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Original article

HVAC System Design by Using Geothermal Heat Pump

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Abstract
A design of a geothermal heat pump heating, ventilating, and air conditioning system has been done for XYZ Indianapolis School by using renewable energy. Geothermal energy is one of the important energies that has proved itself in air conditioning systems all over the world. The design was classified in three points as follows. Using borehole heat exchanger (BHEs) vertical type, heat pump and all its equipment such as mechanical room, pipes, and fitting design to cover the total load of the school the results were 20 boreholes and a total flow rate of 60 gpm (3 gpm/bore) of 20 % propylene solution, and spacing 20 ft with 1.1 Hp and 6.5 in impeller size. For individual borehole design, the results were 15 boreholes and a total power consumption 1.1 kW. For the ventilation design, the peak load for heating is 42779 Btu/hr, and for cooling, it is 47779 Btu/hr. There are 4 boreholes, a 0.5 hp electric pump with a flow rate of 10 gpm a 40ft of H2O pressure drop, and a 10 hp fan integrated with a water-to-water heat pump. The total load we assumed for the heating supply in class 6 and the corridor are 10% and 25 %, respectively. The thermal resistance was calculated for each case according to the fluid flow rate. The ventilation load of the school is designed separately from the heating and cooling loads and connected to a water-water heat pump that will make the system work with high efficiency. Finally, we designed this project using Geothermal Energy, which is cleaner than conventional energy.
Keywords. HVAC System, Geothermal Heat Pump, Borehole Heat Exchanger, Sustainable Energy Solutions.

Introduction

Geothermal energy systems have been widely studied and applied in heating, ventilation, and air conditioning (HVAC) systems due to their sustainability and efficiency. One significant work is by Curtis et al. (2005), which evaluated the performance of geothermal heat pump systems in commercial buildings. The study highlighted notable energy savings, reduced greenhouse gas emissions, and improved reliability compared to conventional HVAC systems. The findings underscored the potential of geothermal systems to provide efficient and eco-friendly alternatives in various settings. Educational institutions have been a focal point for geothermal applications. Sanner et al. (2005) conducted a case study on geothermal heating and cooling systems installed in an elementary school. Using vertical borehole heat exchangers, the project demonstrated substantial cost reductions and increased energy efficiency. This case study emphasized the practicality of geothermal energy in reducing operational costs and providing sustainable solutions for large facilities like schools. Borehole Thermal Energy Storage (BTES) has also been extensively explored for geothermal applications.

Eskilson (1987) investigated the optimization of borehole heat exchanger arrays, focusing on parameters such as spacing, depth, and thermal resistance. The study provided detailed insights into enhancing the performance of geothermal systems by optimizing borehole configurations, making it a valuable reference for large-scale applications requiring high efficiency. Hybrid systems integrating geothermal and solar energy have been proposed as a way to further enhance efficiency and sustainability. Lund and Freeston (2001) explored the synergy between these two renewable energy sources in HVAC systems for schools and office buildings. Their findings revealed that combining geothermal and solar energy significantly improved overall system efficiency while reducing environmental impact. This hybrid approach offers a promising avenue for achieving more sustainable energy systems. Economic feasibility is a critical consideration in adopting geothermal HVAC systems. Omer and Evans (2008) analyzed the cost-effectiveness of ground-source heat pump systems in institutional buildings. Despite higher initial installation costs and ground area requirements, the study found that geothermal systems provided significant long-term savings in energy and maintenance costs. The research highlighted geothermal energy as a viable and affordable renewable energy option for large-scale applications.

