, since energy estimations are made on a monthly and annual basis, some of these discrepancies should become less of an effect over the long term.

The results of the model include hourly energy use for predicting the HVAC performance and monthly and annual heating and cooling energy demand as a whole. All monthly and annual energy predictions were made considering the outdoor air temperature that the particular homeowner would activate the HVAC systems, which further tailors the results making them closer to what will actually occur. The total hourly heating and cooling loads of the home including the garage were plotted and can be seen in Figure 6.2.

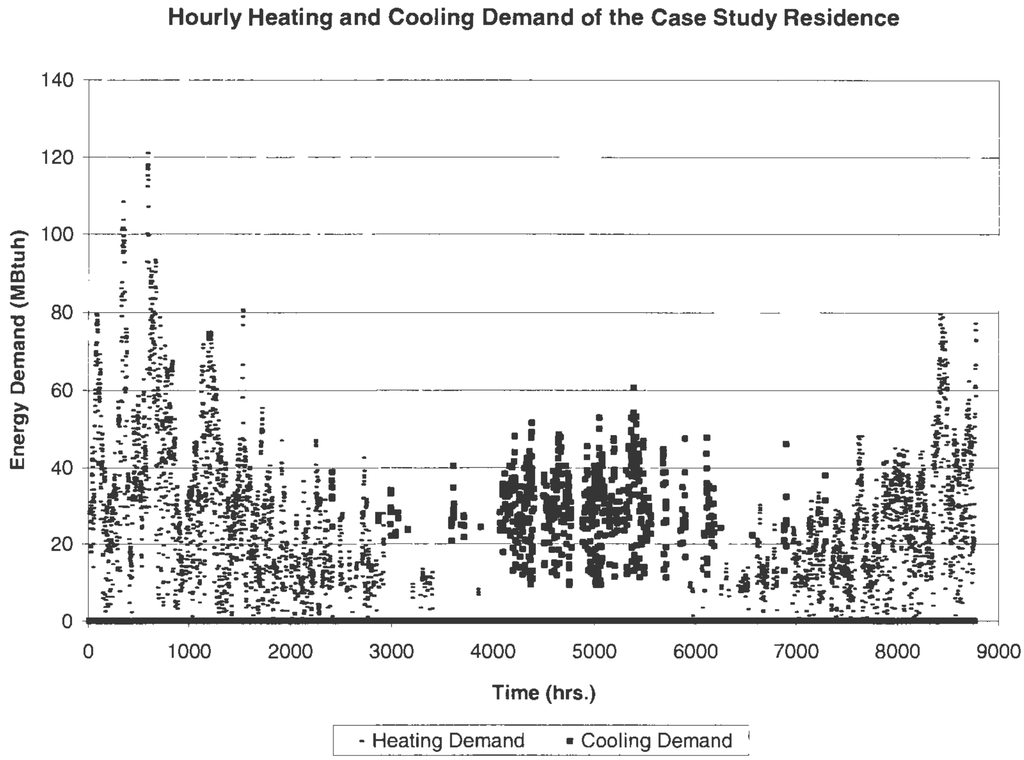

Figure 6.2: Hourly heating and cooling demand of case study residence

It was concluded that the forced air distribution system and the radiant floor heating system in the heating mode will be required to supply approximately 117.1 MMBtu's of energy annually. Also, the forced air distribution system will be required to supply about 19.9 MMBtu's of energy to cool the home annually.

The degree-day analysis was performed to estimate the difference in results for annual energy use in comparison to the hourly simulation model. It was estimated using the degree-day analysis that the home would use approximately 153.27 MMBtu's per year in heating yielding a percent difference of approximately 31 percent from the result of the hourly simulation model. Further, the annual cooling energy estimated using the degree-day analysis was 68.95 MMBtu's per year which is approximately 3.5 times greater than from the hourly simulation model.

Also, the bin method was performed to estimate the difference in results for annual energy use in comparison to the hourly simulation model. In the hourly simulation model no heating energy was estimated when the outdoor air temperature was greater than 50 ^{o}F. As a result, for the bin analysis a balance point temperature of 50 ^{o}F was used in effort to compare the results with similar inputs. The resulting annual heating energy use of the home using the bin method was 80.4 MMBtu's resulting in a percent difference of approximately 31 percent from the hourly simulation model. Further, the annual cooling energy estimated using the bin method was 41.2 MMBtu's per year which is about twice as much than what was estimated from the hourly simulation model.

The HVAC performance model ultimately estimates the monthly and annual cost for each of the considered systems. The operating characteristics of the ground-source heat pump and liquid propane furnace were evaluated on an hourly basis. This was done because the heat pump's efficiency and capacity varies with entering water temperature and the operating characteristics of both systems dictate how much demand will be placed on the radiant floor heating system in the lower level in the heating condition. The costs of operating the radiant floor system were projected assuming each piece of equipment operates at a constant efficiency. Also, the cost of operating a blower for the gas furnace was neglected for the analysis.

The estimation of the hourly entering water temperature was required for estimating the hourly performance of the ground-source heat pump. However, the hourly entering water temperature to the heat pump is only a rough estimate of what may actually happen and that generalizing this estimation for all sites with extreme accuracy would be nearly impossible. Obtaining the most accurate prediction of entering water temperatures for hourly simulation would require knowledge of the ground water movement, rainfall, in-ground evaporation rates, ground hydraulic conductivity, in addition to the thermal conductivity and diffusivity of the soil. Obtaining all of this information would require extensive testing and can vary significantly from site to site.

The amount of heating supplied to the lower level by the ground-source heat pump was determined to be approximately 12.5 MBtuh when the heat pump is operating in high speed. This was done so that the owners of the home could efficiently heat the lower level most of the time with the heat pump and enjoy the comfortable heating of the radiant floor when the heating load to the home is high. To achieve this amount of heating to the lower level the volumetric flow rate of air must be set to the calculated value of approximately 470 CFM when the heat pump is on high speed with a flow hood. It is possible that the diffuser positions may need to be reset in the cooling season to supply the lower level the correct amount of cooling.

Under the assumed configuration and estimated hourly loads to the home some performance characteristics of the HVAC systems were estimated. It was found that the heat pump will operate in high and low speed approximately 2,975 and 972 hours per year respectively. It was also estimated that the electric resistance auxiliary heater on the heat pump will supply about 0.834 MMBtu's per year. In addition, it was found that the auxiliary heat will engage on average at approximately 0 ^{o}F and will contribute about $10 dollars per year to the annual heating costs.

Since there was a substantial difference between annual operating costs and initial installation costs of each alternative it was determined that a life-cycle-cost analysis was necessary to determine which alternative would be most economical over the entire life of the system. It was assumed that the life of each alternative would be twenty-five years. To perform the life-cycle-cost analysis the future fuel costs had to be estimated. Currently, electric and liquid propane rates are fairly volatile making the confidence of these estimated fuel costs for twenty-five years in the future uncertain. Estimations presented by NIST were used for both fuels.

In performing the life-cycle-cost analysis all expenses (excluding maintenance costs) and savings on an annual basis were predicted for each approach. Then, all cash flows were discounted to their respective present values to obtain a net present worth for each approach.

The analysis concluded that approach 2 using the water-to-air ground-coupled geothermal heat pump and electric boiler had the least negative present value of approximately -$46,400 dollars, and was deemed the most economic.

Due to the residence having a substantially greater amount of living area and amount of glass than would be expected in a typical home, the energy use for heating and cooling will be exacerbated. Thus, selecting a highly efficient means for heating and cooling this home was critical for minimizing the amount of money spent to condition the home. Therefore, the decision to implement a geothermal system for means of heating and cooling the home was deemed wise. It was estimated that on average the owners will save roughly $1,250 dollars per year in heating and cooling costs using approach 2 opposed to the conventional approach.

The result showing approach 2 being the most economical was not anticipated. It was initially thought that the water-to-water heat pump would be the most economical for the hydronic heating. This initial hypothesis was based on the fact that the operating costs are much less with the water-to-water heat pump than that of the electric or liquid propane boiler. However, it was found that the initial equipment and installation costs quoted by the subcontractors were high enough that it offset the present value of the system showing it not to be as economical in the long run as expected.

An alternative source had estimated that a reasonable cost for approach 1 would be approximately $40,000 dollars. Using the spreadsheets created for the analysis it was determined that approach 1 would have been the most economical system if had it been quoted at approximately $43,000 dollars or less. Thus, the initial hypothesis would have been correct if the system would have been quoted at a more reasonable price.

Several meetings were scheduled with the subcontractor in regards to the preliminary design of the system. Within these meetings, the contractor expressed multiple concerns for implementing approach 1 which mainly consisted of the requirement of additional controls, installation, and initial cost. More specifically, they mainly argued that installing two heat pumps on a single loop would be very expensive and require additional design and controls and further, the cost of a water-to-water heat pump would be much greater than that of a boiler. However, it was found that this is not necessarily true and in fact, installing multiple heat pumps on a single loop is very common in the commercial arena.

It was concluded that the contractor may have purposely bid approach 1 excessively high due to inadequate experience with this system and/or their unwillingness to install this system in order to discourage the owner from this option. Often, it appears that some contractors prefer to install only certain types of systems, the types of systems that they install everyday. In this situation, the contractor is very familiar with the installation requirements and is confident in its operation. When a contractor is requested to install a system that they are not as familiar with they hesitate due to their liability exposure if the system does not work. It is believed that this is the particular problem that was encountered with requesting the water-to-water heat pump.

CHAPTER 7 - CONCLUSIONS

Predictions of residential building energy use and estimations of costs for alternative HVAC systems are seldom performed for the homeowner. The methods presented within this study can be used to show the homeowner or prospective consumer that energy efficient systems can significantly reduce the costs associated with heating and cooling. As a result, the use of energy efficient technology, such as geothermal HVAC systems, may become more common. A more widespread use of energy efficient technology for heating and cooling residential homes could lead to a net reduction of harmful emissions to the environment associated with conventional methods.

The models developed for this study have the ability to evaluate the energy performance of a non-typical home due to the particular inputs specific to the programs. In addition, these models are able to estimate the performance and operating costs of ground-source geothermal heat pumps, which is uncommon with current energy simulation software packages. The efficiency of a ground-source geothermal heat pump varies throughout the year depending on the load and entering water temperatures and was considered in this study. Moreover, an effort was made to tailor the results of the models to the lifestyle of the particular homeowner.

The use of these models does require some knowledge of topics specific to building energy and may not be known to everyone. Future improvements to the models developed for this study could include easier methods for estimating parameters that are not trivial to someone with a limited background in building energy. In addition, methods to more accurately predict the entering water temperatures to a heat pump on an hourly basis for simulation purposes are needed. There are currently techniques for estimating entering water temperatures; however, they utilize broad assumptions that do not consider many site specific issues.

To monitor the future performance of the HVAC system installed in the case study home, the heat pump and electric boiler will be instrumented and data will be collected. It has been proposed to gather real-time data on a five minute basis. This data will allow actual energy use and HVAC performance characteristics of the case study home to be attained. The actual data will be compared to the predictions of the simulation models to establish the validity of the model.

A specific rig that will contain the instrumentation on the heat pump has been proposed to the heating and cooling contractor and will be installed within the first few months of operation. The instrumentation will include a differential pressure transducer for obtaining the pressure drop across the heat pump, an ultrasonic water flow-meter, and entering and leaving water temperatures. In addition, entering and discharge air temperatures to and from the heat pump as well as outdoor air temperatures will be monitored and the electric boiler will be equipped with a watt transducer. A schematic of the instrument rig for the heat pump is shown in Figure 7.1.