Finally, tools like RETScreen have been instrumental in evaluating the feasibility of geothermal energy systems. The Government of Canada's RETScreen software has been used to analyze the economic and environmental benefits of geothermal HVAC systems in schools and commercial buildings. By providing detailed performance metrics and cost-benefit analyses, RETScreen supports informed decision-making for adopting geothermal systems. Together, these studies and tools demonstrate the potential of geothermal energy as a sustainable and efficient alternative to conventional HVAC systems. Renewable Designing an HVAC for a school is the objective of this paper to provide cooling during the summer by upgrading its

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existing system. The school was built in the 1950s and is in Indianapolis but has no cooling options since it heats with a central boiler system. Baseboard convectors and unit ventilators are used for heating because the classrooms have no ductwork. The rooms located in the East Wing will be used for kindergarten prep and other pre-school activities for grades 1-3 [12]. The design for the geothermal system will include water-to-air heat pumps to handle heating and cooling loads in the classes, water-to-water heat pumps to handle ventilation loads, a central ground heat exchanger, air handling units, and ductwork.[13]

Methodology

In this paper, a design of a geothermal heat pump heating, ventilating, and air conditioning system will plan it for XYZ Indianapolis School by using renewable energy. The data provided includes classroom and ventilation loads, and the property map of the school, which was supplied by Dr. Chiasson [11,12]. Software was used to design vertical ground heat exchanges for both configurations during this study [11,12,13]. Typical occupancy of the school buildings in Indianapolis, IN is 380-560 hours, and according to ASHRAE the Equivalent Full-Load Hours for heating and cooling hours are 480-400. Figure (1) below describes all assumptions that we have achieved from this study to choose the optimum values of design parameters of (GHX). Using all software, to design vertical ground heat exchange for both configurations. By using NASA/RETScreen satellite data (average underground earth temperature is 56.7°F. To optimize the performance of the Ground Heat Exchanger (GHX), we applied suitable design to select the most suitable design parameters. All assumptions made during the design process were intended to maximize thermal efficiency and avoid potential economic losses resulting from inefficient operation or mismatched GHX capacity. all assumptions of the design are shown in Figure (1) below [14,15].

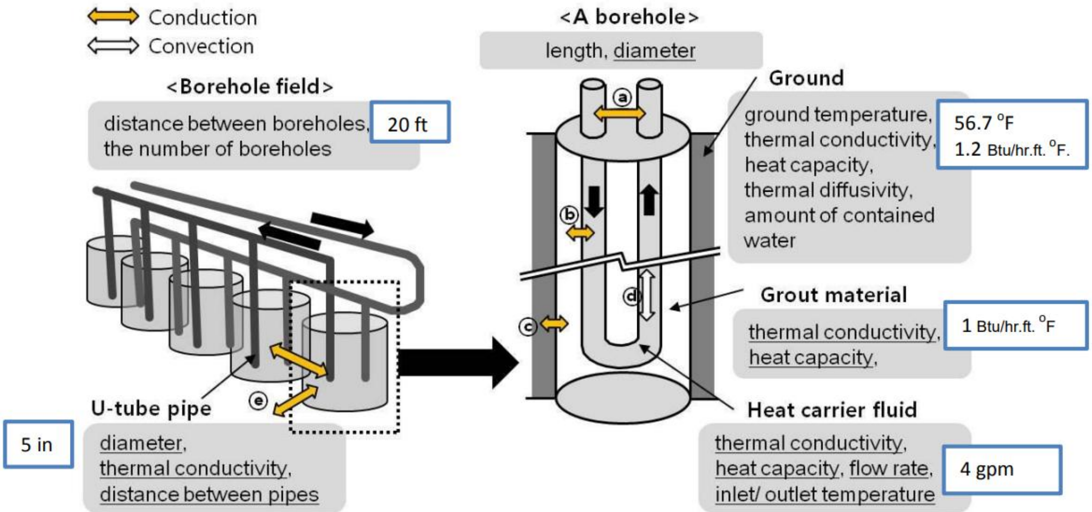

Figure 1. Borehole heat exchanger (BHEs) vertical type.

Mathematical Equations and Methods

The Borehole Thermal Resistance -char method

The borehole thermal resistance of ground heat exchangers with grouted u-tubes consists of two subparts: (i) the pipe resistance (ii) the grout resistance, as shown in Figure (1). The pipe resistance itself consists of to two subparts: (i) Convective resistance due to internal flow (ii) Conductive resistance through the pipe wall, The so-called "grout resistance" is a geometry problem comprised of conduction heat transfer from two (or more) cylindrical heat sources in a circular region (the borehole) of assumed homogeneous properties Hellström (1991) provides simplified solutions for multiple u-tubes where the u-pipes are approximated as line sources in a circular region[13,14].

$$R_b' = \frac{1}{4\pi D_{P,in} h_{in}} + \frac{ln\left[{D_{P,out}}/{D_{P,in}}\right]}{8\pi k_P} \quad (1)$$

$$R'_{P,DoubleU-tube} = \frac{1}{2\pi k_{grout}} \begin{bmatrix} ln\frac{r_b}{r_{P,out}} - \frac{3}{4} + b^2 - \frac{1}{4}ln(1-b^8) \\ -\frac{1}{2}\ln\left(\frac{\sqrt{2}br_b}{r_{p,out}}\right) - \frac{1}{4}ln\left(\frac{2br_b}{r_{p,out}}\right) \end{bmatrix} + \frac{R_P'}{4} \quad (2)$$

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where: – r_p is the borehole radius; $r_{p,\ out}$ is the pipe outer radius; k_{grout} is the thermal conductivity of the grout; b is an eccentricity parameter equivalent to the u-tube shank spacing (distance between pipe edges), S, divided by the borehole diameter; R'p is the pipe thermal resistance given by Equation 13; and subscripts b and p refer to borehole and pipe, respectively [10,11].

We'll use one by Gnielinski (1976) for $0.5 < p_r < 2000$ and $3000 < R_e < 5 \times 10^6$

$$N_u = \frac{(f/8)(R_e - 1000)P_r}{1 + 12.7(f/8)^{\frac{1}{2}}(\ P_r^{\frac{2}{3}} - 1)} \quad (3)$$

Where f is the Moody (or Darcy) friction factor for smooth pipes after Petukhov (1976) for a large range of Reynolds Numbers (3000 < Re < 5x10[6]) [12].

$$f = (0.79 ln R_e - 1.64)^{-2} \quad (4)$$

Now, we can calculate h from Equation 14 and the pipe thermal resistance (R'p) from Equation 13, with knowledge of pipe and fluid properties [13].

The Design Length Equation:
Early versions of the design length equation (e.g. International Ground Source Heat Pump Association (IGSHPA), 1988) did not include a temperature penalty to account for long-term temperature changes around boreholes. The IGSHPA (1988) design equation is probably suitable for residential systems but is not advisable for situations with imbalanced annual loads. The current version of RET Screen uses the IGSHPA (1988) equation. The IGSHPA (1988) equation has been updated to account for imbalanced annual loads (IGSHPA, 2009), but still survives in older design programs. On top of that, the ASHRAE Handbook has pre-calculated temperature penalties that are now questioned (Bernier (2008) [15].

Now, let's look at each of the terms in the design length equation

$$L = \frac{\dot{q_h} R'_b + \dot{q_a} R'_a + \dot{q_m} R'_m + \dot{q_h} R'_h}{(T_f - (T_g + T_P))} \quad (5)$$

where q is the ground load, R' is the effective thermal resistances per unit length, T_f is the average fluid temperature, T_g is the undisturbed ground temperature, and T_p is a temperature penalty to account for the thermal interaction of boreholes. Subscripts b, a, m, and h refers to borehole, annual, monthly, and hourly. Note that positive values of q are associated with heat rejection to the ground (so, cooling loads are positive) as shown in Figure (2) Dayton, OH [11,13].