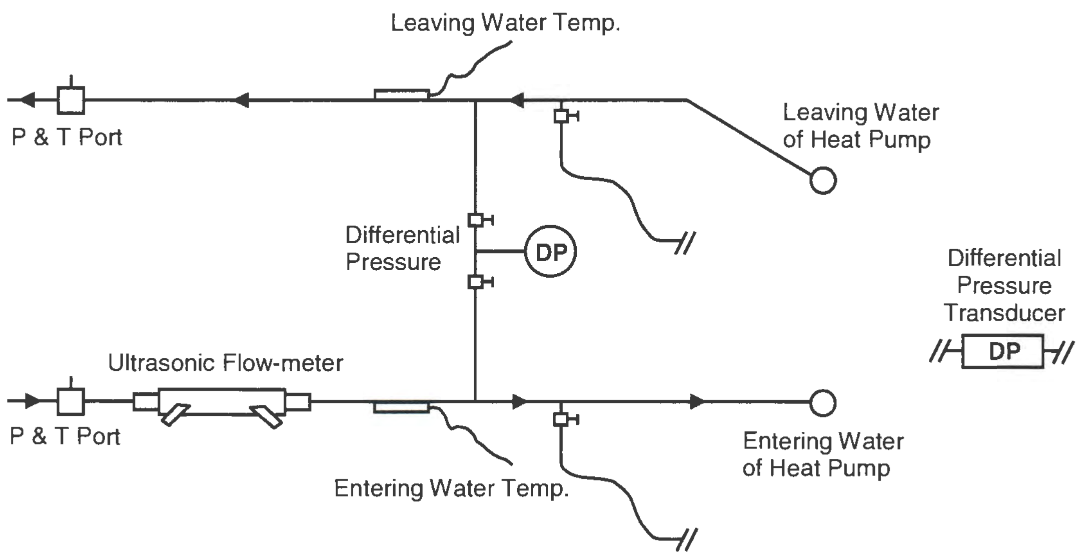

Figure 7.1: Proposed heat pump instrument rig

In conclusion, the work performed on this project changed the outcome of the project by persuading the owners to choose approach 2 instead of 3. Approach 2 utilizes a ground-source heat pump for the forced air distribution system and an electric boiler for the hydronic heating to the radiant floor system. Approach 3 also proposed a ground-source heat pump for the forced air distribution system but proposed a liquid-propane boiler for the radiant floor system; this system was recommended by the contractor. By choosing approach 2, the owners will see significant savings in comparison to the contractor's recommendations and to conventional methods. Therefore, the work done on this project proved to be beneficial to the owners and was considered to be a success overall. Further, a significant amount of knowledge in regards to the scope of this study and some practical experience in the area of building energy was attained. Hopefully, after an evaluation of this report, others will have attained the basic knowledge and tools allowing them to perform this analysis on their own home, and further, become an advocate of geothermal technology.

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APPENDIX A - Area Calculations for Various Construction Types

Table A.1: Upper level glass areas

Upper Level	Area of glass (ft^2)			
Room	SW	SE	NE	NW
Bath off garage	0	0	8	0
Entryway off garage	0	0	0	0
Pantry	0	0	0	0
Kitchen	0	0	62	0
Sunroom	0	55	55	0
Dining room	0	55	0	0
Great-room	0	96	0	0
Foyer	0	0	0	46
Master bedroom	9	55	0	0
Hall off master bedroom	0	0	0	0
Bath off master bedroom	19	4	0	0
Closet off master bedroom	0	0	0	24
Laundry room	0	0	0	0

Table A.2: Lower level glass areas

Lower Level	Area of glass (ft^2)			
Room	SW	SE	NE	NW
Storage room 1	0	0	0	0
Storage room 2	0	0	0	0
Game room	0	0	0	0
Bar area	0	0	0	0
Den	0	44	0	0
Sunken rec. room	0	136	0	0
Hall	0	0	0	0
Mechanical room	0	0	0	0
Bedroom 2	9	44	0	0
Bedroom 3	22	0	0	0
Bath	0	0	0	0

Table A.3: Upper level window frames areas

Upper Level	Window Frame Area (ft^2)			
Room	SW	SE	NE	NW
Bath off garage	0	0	5	0
Entryway off garage	0	0	0	0
Pantry	0	0	0	0
Kitchen	0	0	26	0
Sunroom	0	16	17	0
Dining room	0	16	0	0
Great-room	0	14	0	0
Foyer	0	0	0	5
Master bedroom	6	17	0	0
Hall off master bedroom	0	0	0	0
Bath off master bedroom	8	2	0	0
Closet off master bedroom	0	0	0	6
Laundry room	0	0	0	0

Table A.4: Lower level window frames areas

Lower Level	Window Frame Area (ft^2)			
Room	SW	SE	NE	NW
Storage room 1	0	0	0	0
Storage room 2	0	0	0	0
Game room	0	0	0	0
Bar area	0	0	0	0
Den	0	13	0	0
Sunken rec. room	0	14	0	0
Hall	0	0	0	0
Mechanical room	0	0	0	0
Bedroom 2	6	7	0	0
Bedroom 3	8	0	0	0
Bath	0	0	0	0

Table A.5: Upper level ceiling and floor areas

Upper Level	Ceiling and Floor Area (ft^2)
Room	
Bath off garage	28
Entryway off garage	45
Pantry	67
Kitchen	262
Sunroom	193
Dining room	177
Great-room	321
Foyer	265
Master bedroom	228
Hall off master bedroom	35
Bath off master bedroom	174
Closet off master bedroom	147
Laundry room	69

Table A.6: Lower level floor and ceiling areas

Lower Level Room	Floor and Ceiling Area (ft^2)
Storage room 1	156
Storage room 2	63
Game room	316
Bar area	334
Den	195
Sunken rec. room	556
Hall	141
Mechanical room	56
Bedroom 2	263
Bedroom 3	275
Bath	69

Table A.7: Upper level above grade exterior wall areas

Upper Level	Wall (ft^2)			
Room	NE	SE	SW	NW
Bath off garage	47	58	0	0
Entryway off garage	23	0	0	0
Pantry	0	0	22	0
Kitchen	41	0	0	0
Sunroom	82	65	20	71
Dining room	0	63	0	0
Great-room	28	161	28	0
Foyer	0	0	47	126
Master bedroom	0	65	147	0
Hall off master bedroom	0	0	0	0
Bath off master bedroom	0	47	112	55
Closet off master bedroom	106	0	98	132
Laundry room	0	0	0	0

Table A.8: Lower level above grade exterior exposed wall areas

Lower Level	Wall (ft^2)			
Room	NE	SE	SW	NW
Storage room 1	0	0	0	0
Storage room 2	0	0	0	0
Game room	0	0	0	0
Bar area	0	0	0	0
Den	0	59	18	0
Sunken rec. room	19	169	19	0
Hall	0	0	0	0
Mechanical room	0	0	0	0
Bedroom 2	0	112	176	0
Bedroom 3	0	0	132	47
Bath	0	0	0	0

Table A.9: Lower level below grade exterior wall areas

Lower Level	Wall (ft^2)				
Room	NE	SE	SW	NW	Total
Storage room 1	101	0	18	132	251
Storage room 2	0	0	82	65	147
Game room	169	48	127	0	345
Bar area	156	0	22	0	178
Den	136	0	0	88	224
Sunken rec. room	0	0	0	0	0
Hall	0	0	0	81	81
Mechanical room	0	0	0	75	75
Bedroom 2	0	0	0	0	0
Bedroom 3	92	0	0	136	229
Bath	0	0	0	0	0

Table A.10: Upper level exposed partitions to unconditioned space areas

Upper Level	Partition (ft^2)			
Room	NE	SE	SW	NW
Bath off garage	0	0	0	58
Entryway off garage	0	0	0	42
Pantry	0	0	0	112
Kitchen	0	0	0	0
Sunroom	0	0	0	0
Dining room	0	0	0	0
Great-room	0	0	0	0
Foyer	0	0	0	0
Master bedroom	0	0	0	0
Hall off master bedroom	0	0	0	0
Bath off master bedroom	0	0	0	0
Closet off master bedroom	0	0	0	0
Laundry room	0	0	0	0

Table A.11: Upper level ceiling height and room volumes

Upper Level Room	Ceiling Ht. (ft)	Volume (ft^3)
Bath off garage	10.1	291
Entryway off garage	10.1	463
Pantry	10.1	684
Kitchen	10.1	2646
Sunroom	10.1	1951
Dining room	10.1	1787
Great-room	14.1	4526
Foyer	11.1	2945
Master bedroom	10.1	2307
Hall off master bedroom	10.1	357
Bath off master bedroom	10.1	1756
Closet off master bedroom	10.1	1490
Laundry room	10.1	702

Table A.12: Lower level ceiling height and room volumes

Lower Level Room	Ceiling Ht. (ft)	Volume (ft^3)
Storage room 1	8.8	1368
Storage room 2	8.8	554
Game room	8.8	2778
Bar area	8.8	2936
Den	8.8	1714
Sunken rec. room	10.0	5537
Hall	8.8	1236
Mechanical room	8.8	492
Bedroom 2	8.8	2308
Bedroom 3	8.8	2416
Bath	8.8	604

Table A.13: Upper level rough opening areas

Upper Level	Glass and Window (ft^2)			
Room	SW	SE	NE	NW
Bath off garage	0	0	12	0
Entryway off garage	0	0	0	0
Pantry	0	0	0	0
Kitchen	0	0	105	0
Sunroom	0	72	72	0
Dining room	0	72	0	0
Great-room	0	111	0	0
Foyer	0	0	0	63
Master bedroom	15	72	0	0
Hall off master bedroom	0	0	0	0
Bath off master bedroom	36	6	0	0
Closet off master bedroom	0	0	0	30
Laundry room	0	0	0	0

Table A.14: Lower level rough opening areas

Lower Level	Glass and Window (ft^2)			
Room	SW	SE	NE	NW
Storage room 1	0	0	0	0
Storage room 2	0	0	0	0
Game room	0	0	0	0
Bar area	0	0	0	0
Den	0	57	0	0
Sunken rec. room	0	150	0	0
Hall	0	0	0	0
Mechanical room	0	0	0	0
Bedroom 2	15	51	0	0
Bedroom 3	30	0	0	0
Bath	0	0	0	0

APPENDIX B - R-Value and U-Value Calculations for the Design Cooling Load

Table B.15: U-Value calculation for the above grade exterior exposed wall single and insulation portion

Above Grade Exterior Exposed Wall Shingle and Insulation Portion		
Material	Thickness (in.)	R-Value (ft^2*F*hr)/Btu
Shingle Siding, 1/4" lapping	N/A	0.21
Weather Barrier	N/A	0
Sheathing (D.F. Plywood)	1/2"	0.62
R-19 Insulation	N/A	19
Gypsum or plaster board	1/2"	0.45
Inside Film Resistance	N/A	0.68
Summer - Outside Film Res.	N/A	0.25
U-Value (Btu/(ft^2*F*hr)	0.0471	21.21

Table B.16: U-Value calculation for the above grade exterior exposed wall single and stud portion

Above Grade Exterior Exposed Wall Shingle and Stud Portion		
Material	Thickness (in.)	R-Value (ft^2*F*hr)/Btu
Shingle Siding, 1/4" lapping	N/A	0.21
Weather Barrier	N/A	0
Sheathing (D.F. Plywood)	1/2"	0.62
Douglas fir 2"x6" stud	5-1/2"	5.64
Gypsum or plaster board	1/2"	0.45
Inside Film Resistance	N/A	0.68
Summer - Outside Film Res.	N/A	0.25
U-Value (Btu/(ft^2*F*hr)	0.1274	7.85