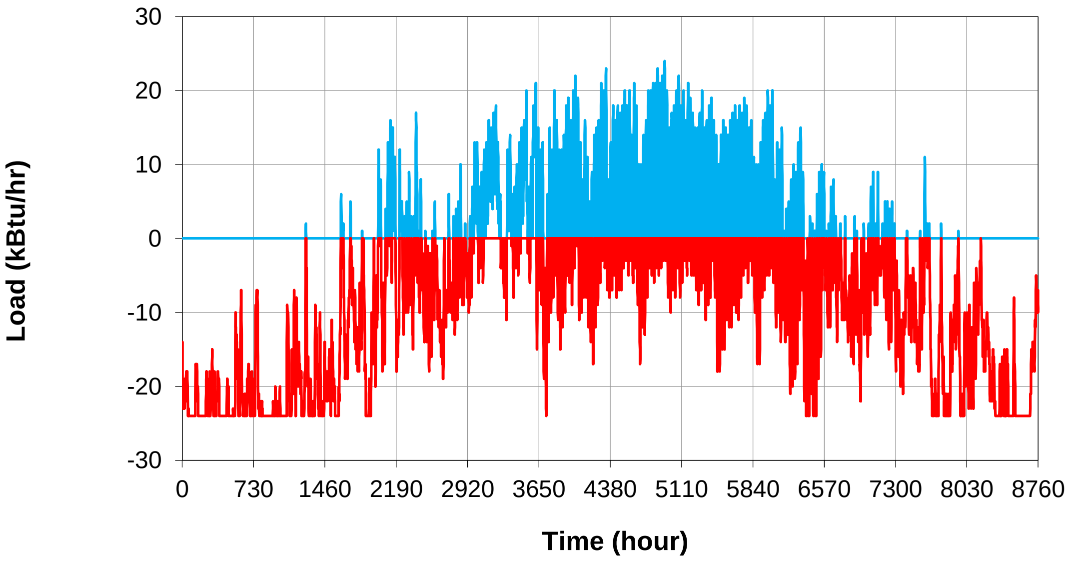

Figure 2. Heating and cooling loads for Minneapolis residence.

Therefore, in GHX design, the main goal is to calculate the total GHX length to, supply fluid temperatures to heat pump(s) within design limits (i.e., meet the building's heating and cooling loads). Ensure these conditions will be met over the design life cycle. This is a complicated function of the thermal storage properties of the ground. The latter point is new for conventional HVAC system designers because HVAC

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design is usually only concerned with peak-hour loads (life cycle simulations may be conducted to aid in system selection on an energy consumption basis, but not for sizing the system). Determining an adequate GHX length is the most important part of the overall design process but probably the least understood [10,12,15].

Ground Temperature

There are a few methods for estimating the undisturbed ground temperature for vertical closed-loop GHX systems (i) Field measurement during in-situ thermal conductivity tests (ii) From maps (iii) Surface meteorological data (a) Correlation by Signourelli and Kohl (2006) (b) Kasuda (1965) (calculates Tg as a function of depth which is necessary for horizontal GHX systems), Finally by NASA/RETScreen satellite which was used in this paper, and from Signorelli and Kohl equation[10].

$$T_g = T_{air} + 1.4°C \qquad (6)$$

3.4 **The Average fluid temperature**:

The average fluid temperature can compute from,

$$T_f = T_{out,GHX} + \frac{\dot{q}}{2\dot{m}c_P} = T_{in,HP} + \frac{\dot{q}}{2\dot{m}c_P} \qquad (7)$$

3.5 The Temperature Penalty:

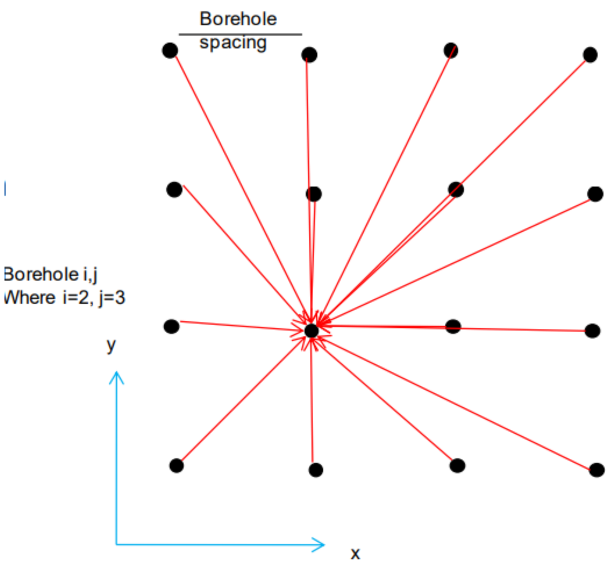

Figure 3. Temperature influence from all other bores.

The goal here is to calculate the temperature response in each borehole due to all the others. This is an effective temperature penalty to account for thermally interacting boreholes. We'll use a method based on superposition of Line Sources, so the temperature change at any particular bore at grid position x,y (DTb x,y) is the sum of the temperature influence from all other bores[11]:

$$T_P = \frac{\sum \Delta T_{b\ i,j}}{x\ bores \times y\ bores} \qquad (8)$$

A finite line source solution by Zeng, Diao, and Fang (2002) can be used to calculate $\Delta T_{b\ i,j}$.

$$\Delta T_{b\ i,j} = \frac{\dot{q}'}{2\pi k} ln \frac{H}{2.2r} \qquad (9)$$

where H is the borehole depth, and r is the radial distance from the line source. A time where end effects become important has been determined numerically by Eskilson (1987):

$$t_{End,Effect} = \frac{H^2}{9\propto} \qquad (10)$$

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Ground Thermal Properties
In general, thermal conductivity of the ground (k) is proportional to density and water content as it available Kavanaugh and Rafferty (1997) [14].

Ground-Source Heat Pump Operation
Heat pumps are devices for converting low-temperature thermal energy into higher-temperature thermal energy. The low-temperature heat is gathered from some material called the "source," and then concentrated and released into another material called the "sink". When used to heat buildings, heat pumps can gather low-temperature heat from sources such as outdoor air, ground water, lakes or ponds, or tubing buried within the earth. All these sources provide "free" low-temperature heat. In comparison to air-source heat pumps, GHPs rely on earth temperatures, which are much closer to room conditions, so the heat pump source and sink temperatures are much closer (i.e., the "lift"), which translates to higher efficiencies [11].

Cooling Mode
A "water-source" heat pump is used in nearly all applications where low-temperature heat is to be extracted from a medium other than air. The fluid conveying the low-temperature heat to the heat pump can be water or a water-based antifreeze solution, and it can come from a variety of sources.
The most common type of water-source heat pump is the water-to-air heat pump. In heating mode, it uses a water-to-refrigerant heat exchanger as its evaporator. Heat output is delivered to a stream of air drawn through the condenser coil and forced through a duct system by the heat pump's blower.
Figure (4) to the right shows an example of a water-to-air heat pump configured as a vertical cabinet unit. Return air enters the upper left side of the cabinet, passes through a filter and then through the refrigerant-to-air heat exchanger. The conditioned air is then drawn through the blower and discharged vertically from the cabinet into a duct system. The compressor and other electrical or refrigeration system components are located in the lower portion of the cabinet [10,12].