Table B.17: U-Value calculation for the above grade exterior exposed wall stone and insulation portion

Above Grade Exterior Exposed Wall Stone and Insulation portion		
Material	Thickness (in.)	R-Value (ft^2*F*hr)/Btu
Centurion Stone Veneer	3	0.6
Weather Barrier	N/A	0
Sheathing (D.F. Plywood)	1/2"	0.62
R-19 Insulation	N/A	19
Gypsum or plaster board	1/2"	0.45
Inside Film Resistance	N/A	0.68
Summer - Outside Film Res.	N/A	0.25
U-Value (Btu/(ft^2*F*hr)	0.0463	21.6

Table B.18: U-Value calculation for the above grade exterior exposed wall stone and stud portion

Above Grade Exterior Exposed Wall Stone and Stud Portion		
Material	Thickness (in.)	R-Value (ft^2*F*hr)/Btu
Centurion Stone Veneer	3"	0.6
Weather Barrier	N/A	0
Sheathing (D.F. Plywood)	1/2"	0.62
Douglas fir 2"x6" stud	5-1/2"	5.64
Gypsum or plaster board	1/2"	0.45
Inside Film Resistance	N/A	0.68
Summer - Outside Film Res.	N/A	0.25
U-Value (Btu/(ft^2*F*hr)	0.1214	8.24

Table B.19: U-Value calculation for the exposed floor on the lower level

Exposed Floor in the Lower Level		
Material	Thickness (in.)	R-Value (ft^2*F*hr)/Btu
Concrete	4	0.36
Insulation foam board	2	10
U-Value (Btu/(ft^2*F*hr)	0.0965	0.36

Table B.20: U-Value calculation for the window frames

Window Frames		
Material	Thickness (in.)	R-Value (ft^2*F*hr)/Btu
Oak	5	4.225
Inside Film Resistance	N/A	0.68
Summer - Outside Film Res.	N/A	0.25
U-Value (Btu/(ft^2*F*hr)	0.1940	5.16

Table B.21: U-Value calculation for the doors

Doors		
Material	Thickness (in.)	R-Value (ft^2*F*hr)/Btu
Oak	2	1.69
Inside Film Resistance	N/A	0.68
Summer - Outside Film Res.	N/A	0.25
U-Value (Btu/(ft^2*F*hr)	0.3817	2.62

Table B.22: U-Value calculation for the unconditioned partition to the garage insulation portion

Unconditioned Partition to Garage Insulation Portion		
Material	Thickness (in.)	R-Value (ft^2*F*hr)/Btu
R-19 Insulation	N/A	19
Gypsum or plaster board	1/2"	0.45
Gypsum or plaster board	1/2"	0.45
Inside Film Resistance	N/A	0.68
U-Value (Btu/(ft^2*F*hr)	0.0486	20.58

Table B.23: U-Value calculation for the unconditioned partition to the garage stud portion

Unconditioned Partition to Garage Stud Portion		
Material	Thickness (in.)	R-Value (ft^2*F*hr)/Btu
Douglas fir 2"x6" stud	5-1/2"	5.64
Gypsum or plaster board	1/2"	0.45
Gypsum or plaster board	1/2"	0.45
Inside Film Resistance	N/A	0.68
U-Value (Btu/(ft^2*F*hr)	0.1386	7.22

Table B.24: U-Value calculation for the exterior basement walls below grade insulation portion

Exterior Basement Walls Below Grade Insulation Portion		
Material	Thickness (in.)	R-Value (ft^2*F*hr)/Btu
Concrete	8	0.72
Insulation	N/A	13
Gypsum or plaster board	1/2"	0.45
U-Value (Btu/(ft^2*F*hr)	0.0706	14.17

Table B.25: U-Value calculation for the exterior basement walls below grade stud portion

Exterior Basement Walls Below Grade Stud Portion		
Material	Thickness (in.)	R-Value (ft^2*F*hr)/Btu
Concrete	8	0.72
Douglas fir 2"x4" stud	3-1/2"	3.59
Gypsum or plaster board	1/2"	0.45
U-Value (Btu/(ft^2*F*hr)	0.2102	4.76

Table B.26: U-Value calculation for the upper level below roof insulation portion

Upper Level Ceiling Below Roof Insulation Portion		
Material	Thickness (in.)	R-Value (ft^2*F*hr)/Btu
Gypsum or plaster board	1/2"	0.45
R-38 Insulation	N/A	38
U-Value (Btu/(ft^2*F*hr)	0.0260	38.45

Table B.27: U-Value calculation for the lower level exterior exposed wall section stone and insulation portion

Lower Level Exterior Exposed Wall Stone and Insulation Portion		
Material	Thickness (in.)	R-Value (ft^2*F*hr)/Btu
Centurion Stone Veneer	3	0.6
Weather Barrier	N/A	0
Sheathing (D.F. Plywood)	1/2"	0.62
R-13 Insulation	N/A	13
Gypsum or plaster board	1/2"	0.45
Inside Film Resistance	N/A	0.68
Summer - Outside Film Res.	N/A	0.25
U-Value (Btu/(ft^2*F*hr)	0.0641	15.6

Table B.28: U-Value calculation for the lower level exterior exposed wall section stone and stud portion

Lower Level Exterior Exposed Wall Stone and Stud Portion		
Material	Thickness (in.)	R-Value (ft^2*F*hr)/Btu
Centurion Stone Veneer	3"	0.6
Weather Barrier	N/A	0
Sheathing (D.F. Plywood)	1/2"	0.62
Douglas fir 2"x4" stud	3.5	3.59
Gypsum or plaster board	1/2"	0.45
Inside Film Resistance	N/A	0.68
Summer - Outside Film Res.	N/A	0.25
U-Value (Btu/(ft^2*F*hr)	0.1616	6.19

Table B.29: U-Value calculation for the section between floors insulation portion

Between Floors Wall Section Insulation Portion		
Material	Thickness (in.)	R-Value (ft^2*F*hr)/Btu
Centurion Stone Veneer	3"	0.6
Sheathing (D.F. Plywood)	1/2"	0.62
Insulation	N/A	19
Gypsum or plaster board	1/2"	0.45
U-Value (Btu/(ft^2*F*hr)	0.0484	20.67

APPENDIX C - R-Value and U-Value Calculations for the Heating Load

Table C.30: U-Value calculation for the upper level wall Section shingle and insulation portion

Upper Level Wall Section Shingle and Insulation Portion		
Material	Thickness (in.)	R-Value $(ft^2*F*hr)/Btu$
Shingle Siding, 1/4" lapping	N/A	0.21
Weather Barrier	N/A	0
Sheathing (D.F. Plywood)	1/2"	0.62
R-19 Insulation	N/A	19
Gypsum or plaster board	1/2"	0.45
Inside Film Resistance	N/A	0.68
Winter - Outside Film Res.	N/A	0.17
U-Value $(Btu/(ft^2*F*hr)$	0.0473	21.13

Table C.31: U-Value calculation for the upper level wall section shingle and stud portion

Upper Level Wall Section Shingle and Stud Portion		
Material	Thickness (in.)	R-Value $(ft^2*F*hr)/Btu$
Shingle Siding, 1/4" lapping	N/A	0.21
Weather Barrier	N/A	0
Sheathing (D.F. Plywood)	1/2"	0.62
Douglas fir 2"x6" stud	5-1/2"	5.64
Gypsum or plaster board	1/2"	0.45
Inside Film Resistance	N/A	0.68
Winter - Outside Film Res.	N/A	0.17
U-Value $(Btu/(ft^2*F*hr)$	0.1287	7.77

Table C.32: U-Value calculation for the upper level wall section stone and insulation portion

Upper Level Wall Section Stone and Insulation Portion		
Material	Thickness (in.)	R-Value $(ft^2*F*hr)/Btu$
Centurion Stone Veneer	3	0.6
Weather Barrier	N/A	0
Sheathing (D.F. Plywood)	1/2"	0.62
R-19 Insulation	N/A	19
Gypsum or plaster board	1/2"	0.45
Inside Film Resistance	N/A	0.68
Winter - Outside Film Res.	N/A	0.17
U-Value $(Btu/(ft^2*F*hr)$	0.0465	21.52

Table C.33: U-Value calculation for the upper level wall section stone and stud portion

Upper Level Wall Section Stone and Stud Portion		
Material	Thickness (in.)	R-Value $(ft^2*F*hr)/Btu$
Centurion Stone Veneer	3"	0.6
Weather Barrier	N/A	0
Sheathing (D.F. Plywood)	1/2"	0.62
Douglas fir 2"x6" stud	5-1/2"	5.64
Gypsum or plaster board	1/2"	0.45
Inside Film Resistance	N/A	0.68
Winter - Outside Film Res.	N/A	0.17
U-Value $(Btu/(ft^2*F*hr)$	0.1226	8.16

Table C.34: U-Value calculation for the lower level exposed floors

Lower Level Exposed Floors		
Material	Thickness (in.)	R-Value $(ft^2*F*hr)/Btu$
Concrete	4	0.36
Insulation foam board	2	10
U-Value $(Btu/(ft^2*F*hr)$	0.0965	10.36

Table C.35: U-Value calculation for the window frames

Window Frames		
Material	Thickness (in.)	R-Value $(ft^2*F*hr)/Btu$
Oak	5	4.2
Inside Film Resistance	N/A	0.68
Winter - Outside Film Res.	N/A	0.17
U-Value $(Btu/(ft^2*F*hr)$	0.1970	5.08

Table C.36: U-Value calculation for the doors

Doors		
Material	Thickness (in.)	R-Value $(ft^2*F*hr)/Btu$
Oak	2	1.69
Inside Film Resistance	N/A	0.68
Winter - Outside Film Res.	N/A	0.17
U-Value $(Btu/(ft^2*F*hr)$	0.3937	2.54

Table C.37: U-Value calculation for upper level partition to an unconditioned space insulation portion

Upper Level Wall Section to Partition to Garage Insulation Portion		
Material	Thickness (in.)	R-Value $(ft^2*F*hr)/Btu$
R-19 Insulation	N/A	19
Gypsum or plaster board	1/2"	0.45
Gypsum or plaster board	1/2"	0.45
Inside Film Resistance	N/A	0.68
U-Value $(Btu/(ft^2*F*hr)$	0.0486	20.58

Table C.38: U-Value calculation for upper level partition to an unconditioned space stud portion

Upper Level Wall Section Partition to Garage Stud Portion		
Material	Thickness (in.)	R-Value $(ft^2*F*hr)/Btu$
Douglas fir 2"x6" stud	5-1/2"	5.64
Gypsum or plaster board	1/2"	0.45
Gypsum or plaster board	1/2"	0.45
Inside Film Resistance	N/A	0.68
U-Value $(Btu/(ft^2*F*hr)$	0.1386	7.22

Table C.39: U-Value calculation for lower level exterior below grade wall section insulation portion

Lower Level Wall Section Exterior Below Grade Insulation Portion		
Material	Thickness (in.)	R-Value $(ft^2*F*hr)/Btu$
Concrete	8	0.72
Insulation	N/A	13
Gypsum or plaster board	1/2"	0.45
U-Value $(Btu/(ft^2*F*hr)$	0.0706	14.17

Table C.40: U-Value calculation for lower level exterior below grade wall section stud portion

Lower Level Exterior Below Grade Wall Section Stud Portion		
Material	Thickness (in.)	R-Value $(ft^2*F*hr)/Btu$
Concrete	8	0.72
Douglas fir 2"x4" stud	3-1/2"	3.59
Gypsum or plaster board	1/2"	0.45
U-Value $(Btu/(ft^2*F*hr)$	0.2102	4.76