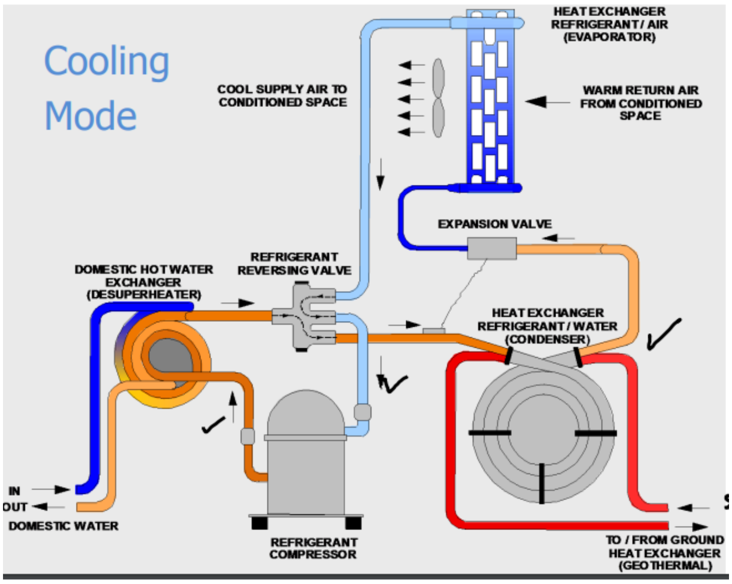

Figure 3.A water to air heat pump in cooling mode.

$$\dot{q}_{ground} = \dot{q}_{building}\left(\frac{COP_C+1}{COP_C}\right) \qquad (11)$$

Heating Mode
One of the most opportune situations for a heat pump is when low-temperature heat can be extracted from a stream of water and higher-temperature heat can be dissipated into another stream of water. Heat pumps configured in Figure (5) for such situations are called "water-to-water" heat pumps. They can be used in a wide variety of applications for building heating and cooling, as well as applications such as domestic water heating, pool heating or processes where either heated water, chilled water or both are required [12,13].

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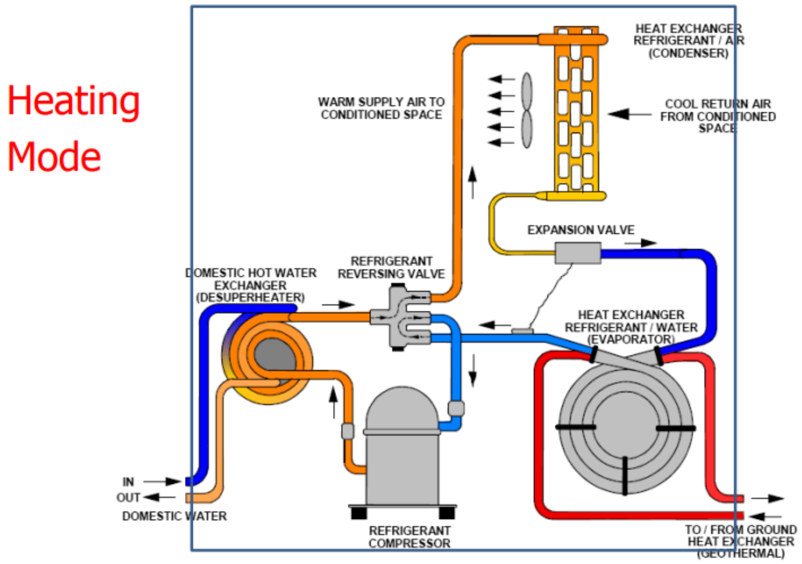

Figure 5. A water to air heat pump in heating mode.

$$\dot{q}_{ground} = \dot{q}_{building}\left(\frac{COP_h - 1}{COP_h}\right) \quad (12)$$

Pressure Drop Calculations

In closed-loop piping systems, head loss (or pressure drop) is solely due to friction loss (due to fluid flow). The Darcy-Weisbach Equation, which relates the head loss (pressure drop) due to friction (h_f) along a given length of pipe to the average velocity of the fluid flow [15].

$$\Delta P = f\frac{L}{D}\frac{V^2}{2g} \quad (13)$$

Where h_f is in ft or m of fluid, f is the Darcy friction factor, L is the pipe length, D is the internal pipe diameter, v is the average fluid velocity, and g is acceleration due to gravity [11].