Table C.41: U-Value calculation for the upper level ceiling below roof insulation portion

Upper Level Ceiling Below Roof Insulation Portion		
Material	Thickness (in.)	R-Value (ft^2*F*hr)/Btu
Gypsum or plaster board	1/2"	0.45
R-38 Insulation	N/A	38
U-Value (Btu/(ft^2*F*hr)	0.0260	38.45

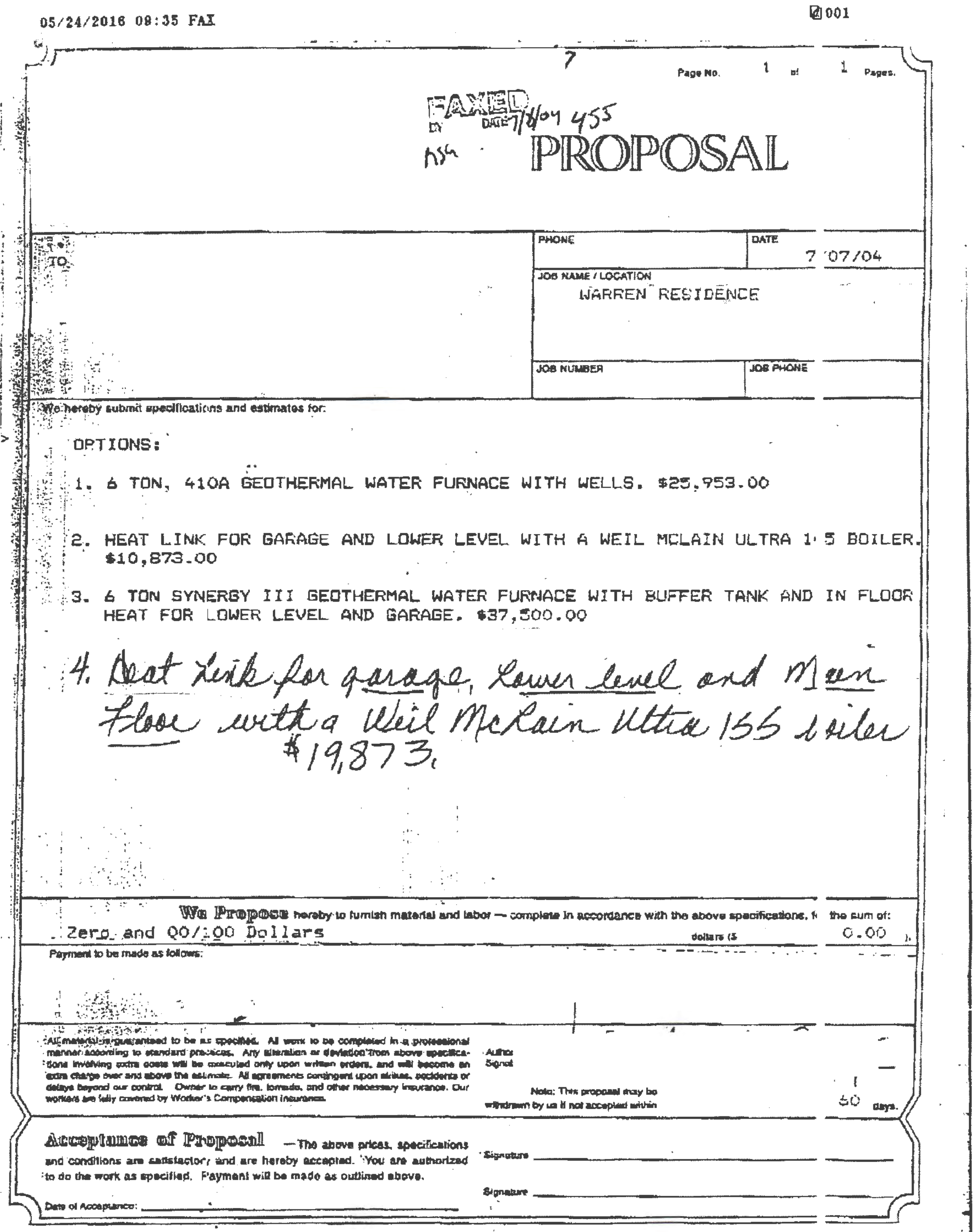
Table C.42: U-Value calculation for the glass

Glass		
Contributor	Thickness	R-Value (ft^2*F*hr)/Btu
Glass	N/A	3.13
U-Value (Btu/(ft^2*F*hr)	0.3200	3.13

Table C.43: U-Value calculation for the wall section between floors insulation portion

Wall Section Between Floors Insulation Portion		
Material	Thickness (in.)	R-Value (ft^2*F*hr)/Btu
Centurion Stone Veneer	3"	0.6
Inside film	N/A	0.68
Outside film - Winter	N/A	0.17
Sheathing (D.F. Plywood)	1/2"	0.62
Insulation	N/A	13
Gypsum or plaster board	1/2"	0.45
U-Value (Btu/(ft^2*F*hr)	0.0644	15.52

APPENDIX D - Bid Proposals from Subcontractors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FAXED BY DATE 7/8/04 455

PROPOSAL

TO:

PHONE

DATE 7/07/04

JOB NAME / LOCATION WARREN RESIDENCE

JOB NUMBER

JOB PHONE

We hereby submit specifications and estimates for:

OPTIONS:

1. 6 TON, 410A GEOTHERMAL WATER FURNACE WITH WELLS. $25,953.00

2. HEAT LINK FOR GARAGE AND LOWER LEVEL WITH A WEIL MCLAIN ULTRA 155 BOILER. $10,873.00

3. 6 TON SYNERGY III GEOTHERMAL WATER FURNACE WITH BUFFER TANK AND IN FLOOR HEAT FOR LOWER LEVEL AND GARAGE. $37,500.00

4. Heat Link for garage, Lower level and Main Floor with a Weil McLain Ultra 155 boiler $19,873.

We Propose hereby to furnish material and labor — complete in accordance with the above specifications, for the sum of: Zero and 00/100 Dollars dollars ($ 0.00).

Payment to be made as follows:

All material is guaranteed to be as specified. All work to be completed in a professional manner according to standard practices. Any alteration or deviation from above specifications involving extra costs will be executed only upon written orders, and will become an extra charge over and above the estimate. All agreements contingent upon strikes, accidents or delays beyond our control. Owner to carry fire, tornado, and other necessary insurance. Our workers are fully covered by Worker's Compensation Insurance.

Authorized Signature

Note: This proposal may be withdrawn by us if not accepted within 60 days.

Acceptance of Proposal — The above prices, specifications and conditions are satisfactory and are hereby accepted. You are authorized to do the work as specified. Payment will be made as outlined above.

Signature

Signature

Date of Acceptance:

Figure D.1: First proposal submitted by subcontractor a – page 1

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03/25/2009 15:24 5152746403 CITY WIDE HEATING ☑001 PAGE 02

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PROPOSAL

TO S	PHONE 5
	DATE 5/2 /04
	JOB NAME / LOCATION WARREN RESIDENCE
	JOB NUMBER
	JOB PHONE

We hereby submit specifications and estimates for:

INSTALLATION OF THE FOLLOWING EQUIPMENT:

1- BRYANT 92% GAS FURNACE, MODEL 340MAV060120, 120,000 B.T.U.
1- BRYANT 5 TON EVAPORATOR COIL.
1- BRYANT 5 TON, 12. S.E.E.R. CONDENSING UNIT, MODEL 563CNX060.
1- HONEYWELL HEATING-COOLING PROGRAMMABLE THERMOSTAT.
1- APRILAIRE MODEL 600 HUMIDIFER.

INSTALLATION OF ALL NECESSARY SHEET METAL, REGISTERS, GRILLS, DRYER VENT, BATH VENTING, GAS LINE TO FURNACE, DRYER, RANGE AND FIREPLACE. ALL LABOR AND MATERIAL TO MAKE JOB COMPLETE. BOTH LABOR AND MATERIAL TO BE OF THE HIGHEST QUALITY AVAILABLE.

TWENTY YEAR WARRANTY ON HEAT EXCHANGER BY MFG.
FIVE YEAR WARRANTY ON COMPRESSOR BY MFG.
FIVE YEAR WARRANTY ON BRYANT PARTS BY MFG.
ONE YEAR WARRANTY ON ALL OTHER PARTS, LABOR AND MATERIAL.

We Propose hereby to furnish material and labor — complete in accordance with the above specifications, for the sum of:
Twelve Thousand Seven Hundred and 00/100 Dollars dollars ($ 12,7 0.00)

Payment to be made as follows:

All material is guaranteed to be as specified. All work to be completed in a professional manner according to standard practices. Any alteration or deviation from above specifications involving extra costs will be executed only upon written orders, and will become an extra charge over and above the estimate. All agreements contingent upon strikes, accidents or delays beyond our control. Owner to carry fire, tornado, and other necessary insurance. Our workers are fully covered by Worker's Compensation Insurance.

Authorized Signature

Note: This proposal may be withdrawn by us if not accepted within 60 days.

Acceptance of Proposal — The above prices, specifications and conditions are satisfactory and are hereby accepted. You are authorized to do the work as specified. Payment will be made as outlined above.

Signature ____________________

Signature ____________________

Date of Acceptance: ____________________

Figure D.2: First proposal submitted by subcontractor a – page 2

08/11/2004 16:39 5152746403 CITY WIDE HEATING PAGE 02

Page No. 1 of 1 Pages.

PROPOSAL

TO	PHONE	DATE 8/11/04
	JOB NAME / LOCATION WARREN RESIDENCE	
	JOB NUMBER	JOB PHONE

We hereby submit specifications and estimates for:

INSTALLATION OF THE FOLLOWING EQUIPMENT:

1- WATER FURNACE PREMIER II "E", R410A. 2 STAGE. 6 TON MODEL E072TL101NBDSSA.
1- 20 K.W. HEATER.
1- WATER FURNACE THERMOSTAT.
1- DESUPERHEATER.
VERTICAL WELLS INCLUDED.

INSTALLATION OF ALL NECESSARY SHEET METAL, REGISTERS, GRILLS, DRYER VENT, BATH VENTING AND GAS LINE TO FURNACE. ALL LABOR AND MATERIAL TO MAKE JOB COMPLETE. BOTH LABOR AND MATERIAL TO BE OF THE HIGHEST QUALITY AVAILABLE.

TEN YEAR PARTS AND LABOR WARRANTY ON WATER FURNACE BY MFG.
ONE YEAR WARRANTY ON ALL OTHER PARTS, LABOR AND MATERIAL.

OPTION: HEAT LINK FOR GARAGE AND LOWER LEVEL WITH A WEIL MCLAIN ULTRA 105 BOILER. $10,873.00

GUARDIAN ERV WITH DEHUMIDIFICATION CONTROL. $1,600.00

We Propose hereby to furnish material and labor — complete in accordance with the above specifications, for the sum of:
Twenty Six Thousand and 00/100 Dollars dollars ($ 26,000.00).

Payment to be made as follows:

All material is guaranteed to be as specified. All work to be completed in a professional manner according to standard practices. Any alteration or deviation from above specifications involving extra costs will be executed only upon written orders, and will become an extra charge over and above the estimate. All agreements contingent upon strikes, accidents or delays beyond our control. Owner to carry fire, tornado, and other necessary insurance. Our workers are fully covered by Worker's Compensation Insurance.

Authorized Signature ______

Note: This proposal may be withdrawn by us if not accepted within 60 days.

Acceptance of Proposal — The above prices, specifications and conditions are satisfactory and are hereby accepted. You are authorized to do the work as specified. Payment will be made as outlined above.