Pump Sizing

The pump work (or capacity) in horsepower (hp) is given by:

$$\dot{W}_{pump} = \frac{\dot{Q}\,\Delta h}{396\,\eta_{pump}} \quad (14)$$

where W_{pump} is the pump work (or capacity) in horsepower, $\dot{Q}$ is the volumetric flow rate in gpm, Δh is the head loss in ft, and η_{pump} is the mechanical efficiency of the pump (~0.7). The motor electrical input power (in kW) is determined from,

$$\dot{W}_{motor} = \frac{0.746(kW/h_p)\dot{W}_{pump}}{\eta_{motor}} \quad (15)$$

where symbols are as described above η_{motor} is the electrical efficiency of the motor (standard efficiency ~0.8) [10].

Ductwork distribution system:

This design calls for the installation HVAC system to provide cooling during the summer. Due to the lack of a heat recovery unit, we need to install an additional water to water heat pump to handle the extra loads to maintain the heat pumps working efficiently. According to provided data of ventilation loads in this school we should install a water-to-water pump as shown on Figure (6) below, and it would be as a part of central ventilation system in both design configurations. The air-to-air recovery unit is to minimize heating and cooling energy losses [15].

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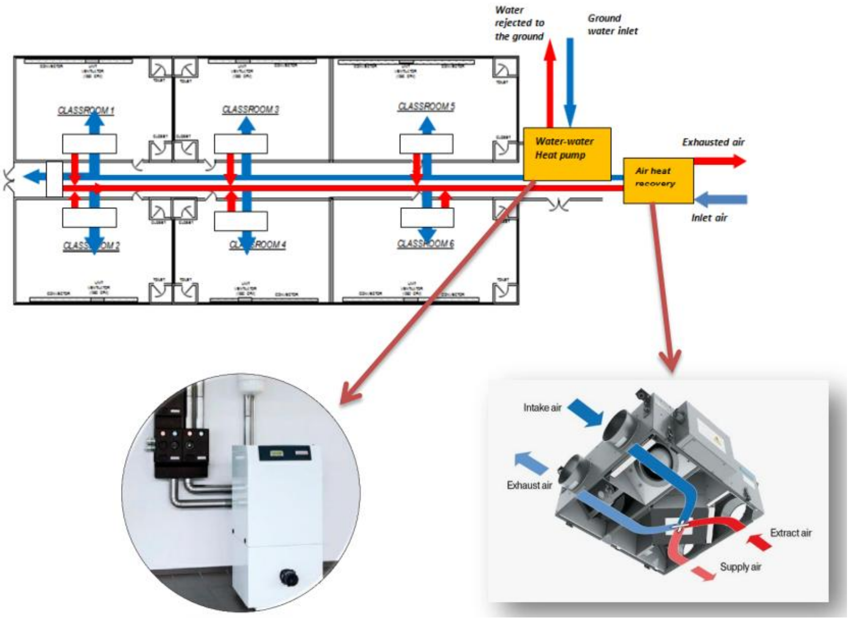

Figure 4. Air to Air recovery unit and water to water heat pump.

Building loads calculations

The total load is classified into cooling and heating loads as indicated in Table (1). The design of the loads was based on the maximum load over the year, which is in January and July (i.e, heating and cooling loads, respectively). Calculation of building loads to select the optimum ground heat pump size that can handle the load for each space. According to the given data, the load calculations are shown in the table below.

Table 1. Classroom loads.

Classroom loads	Class 1	Class 2	Class 3	Class 4	Class 5	Class 6	East Wing Corridor	Total / day
Total heating	604942.7	604346.8	582573.5	581976.9	707199.5	706325.7	748547.7	4535912.8
Total cooling	397852.9	399200.8	392581.9	393933.4	450369.8	451772.1	295168.2	2780879.1
Peak heating	30,728	30,703	29798	29772	35781	35,743	41,477	234,002
Peak cooling	30,985	26,062	30,805	25,882	36,012	29,947	16,902	196,595
Mon PLF h	0.820284423	0.8201624	0.8146231	0.8144914	0.8235323	0.8233823	0.7519779	
Mon PLF C	0.535010849	0.6382328	0.5310058	0.6341817	0.5210934	0.628578	0.7276504	

Heat Pump Selection

Selecting a pump for each class based on the space load calculation for both cooling and heating. In selecting the pump size that handles the cooling load for each class by using Bosch-fhp catalog heat pump data considering the cooling capacity, sensible to the total ratio (SHR) at design conditions which are entering water temperature (EWT) 85°F, 40°F in cooling and heating, respectively with entering air temperature (EAT) 70°F as shown on ASHRAE applications handbook [12,13,14].