Signature ______

Date of Acceptance: ______ Signature ______

Figure D.3: Revised proposal submitted by subcontractor a

- Septics
- Trenching
- Back Hoe Work
- Waterlines
- Mechanical Waste Sytems
- Full Fleet Service Technicians

Ryan & Terry Warren

August 18, 2004

RE: Geothermal New Construction
WaterFurnace E072 & EW060 Series Installation to Include—

- Set units on pad
- New supply plenum
- New return drop and Spacegard air filtration
- Duct system for home (separate supply trunks for up and downstairs), including up to 24-1st and 20 basement supply runs
- Carrier Comfort Zone 2 zone system
- Bath & Dryer venting
- 24v wiring
- Condensate to floor drain
- 8 ton vertical loop installation
- Loop installation does not include final grade
- Customer to provide 230v
- Desuperheater connection to customer's electric water heater
- Customer to have all radiant floor piping in, and Morrell to do all above work for two basement and one garage zone—includes 50 gallon storage tank, all stats, Wirsbo manifolds, and copper piping
- Applicable permits
- Installation based on typical floor joist construction

Installation Total **53,347.00**

OPTION NUMBERS ______________________________

OPTION SUBTOTAL __________

TOTAL INSTALLATION PRICE __________

Thank you for your time and consideration

Figure D.4: Proposal submitted by subcontractor b – option 1

- Septics
- Trenching
- Back Hoe Work
- Waterlines
- Mechanical Waste Sytems
- Full Fleet Service Technicians

Ryan & Terry Warren

August 18, 2004

RE: Geothermal New Construction
WaterFurnace Synergy RTV066 & EW020 Series Installation to Include—

- Set units on pad
- New supply plenum
- New return drop and Spacegard air filtration
- Duct system for home (separate supply trunks for up and downstairs), including up to 24-1st and 20 basement supply runs
- Carrier Comfort Zone 2 zone system
- Bath & Dryer venting
- 24v wiring
- Condensate to floor drain
- 7 ton vertical loop installation
- Loop installation does not include final grade
- Customer to provide 230v
- EW020 connection to customer's electric water heater
- Customer to have all radiant floor piping in, and Morrell to do all above work for two basement and one garage zone—includes 50 gallon storage tank, all stats, Wirsbo manifolds, and copper piping
- Applicable permits
- Installation based on typical floor joist construction

Installation Total **49,251.00**

OPTION NUMBERS ______________________

OPTION SUBTOTAL ________

TOTAL INSTALLATION PRICE ________

Thank you for your time and consideration

Figure D.5: Proposal submitted by subcontractor b – option 2

APPENDIX E - Annual Loan Payments for each Approach

Table E.44: Annual loan payments for approach 1

Approach 1 - E060 & EW040				
End of Year	Payment to Bank ($)	Interest Paid ($)	Principal Payment ($)	Loan Balance ($)
0	$0	$0	$0	$50,000
1	$6,665	$2,800	$3,865	$46,135
2	$6,665	$2,584	$4,082	$42,053
3	$6,665	$2,355	$4,310	$37,743
4	$6,665	$2,114	$4,552	$33,191
5	$6,665	$1,859	$4,807	$28,385
6	$6,665	$1,590	$5,076	$23,309
7	$6,665	$1,305	$5,360	$17,949
8	$6,665	$1,005	$5,660	$12,289
9	$6,665	$688	$5,977	$6,312
10	$6,665	$353	$6,312	$0
Total	$66,652	$16,652	$50,000	N/A

Table E.45: Annual loan payments for approach 2

Approach 2 - E060 & Electric Boiler				
End of Year	Payment to Bank ($)	Interest Paid ($)	Principal Payment ($)	Loan Balance ($)
0	$0	$0	$0	$35,900
1	$4,786	$2,010	$2,775	$33,125
2	$4,786	$1,855	$2,931	$30,194
3	$4,786	$1,691	$3,095	$27,099
4	$4,786	$1,518	$3,268	$23,831
5	$4,786	$1,335	$3,451	$20,380
6	$4,786	$1,141	$3,644	$16,736
7	$4,786	$937	$3,848	$12,887
8	$4,786	$722	$4,064	$8,823
9	$4,786	$494	$4,292	$4,532
10	$4,786	$254	$4,532	$0
Total	$47,856	$11,956	$35,900	N/A

Table E.46: Annual loan payments for approach 3

Approach 3 - E060 & Liquid Propane Boiler				
End of Year	Payment to Bank ($)	Interest Paid ($)	Principal Payment ($)	Loan Balance ($)
0	$0	$0	$0	$37,000
1	$4,932	$2,072	$2,860	$34,140
2	$4,932	$1,912	$3,020	$31,119
3	$4,932	$1,743	$3,190	$27,930
4	$4,932	$1,564	$3,368	$24,561
5	$4,932	$1,375	$3,557	$21,005
6	$4,932	$1,176	$3,756	$17,249
7	$4,932	$966	$3,966	$13,282
8	$4,932	$744	$4,188	$9,094
9	$4,932	$509	$4,423	$4,671
10	$4,932	$262	$4,671	$0
Total	$49,323	$12,323	$37,000	N/A

Table E.47: Annual loan payments for approach 4

Approach 4 – Liquid Propane Boiler, Furnace, Air-Conditioner				
End of Year	Payment to Bank ($)	Interest Paid ($)	Principal Payment ($)	Loan Balance ($)
0	$0	$0	$0	$23,700
1	$3,159	$1,327	$1,832	$21,868
2	$3,159	$1,225	$1,935	$19,933
3	$3,159	$1,116	$2,043	$17,890
4	$3,159	$1,002	$2,157	$15,733
5	$3,159	$881	$2,278	$13,454
6	$3,159	$753	$2,406	$11,048
7	$3,159	$619	$2,541	$8,508
8	$3,159	$476	$2,683	$5,825
9	$3,159	$326	$2,833	$2,992
10	$3,159	$168	$2,992	$0
Total	$31,593	$7,893	$23,700	N/A

APPENDIX F - Annual Operation Costs for each Approach

Table F.48: Annual operation cost for approach 1

Approach 1 - E060 and EW040			
Year	Year	Annual Heating Cost ($)	Annual Cooling Cost ($)
2004	0	$0	$0
2005	1	$310	$128
2006	2	$310	$128
2007	3	$310	$128
2008	4	$310	$128
2009	5	$310	$128
2010	6	$354	$145
2011	7	$363	$149
2012	8	$375	$154
2013	9	$391	$161
2014	10	$406	$167
2015	11	$419	$172
2016	12	$434	$179
2017	13	$447	$184
2018	14	$462	$190
2019	15	$475	$195
2020	16	$487	$200
2021	17	$506	$208
2022	18	$521	$214
2023	19	$533	$219
2024	20	$552	$227
2025	21	$571	$235
2026	22	$589	$242
2027	23	$608	$250
2028	24	$627	$258
2029	25	$645	$265

Table F.49: Annual operation cost for approach 2

Approach 2 - E060 and Electric Boiler			
Year	Year	Annual Heating Cost ($)	Annual Cooling Cost ($)
2004	0	$0	$0
2005	1	$616	$128
2006	2	$616	$128
2007	3	$616	$128
2008	4	$616	$128
2009	5	$616	$128
2010	6	$616	$145
2011	7	$702	$149
2012	8	$721	$154
2013	9	$746	$161
2014	10	$776	$167
2015	11	$807	$172
2016	12	$832	$179
2017	13	$863	$184
2018	14	$887	$190
2019	15	$918	$195
2020	16	$943	$200
2021	17	$967	$208
2022	18	$1,004	$214
2023	19	$1,035	$219
2024	20	$1,060	$227
2025	21	$1,097	$235
2026	22	$1,134	$242
2027	23	$1,171	$250
2028	24	$1,208	$258
2029	25	$1,245	$265

Table F.50: Annual operation cost for approach 3

Approach 3 - E060 and Liquid Propane Boiler			
Year	Year	Annual Heating Cost ($)	Annual Cooling Cost ($)
2004	0	$0	$0
2005	1	$719	$128
2006	2	$663	$128
2007	3	$658	$128
2008	4	$678	$128
2009	5	$694	$128
2010	6	$714	$145
2011	7	$759	$149
2012	8	$790	$154
2013	9	$819	$161
2014	10	$850	$167
2015	11	$881	$172
2016	12	$915	$179
2017	13	$951	$184
2018	14	$985	$190
2019	15	$1,016	$195
2020	16	$1,049	$200
2021	17	$1,083	$208
2022	18	$1,126	$214
2023	19	$1,162	$219
2024	20	$1,196	$227
2025	21	$1,239	$235
2026	22	$1,283	$242
2027	23	$1,326	$250
2028	24	$1,369	$258
2029	25	$1,412	$265

Table F.51: Annual operation cost for approach 4

Approach 4 - Gas Furnace, CAC, and Liquid Propane Boiler			
Year	Year	Annual Heating Cost ($)	Annual Cooling Cost ($)
2004	0	$0	$0
2005	1	$1,586	$151
2006	2	$1,411	$151
2007	3	$1,396	$151
2008	4	$1,459	$151
2009	5	$1,507	$151
2010	6	$1,570	$151
2011	7	$1,618	$172
2012	8	$1,697	$176
2013	9	$1,760	$182
2014	10	$1,824	$190
2015	11	$1,887	$197
2016	12	$1,966	$203
2017	13	$2,046	$211
2018	14	$2,125	$217
2019	15	$2,188	$224
2020	16	$2,268	$230
2021	17	$2,347	$236
2022	18	$2,442	$245
2023	19	$2,521	$253
2024	20	$2,601	$259
2025	21	$2,696	$268
2026	22	$2,791	$277
2027	23	$2,886	$286
2028	24	$2,981	$295
2029	25	$3,077	$304

APPENDIX G - Annual Tax Savings for each Approach

Table G.52: Annual tax savings for approach 1

Approach 1 - E060 & EW040		
End of Year	Tax Savings ($)	Tax Savings Compared to Conventional ($)
0	$0	$0
1	$1,260	$663
2	$1,163	$612
3	$1,060	$557
4	$951	$500
5	$836	$440
6	$715	$376
7	$587	$309
8	$452	$238
9	$310	$163
10	$159	$84
Total	$7,494	$3,942

Table G.53: Annual tax savings for approach 2

Approach 2 - E060 & Electric Boiler		
End of Year	Tax Savings ($)	Tax Savings Compared to Conventional ($)
0	$0	$0
1	$905	$307
2	$835	$284
3	$761	$259
4	$683	$232
5	$601	$204
6	$514	$175
7	$422	$143
8	$325	$110
9	$222	$76
10	$114	$39
Total	$5,380	$1,828

Table G.54: Annual tax savings for approach 3

Approach 3 - E060 & Liquid Propane Boiler		
End of Year	Tax Savings ($)	Tax Savings Compared to Conventional ($)
0	$0	$0
1	$932	$335
2	$860	$309
3	$784	$282
4	$704	$253
5	$619	$222
6	$529	$190
7	$435	$156
8	$335	$120
9	$229	$82
10	$118	$42
Total	$5,545	$1,993

Table G.55: Annual tax savings for approach 4

Approach 4 – Liquid Propane Boiler, Liquid Propane Furnace, Air-Conditioner		
End of Year	Tax Savings ($)	Tax Savings Compared to Conventional ($)
0	$0	N/A
1	$597	N/A
2	$551	N/A
3	$502	N/A
4	$451	N/A
5	$396	N/A
6	$339	N/A
7	$278	N/A
8	$214	N/A
9	$147	N/A
10	$75	N/A
Total	$3,552	N/A

APPENDIX H - Annual Operation Savings for each Approach

Table H.56: Annual Operation Savings for each approach in comparison to conventional (approach 4)

Annual Operation Savings Compared to Conventional Approach				
Year	Year	Approach 1 Heating and Cooling Savings ($)	Approach 2 Heating and Cooling Savings ($)	Approach 3 Heating and Cooling Savings ($)
2004	0	$0	$0	$0
2005	1	$1,299	$993	$890
2006	2	$1,124	$818	$771
2007	3	$1,108	$802	$761
2008	4	$1,172	$866	$804
2009	5	$1,219	$913	$836
2010	6	$1,222	$959	$861
2011	7	$1,277	$938	$881
2012	8	$1,343	$998	$928
2013	9	$1,391	$1,036	$962
2014	10	$1,440	$1,070	$996
2015	11	$1,493	$1,105	$1,031
2016	12	$1,557	$1,159	$1,076
2017	13	$1,626	$1,210	$1,122
2018	14	$1,690	$1,264	$1,167
2019	15	$1,743	$1,300	$1,202
2020	16	$1,811	$1,355	$1,248
2021	17	$1,870	$1,408	$1,292
2022	18	$1,952	$1,469	$1,347
2023	19	$2,022	$1,520	$1,393
2024	20	$2,081	$1,573	$1,436
2025	21	$2,158	$1,632	$1,490
2026	22	$2,236	$1,692	$1,543
2027	23	$2,314	$1,752	$1,596
2028	24	$2,392	$1,811	$1,650
2029	25	$2,470	$1,871	$1,703

APPENDIX I - Present Value Analysis for each Approach

Table I.57: Annual present values for the net cash flow and overall present worth of the system for approach 1

End of Year	Annual Payment to Bank for Loan ($)	Rebate ($)	Annual Heating Cost ($)	Annual Cooling Cost ($)	Annual Tax Savings ($)	Annual Expenses ($)	Annual Savings ($)	Present Value ($)
0	$0	$2,400	$0	$0	$0	$0	$2,400	$2,400
1	$6,665	$0	$310	$128	$1,260	$7,103	$1,961	-$5,591
2	$6,665	$0	$310	$128	$1,163	$7,103	$1,736	-$5,440
3	$6,665	$0	$310	$128	$1,060	$7,103	$1,666	-$5,296
4	$6,665	$0	$310	$128	$951	$7,103	$1,672	-$5,159
5	$6,665	$0	$310	$128	$836	$7,103	$1,659	-$5,029
6	$6,665	$0	$354	$145	$715	$7,164	$1,598	-$4,952
7	$6,665	$0	$363	$149	$587	$7,177	$1,586	-$4,843
8	$6,665	$0	$375	$154	$452	$7,195	$1,581	-$4,741
9	$6,665	$0	$391	$161	$310	$7,217	$1,554	-$4,648
10	$6,665	$0	$406	$167	$159	$7,239	$1,524	-$4,559
11	$0	$0	$419	$172	$0	$591	$1,493	-$364
12	$0	$0	$434	$179	$0	$613	$1,557	-$361
13	$0	$0	$447	$184	$0	$630	$1,626	-$356
14	$0	$0	$462	$190	$0	$652	$1,690	-$352
15	$0	$0	$475	$195	$0	$670	$1,743	-$346
16	$0	$0	$487	$200	$0	$687	$1,811	-$340
17	$0	$0	$506	$208	$0	$714	$1,870	-$338
18	$0	$0	$521	$214	$0	$735	$1,952	-$333
19	$0	$0	$533	$219	$0	$753	$2,022	-$326
20	$0	$0	$552	$227	$0	$779	$2,081	-$323
21	$0	$0	$571	$235	$0	$806	$2,158	-$320
22	$0	$0	$589	$242	$0	$832	$2,236	-$316
23	$0	$0	$608	$250	$0	$858	$2,314	-$312
24	$0	$0	$627	$258	$0	$884	$2,392	-$307
25	$0	$0	$645	$265	$0	$911	$2,470	-$303
Total	$66,652	$2,400	$11,315	$4,655	$7,494	$82,623	$48,353	-$52,854

Table I.58: Annual present values for the net cash flow and overall present worth of the system for approach 2

End of Year	Annual Payment to Bank for Loan ($)	Rebate ($)	Annual Heating Cost ($)	Annual Cooling Cost ($)	Annual Tax Savings ($)	Annual Expenses ($)	Annual Savings ($)	Present Value ($)
0	$0	$1,500	$0	$0	$0	$0	$1,500	$1,500
1	$4,786	$0	$616	$128	$905	$5,529	$1,300	-$4,426
2	$4,786	$0	$616	$128	$835	$5,529	$1,102	-$4,299
3	$4,786	$0	$616	$128	$761	$5,529	$1,061	-$4,179
4	$4,786	$0	$616	$128	$683	$5,529	$1,098	-$4,064
5	$4,786	$0	$616	$128	$601	$5,529	$1,117	-$3,955
6	$4,786	$0	$616	$145	$514	$5,547	$1,133	-$3,865
7	$4,786	$0	$702	$149	$422	$5,637	$1,081	-$3,833
8	$4,786	$0	$721	$154	$325	$5,661	$1,108	-$3,752
9	$4,786	$0	$746	$161	$222	$5,692	$1,112	-$3,681
10	$4,786	$0	$776	$167	$114	$5,729	$1,109	-$3,616
11	$0	$0	$807	$172	$0	$979	$1,105	-$604
12	$0	$0	$832	$179	$0	$1,010	$1,159	-$596
13	$0	$0	$863	$184	$0	$1,046	$1,210	-$590
14	$0	$0	$887	$190	$0	$1,077	$1,264	-$582
15	$0	$0	$918	$195	$0	$1,113	$1,300	-$575
16	$0	$0	$943	$200	$0	$1,143	$1,355	-$565
17	$0	$0	$967	$208	$0	$1,175	$1,408	-$556
18	$0	$0	$1,004	$214	$0	$1,219	$1,469	-$552
19	$0	$0	$1,035	$219	$0	$1,255	$1,520	-$544
20	$0	$0	$1,060	$227	$0	$1,287	$1,573	-$534
21	$0	$0	$1,097	$235	$0	$1,332	$1,632	-$528
22	$0	$0	$1,134	$242	$0	$1,376	$1,692	-$523
23	$0	$0	$1,171	$250	$0	$1,421	$1,752	-$516
24	$0	$0	$1,208	$258	$0	$1,465	$1,811	-$510
25	$0	$0	$1,245	$265	$0	$1,510	$1,871	-$502
Total	$47,856	$1,500	$21,812	$4,655	$5,380	$74,324	$34,843	-$46,445

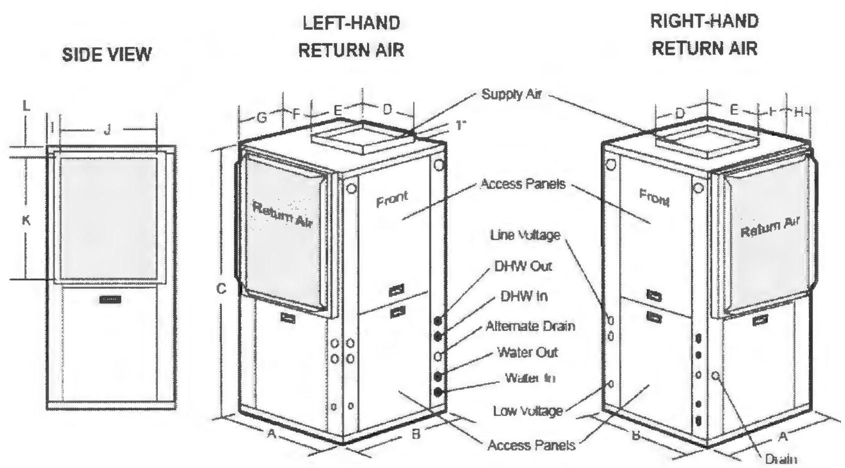
Table I.59: Annual present values for the net cash flow and overall present worth of the system for approach 3

End of Year	Annual Payment to Bank for Loan ($)	Rebate ($)	Annual Heating Cost ($)	Annual Cooling Cost ($)	Annual Tax Savings ($)	Annual Expenses ($)	Annual Savings ($)	Present Value ($)
0	$0	$1,500	$0	$0	$0	$0	$1,500	$1,500
1	$4,932	$0	$719	$128	$932	$5,779	$1,225	-$4,638
2	$4,932	$0	$663	$128	$860	$5,723	$1,081	-$4,453
3	$4,932	$0	$658	$128	$784	$5,718	$1,042	-$4,323
4	$4,932	$0	$678	$128	$704	$5,738	$1,057	-$4,222
5	$4,932	$0	$694	$128	$619	$5,754	$1,058	-$4,120
6	$4,932	$0	$714	$145	$529	$5,792	$1,051	-$4,041
7	$4,932	$0	$759	$149	$435	$5,840	$1,038	-$3,972
8	$4,932	$0	$790	$154	$335	$5,877	$1,048	-$3,897
9	$4,932	$0	$819	$161	$229	$5,912	$1,045	-$3,824
10	$4,932	$0	$850	$167	$118	$5,950	$1,039	-$3,755
11	$0	$0	$881	$172	$0	$1,053	$1,031	-$649
12	$0	$0	$915	$179	$0	$1,093	$1,076	-$645
13	$0	$0	$951	$184	$0	$1,135	$1,122	-$640
14	$0	$0	$985	$190	$0	$1,175	$1,167	-$634
15	$0	$0	$1,016	$195	$0	$1,211	$1,202	-$626
16	$0	$0	$1,049	$200	$0	$1,250	$1,248	-$618
17	$0	$0	$1,083	$208	$0	$1,291	$1,292	-$611
18	$0	$0	$1,126	$214	$0	$1,341	$1,347	-$607
19	$0	$0	$1,162	$219	$0	$1,382	$1,393	-$599
20	$0	$0	$1,196	$227	$0	$1,423	$1,436	-$590
21	$0	$0	$1,239	$235	$0	$1,474	$1,490	-$585
22	$0	$0	$1,283	$242	$0	$1,525	$1,543	-$579
23	$0	$0	$1,326	$250	$0	$1,576	$1,596	-$573
24	$0	$0	$1,369	$258	$0	$1,627	$1,650	-$566
25	$0	$0	$1,412	$265	$0	$1,678	$1,703	-$558
Total	$49,323	$1,500	$24,339	$4,655	$5,545	$78,317	$32,481	-$48,826

Table I.60: Annual present values for the net cash flow and overall present worth of the system for approach 4

End of Year	Annual Payment to Bank for Loan ($)	Rebate ($)	Annual Heating Cost ($)	Annual Cooling Cost ($)	Annual Tax Savings ($)	Annual Expenses ($)	Annual Savings ($)	Present Value ($)
0	$0	$0	$0	$0	$0	$0	$0	$0
1	$3,159	$0	$1,586	$151	$597	$4,896	$0	-$4,113
2	$3,159	$0	$1,411	$151	$551	$4,721	$0	-$3,819
3	$3,159	$0	$1,396	$151	$502	$4,705	$0	-$3,683
4	$3,159	$0	$1,459	$151	$451	$4,769	$0	-$3,621
5	$3,159	$0	$1,507	$151	$396	$4,816	$0	-$3,547
6	$3,159	$0	$1,570	$151	$339	$4,880	$0	-$3,487
7	$3,159	$0	$1,618	$172	$278	$4,949	$0	-$3,432
8	$3,159	$0	$1,697	$176	$214	$5,032	$0	-$3,388
9	$3,159	$0	$1,760	$182	$147	$5,102	$0	-$3,334
10	$3,159	$0	$1,824	$190	$75	$5,173	$0	-$3,282
11	$0	$0	$1,887	$197	$0	$2,084	$0	-$1,284
12	$0	$0	$1,966	$203	$0	$2,170	$0	-$1,279
13	$0	$0	$2,046	$211	$0	$2,257	$0	-$1,273
14	$0	$0	$2,125	$217	$0	$2,342	$0	-$1,265
15	$0	$0	$2,188	$224	$0	$2,413	$0	-$1,247
16	$0	$0	$2,268	$230	$0	$2,498	$0	-$1,235
17	$0	$0	$2,347	$236	$0	$2,583	$0	-$1,222
18	$0	$0	$2,442	$245	$0	$2,688	$0	-$1,217
19	$0	$0	$2,521	$253	$0	$2,775	$0	-$1,202
20	$0	$0	$2,601	$259	$0	$2,860	$0	-$1,186
21	$0	$0	$2,696	$268	$0	$2,964	$0	-$1,176
22	$0	$0	$2,791	$277	$0	$3,068	$0	-$1,165
23	$0	$0	$2,886	$286	$0	$3,172	$0	-$1,153
24	$0	$0	$2,981	$295	$0	$3,277	$0	-$1,139
25	$0	$0	$3,077	$304	$0	$3,381	$0	-$1,125
Total	$31,593	$0	$52,650	$5,331	$3,552	$89,575	$0	-$53,876

APPENDIX J - Equipment Specifications for WaterFurnace Heat Pump



MODEL	WATER CONNECTION	AUX HEAT KNOCKOUTS	A	B	C	D	E	F	G	H	I	J	K	L
E036, E040, E047, E048, E058, E060, E066, E072	1.00" Swivel	1-3/8"	31.2	25.5	58.4	18.0	18.0	6.6	6.5	3.7	1.1	27.3	34.0	1.7

Figure J.6: Vertical configuration dimensional data
Source: WaterFurnace, 2004d

Table J.61: E060 heating capacity data on high capacity

EWT	GPM	PSI	CFM	Heating Only				
				HC	KW	HE	LAT	COP
30	8	2.1	1000	29.2	2.99	19	97	2.86
			1700	34.1	3.09	23.5	88.6	3.23
			2000	34.9	3.25	23.8	86.2	3.15
	11	3.1	1000	31.1	2.98	20.9	98.8	3.05
			1700	36.5	3.12	25.9	89.9	3.43
			2000	37.5	3.32	26.2	87.4	3.31
	14	4.8	1000	31.1	2.91	21.1	98.8	3.12
			1700	37	3.2	26.1	90.2	3.4
			2000	37.9	3.35	26.5	87.5	3.31
50	8	2	1000	44.4	3.43	32.7	111.1	3.8
			1700	51.7	3.58	39.4	98.1	4.23
			2000	52.8	3.69	40.2	94.4	4.19
	11	2.9	1000	47.1	3.51	35.1	113.6	3.93
			1700	54.7	3.65	42.2	99.8	4.39
			2000	55.8	3.74	43	95.8	4.37
	14	4.6	1000	48	3.56	35.9	114.5	3.95
			1700	55.6	3.69	43	100.3	4.42
			2000	56.8	3.78	43.9	96.3	4.4
70	8	2	1000	60.3	4.21	45.9	125.8	4.2
			1700	68.2	4.2	53.9	107.2	4.76
			2000	69.7	4.22	55.3	102.3	4.84
	11	2.8	1000	62.7	4.3	48	128.1	4.27
			1700	70.7	4.26	56.1	108.5	4.86
			2000	72.3	4.27	57.7	103.5	4.96
	14	4.4	1000	64.5	4.37	49.6	129.8	4.33
			1700	72.5	4.31	57.8	109.5	4.93
			2000	74.2	4.31	59.5	104.4	5.04
90	8	1.9	1000	75.8	4.94	58.9	140.1	4.49
			1700	83.4	4.72	67.3	115.4	5.18
			2000	85.4	4.65	69.6	109.6	5.38
	11	2.7	1000	77.3	5.04	60.1	141.6	4.49
			1700	84.5	4.77	68.2	116	5.19
			2000	86.6	4.67	70.7	110.1	5.43
	14	4.2	1000	80.2	5.13	62.7	144.3	4.58
			1700	87.4	4.83	70.9	117.6	5.3
			2000	89.7	4.72	73.6	111.5	5.57

Source: WaterFurnace, 2004a

Table J.62: E060 heating capacity data on low capacity

EWT	GPM	PSI	CFM	Heating Only				
				HC	KW	HE	LAT	COP
30	5	1.1	700	16.2	1.5	11	91.4	3.16
			900	16.9	1.52	11.7	87.4	3.27
			1100	17.5	1.54	12.3	84.7	3.34
	8	2.1	700	16.1	1.53	10.8	91.2	3.08
			900	16.8	1.54	11.5	87.3	3.18
			1100	17.4	1.56	12	84.6	3.26
	11	3.1	700	16.7	1.53	11.5	92.1	3.2
			900	17.4	1.54	12.2	87.9	3.31
			1100	18	1.56	12.7	85.2	3.39
50	5	1	700	24.7	1.63	19.2	102.7	4.45
			900	25.9	1.62	20.4	96.6	4.67
			1100	26.6	1.62	21.1	92.4	4.81
	8	2	700	25.1	1.66	19.4	103.2	4.43
			900	26.3	1.65	20.6	97	4.67
			1100	26.9	1.64	21.3	92.6	4.8
	11	2.9	700	25.9	1.67	20.2	104.2	4.54
			900	27.1	1.66	21.4	97.9	4.79
			1100	27.7	1.65	22.1	93.3	4.92
70	5	0.9	700	33.6	1.82	27.4	114.4	5.42
			900	35.2	1.78	29.2	106.3	5.79
			1100	35.8	1.75	29.8	100.1	5.98
	8	2	700	34.9	1.85	28.6	116.2	5.54
			900	36.7	1.81	30.5	107.7	5.95
			1100	37.1	1.77	31.1	101.2	6.13
	11	2.8	700	35.6	1.87	29.3	117.1	5.58
			900	37.4	1.83	31.2	108.5	6
			1100	37.8	1.79	31.7	101.8	6.19
90	5	0.9	700	41	1.97	34.3	124.3	6.1
			900	43.1	1.9	36.6	114.3	6.63
			1100	43.2	1.85	36.9	106.4	6.85
	8	1.9	700	43.8	2	37	128	6.42
			900	46	1.93	39.5	117.4	7
			1100	46	1.86	39.7	108.7	7.24
	11	2.7	700	44.1	2.04	37.2	128.4	6.34
			900	46.3	1.96	39.7	117.7	6.93
			1100	46.2	1.89	39.8	108.9	7.17

Source: WaterFurnace, 2004a

Table J.63: E060 cooling capacity data on high capacity

EWT	GPM	PSI	FT	CFM	Cooling Only				
					TC	SC	KW	HR	EER
50	8	2	4.7	1000	61	34.1	2.73	70.3	22.4
				1700	68.8	43.6	2.98	78.9	23.1
				2000	70.2	47.4	3.17	81	22.1
	11	2.9	6.7	1000	61.7	34.5	2.67	70.8	23.1
				1700	69.5	44	2.92	79.4	23.8
				2000	70.9	47.9	3.1	81.5	22.8
	14	4.6	10.6	1000	62.3	34.8	2.62	71.2	23.7
				1700	70.2	44.5	2.87	79.9	24.5
				2000	71.6	48.3	3.05	82	23.5
70	8	2	4.5	1000	54.8	32.5	3.26	65.9	16.8
				1700	61.2	41.6	3.56	73.3	17.2
				2000	63.1	45.2	3.79	76.1	16.6
	11	2.8	6.4	1000	55.4	32.8	3.19	66.3	17.4
				1700	61.8	42	3.49	73.7	17.7
				2000	63.8	45.6	3.71	76.4	17.2
	14	4.4	10.1	1000	55.9	33.1	3.13	66.6	17.9
				1700	62.4	42.4	3.43	74.1	18.2
				2000	64.4	46.1	3.65	76.9	17.6
90	8	1.9	4.3	1000	47.1	29.9	3.77	60	12.5
				1700	51.9	38.3	4.12	66	12.6
				2000	54.2	41.6	4.39	69.2	12.3
	11	2.7	6.1	1000	47.6	30.2	3.69	60.2	12.9
				1700	52.5	38.6	4.04	66.3	13
				2000	54.7	42	4.3	69.4	12.7
	14	4.2	9.6	1000	47.9	30.4	3.63	60.3	13.2
				1700	53	39.1	3.97	66.6	13.4
				2000	55.3	42.4	4.23	69.7	13.1
110	8	1.8	4.1	1000	38.6	30.6	4.2	52.9	9.2
				1700	42.3	39.4	4.6	58	9.2
				2000	44.6	42.8	4.9	61.4	9.1
	11	2.6	5.9	1000	39	30.9	4.11	53	9.5
				1700	42.8	39.8	4.5	58.1	9.5
				2000	45.1	43.2	4.8	61.5	9.4
	14	3.9	9	1000	39.4	31.2	4.04	53.2	9.8
				1700	43.2	40.2	4.42	58.3	9.8
				2000	45.5	43.7	4.72	61.7	9.6

Source: WaterFurnace, 2004a

Table J.64: E060 cooling capacity data on low capacity

EWT	GPM	PSI	FT	CFM	Cooling Only				
					TC	SC	KW	HR	EER
50	5	1	2.3	700	35.3	20.6	1.07	38.9	32.8
				900	37.6	23.6	1.1	41.3	34.2
				1100	38.7	26.8	1.16	42.7	33.5
	8	2	4.6	700	35.3	20.8	1.03	38.9	34.2
				900	37.7	23.8	1.06	41.3	35.6
				1100	38.8	27	1.11	42.6	34.9
	11	2.9	6.7	700	35.4	20.8	1.01	38.9	35
				900	37.8	23.8	1.04	41.3	36.5
				1100	38.9	27	1.09	42.7	35.7
70	5	0.9	2.2	700	30.8	19.9	1.41	35.6	21.9
				900	32.3	22.3	1.43	37.2	22.6
				1100	33.3	25	1.49	38.4	22.4
	8	2	4.6	700	31	20.1	1.36	35.6	22.7
				900	32.5	22.5	1.39	37.2	23.4
				1100	33.5	25.3	1.45	38.4	23.2
	11	2.8	6.4	700	31.2	20.2	1.34	35.7	23.3
				900	32.7	22.6	1.36	37.3	24
				1100	33.7	25.4	1.42	38.5	23.7
90	5	0.9	2.1	700	26.2	19	1.74	32.1	15.1
				900	27	20.8	1.76	33	15.3
				1100	27.8	23.1	1.81	34	15.3
	8	1.9	4.4	700	26.4	19.2	1.7	32.2	15.5
				900	27.3	21	1.72	33.2	15.8
				1100	28.1	23.4	1.78	34.2	15.8
	11	2.7	6.1	700	26.7	19.4	1.68	32.4	15.9
				900	27.5	21.2	1.7	33.3	16.2
				1100	28.4	23.6	1.75	34.4	16.2
110	5	0.9	2	700	21.7	17.8	2.02	28.6	10.7
				900	22	19.2	2.04	29	10.8
				1100	22.7	21.1	2.08	29.8	10.9
	8	1.8	4.2	700	22	18	2	28.8	11
				900	22.3	19.4	2.02	29.2	11.1
				1100	23	21.3	2.06	30	11.2
	11	2.6	5.9	700	22.3	18.3	1.97	29	11.3
				900	22.6	19.7	1.99	29.4	11.4
				1100	23.3	21.6	2.03	30.3	11.5

Source: WaterFurnace, 2004a

APPENDIX K - Equipment Specifications for Electric Boiler

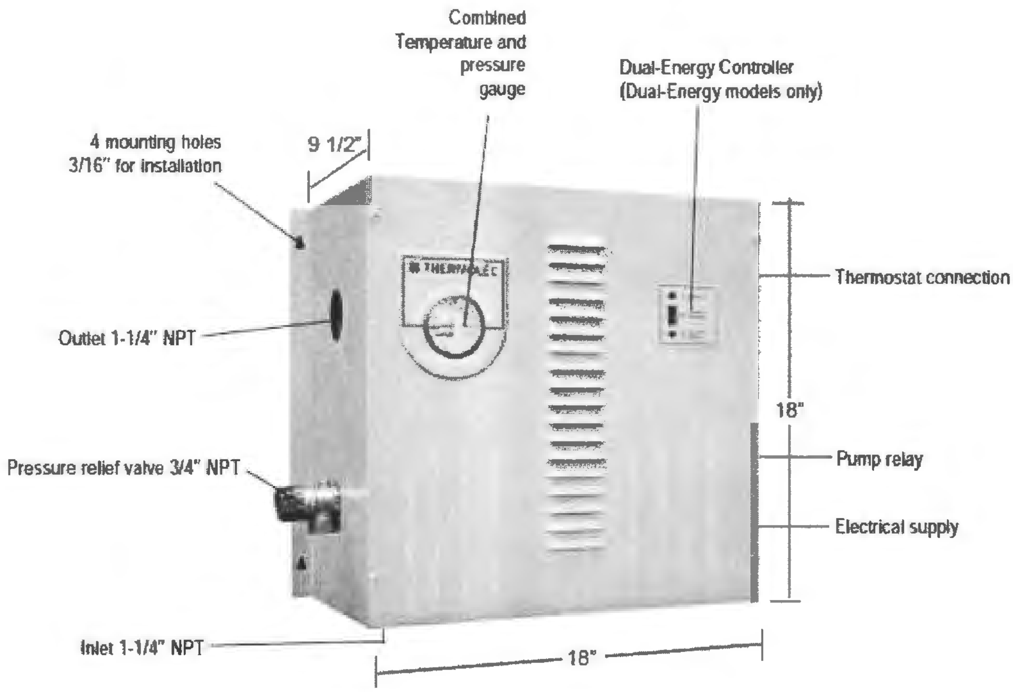


Figure K.7: Physical dimensions of electric boiler
Source: Thermolec, 2004

APPENDIX L - Degree-Day Method

The degree-day method was performed to estimate the annual energy use for the case study home. The degree-day method is the simplest method evaluated in this study. Generally, this method is only recommended to estimate energy demands of a single-family residential home. However, studies have shown that the degree-day analysis may result in inaccurate estimations and caution a practitioner from using this method. Nevertheless, the method will be demonstrated to illustrate the differences in results to the bin and improved methods for annual energy use.

The degree-day method is based on heating degree days and cooling degree days. The number of degree days for a particular location is the measure of departure from a given standard (usually 65°F), one degree day for each degree of departure either above or below the standard during one day. To calculate the number of degree days that could be seen at the case study home site, hourly outdoor air temperatures were used from the TMY2 data for Des Moines, Iowa. The number of annual heating degree days was found by

$$HDD = \sum_{i=1}^{8,760} \left[T_{Std.} - T_{outdoor,,i}\right]^{+} \quad \text{(L.1)}$$

The number of annual cooling degree days was found by

$$CDD = \sum_{i=1}^{8,760} \left[T_{outdoor,,i} - T_{std.}\right]^{+} \quad \text{(L.2)}$$

where the + sign denotes that only positive values are to be summed for the calculation. The numbers of heating and cooling degree days for Des Moines, Iowa were found to be 6654 and 1076.

The annual energy use of the case study home for each the heating and cooling mode can be estimated using degree days by

$$Annual\ Energy\ Demand = \frac{Q_{design}(DD)*24}{\Delta T_{design}} \qquad (L.3)$$

The design heating and cooling loads estimated in Chapter 2 were 73.91 and 53.4 MBtu's per hour respectively. These design loads were determined using a delta T for heating and cooling used in Chapter 2 was assumed to be 77 °F (68 °F indoors and -9 °F outdoors) and 20 °F (75 °F indoors and 95 °F outdoors). The annual heating and cooling energy demand for the case study home was then estimated to be 153.27 and 68.95 MBtu's per year. A summary of the inputs and results obtained using the degree-day analysis can be seen in Table L.1.

Table L.65 Summary of degree day method inputs and results

Mode	Design Load (MBtuh)	Degree Days	ΔT (F)	Estimated Annual Energy Demand (MMBtuh)
Heating	73.9	6654	77	153.27
Cooling	53.4	1076	20	68.95

APPENDIX M - Bin Method

This section illustrates using the bin method for estimating annual energy use of the case study home. For a bin analysis, multiple ranges of outdoor air temperatures are separated into bins corresponding to the number of hours at which those ranges of temperatures occur annually at a particular geographical location. Bins having the number of hours per year that a range of temperatures occur were determined using TMY2 weather data. The temperature band for each bin was chosen to be 5°F, beginning at -20°F and ending at 120 °F.

To perform a bin analysis the amount of heating and cooling energy required to maintain the home at a specified indoor air temperature as a function of outdoor air temperature was estimated. To estimate this amount of energy, design heating and cooling loads were used. The design heating and cooling loads were determined to be 73.9 and 53.4 MBtuh. To estimate the building load as a function of outdoor temperature a linear correlation was used, i.e., it was assumed that the building load varies linearly with outdoor temperature.

The linear profile of the building heating and cooling loads were determined using two points, the design load and the corresponding balance point temperature. The balance point temperature is defined as the outdoor air temperature at which no heating or cooling of the home is required as a result of the internal gains of the home. Thus, the balance point temperature of a building in the heating condition is defined as the temperature of outdoor air where for a specified value of the interior temperature, the total heat loss is equal to the total interior heat gain. Conversely, the balance point temperature of a building in the cooling condition is the temperature of outdoor air where for a specified value of the interior temperature, the interior heat gain is equal to the energy required to raise the temperature of the space to the specified value of interior temperature. For the analysis, the heating and cooling balance point temperatures were assumed to be 55°F and 70°F respectively.

To estimate the balance point temperature, recommendations from the International Ground Source Heat Pump Association (IGSHPA) were used. IGSHPA recommends estimated heating balance point temperatures as a function of type of construction of the home. The three types of construction are categorized as average, energy efficient and super insulated. These construction types correspond to recommended balance point temperatures of 60°F for an average home, 55°F for an energy efficient home and 50°F for a super insulated home. Similarly, the IGSHPA recommends a balance point temperature for the cooling mode; however this value is not corresponded to a construction type. The recommended balance point temperature in the cooling mode for all residential structures is 70°F.

As recommended the heating balance point temperature was assumed to be 55°F and 70°F degrees Fahrenheit. The linear correlations of the load profiles were plotted and lines were fit to obtain the linear function in equation form.

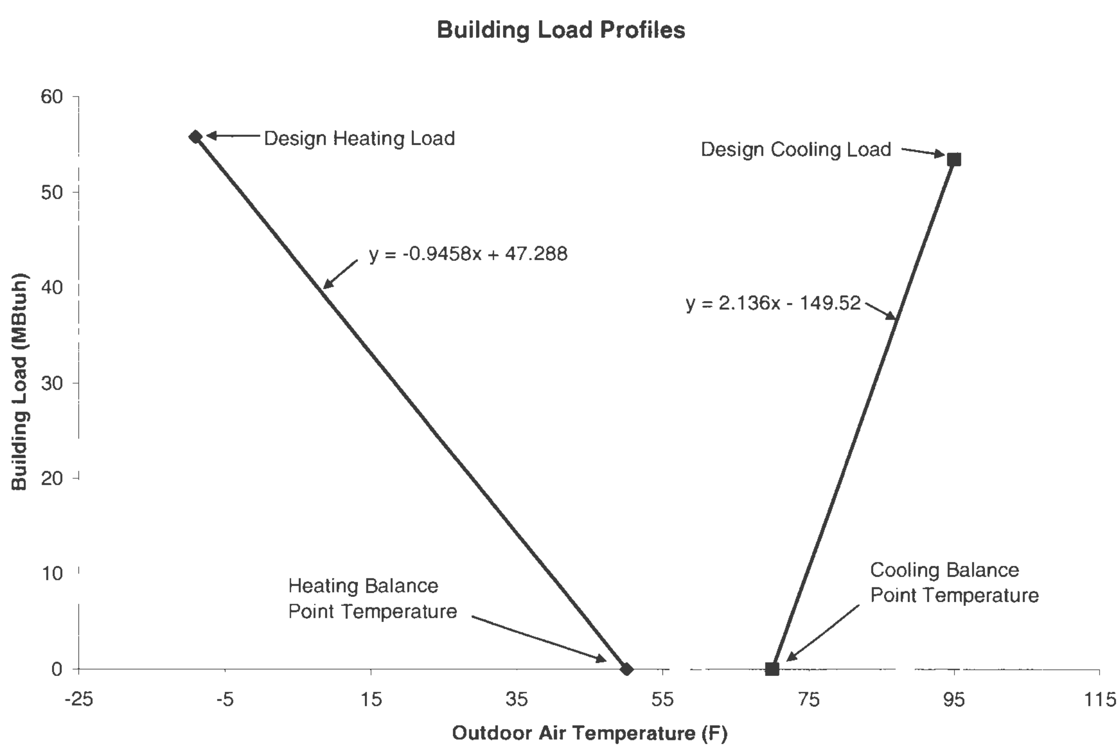


Figure M.8: Building load profile as a function of outdoor air temperature

The total heating and cooling energy use for each bin is the product of the hours in each bin and the respective building load. Thus, the estimated annual energy use is the sum of the energy use for all bins.

Table M.66: Bin method Results

Range (F)	Hours at Range	EWT (F)	Building Load (Btuh)	Total Heating Energy (MBtu)	Total Cooling Energy (MBtu)
-25 / -20	0	34	69581	0	0
-20 / -15	10	35	64782	648	0
-15 / -10	18	36	59984	1,080	0
-10 / -5	51	38	55185	2,814	0
-5 / 0	138	39	50386	6,953	0
0 / 5	101	40	45588	4,604	0
5 / 10	129	41	40789	5,262	0
10 / 15	191	43	35990	6,874	0
15 / 20	291	44	31192	9,077	0
20 / 25	301	45	26393	7,944	0
25 / 30	411	46	21594	8,875	0
30 / 35	818	47	16795	13,739	0
35 / 40	671	49	11997	8,050	0
40 / 45	478	50	7198	3,441	0
45 / 50	451	51	2399	1,082	0
50 / 55	582	53	0	0	0
55 / 60	694	55	0	0	0
60 / 65	931	58	0	0	0
65 / 70	779	60	0	0	0
70 / 75	556	63	6675	0	3,711
75 / 80	483	65	20025	0	9,672
80 / 85	398	68	33375	0	13,283
85 / 90	178	70	46725	0	8,317
90 / 95	82	72	60075	0	4,926
95 / 100	18	75	73425	0	1,322
100 / 105	0	77	86775	0	0
Annual Energy Use (MMBtu)				80.44	41.23

The resulting annual heating and cooling energy use of the home was 80.44 and 41.23 MMBtu.