Table 2. Heat pump selection.

Table. Pump size and percentage of supplement heating.				Cooling			Heating		
Class #	Peak cooling	Peak heating	Heat pump type	Total capacity (MBtuH)	EER COPc=EER/3.412	Sensible to total Ratio	Total capacity (MBtuH)	COPh	Sup %
1	30,985	30,728	EC041	32.48	10.9	0.68	32.46	3.3	0
2	26,062	30,703	EC041	32.48	10.9	0.68	32.46	3.3	0
3	30,805	29,798	EC041	32.48	10.9	0.68	32.46	3.3	0
4	25,882	29,772	EC036	30.59	11	0.68	31.24	3.3	0
5	36,012	35,781	EC042	36.42	11.2	0.69	36.32	3.4	0
6	29,947	35,743	EC041	32.48	10.9	0.68	32.46	3.3	9.2

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Corridor	16,902	41,477	EC036	30.59	11	0.69	31.24	3.3	24.6

$$\% \; Heat \; Supply = \frac{(peak \; heating \; load - total \; capacity \; heating)}{(peak \; heating \; load)} \times 100$$

Design of Central Vertical Ground Heat Exchanger

Borehole resistance calculation

The borehole diameter is 5 in, and the flow rate is varied for each class according to load calculation (6.2 - 8.6) gpm. Soil profile consists of inter-mixed sand and clay, k is 1.2 $(Btu/hr.ft.°F)_{T=40°F}$, and standard bentonite grout k is 1 $(Btu/hr.ft.°F)_{T=40°F}$. The working fluid is 20 % solution of propylene glycol because the minimum earth temperature for Indianapolis is -1.4 ℃ (28.76)°F . This met 20% propylene glycol whose freezing point is 19.2 °F. A (1-in) u-tube with tubes touching the borehole wall, center spacing$(\frac{D}{2} - \frac{d}{2})$. Center spacing is 1.8425 in. borehole diameter is 5 in, and borehole depth is 250. The inside diameter (ID) is 1.051, and the outer diameter (OD) is 1.315 in. The borehole thermal resistance per unit length was calculated by using Equation (2), and it is equal to R'b = 0.1234 $(ft.°F.hr/Btu)$[12,13].

Calculating The total Cooling and Heating Length

The average underground earth temperature was calculated from Signorelli and Kohl Equation (6) T_g = 56.6 °F. System's average ground volumetric heat capacity is 35 $Btu/(°F.ft^3)$.The maximum design heat pump entering fluid temperature is 90°F , and the minimum design heat pump entering fluid temperature is 32°F. The annual load factors are estimated 0.25 for heating and 0.3 for cooling, and the monthly load factors are 0.3 for heating and 0.4 for cooling. COP_c is 5 for cooling and COP_h is 3.3 for heating. Based on the above details and assumption, and by entering the data in the VERTICAL GHX package solver, the results were as, [14,15]

Table 3.Input data VERTICAL GHX package solver

Input data according to the load calculations		
Total heating	140,613,296.8	Btu/month
Total cooling	86,207,252.1	Btu/month
Peak heating	278,557.1	23.2130917 (tons)
Peak cooling	271,528.5	22.627375 (tons)
Mon. PLF	0.678483394	
Mon. PLF	0.426732231	

Central ground heat exchanger	**No. bores**	**Bore depth(ft)**	**Cooling length(ft)**	**Heating length(ft)**
	20	215	4,302	2,900

Layout of The Ground Boreholes Field

The best distribution for 20 bores in the field considering the property line and facilities of the school building.

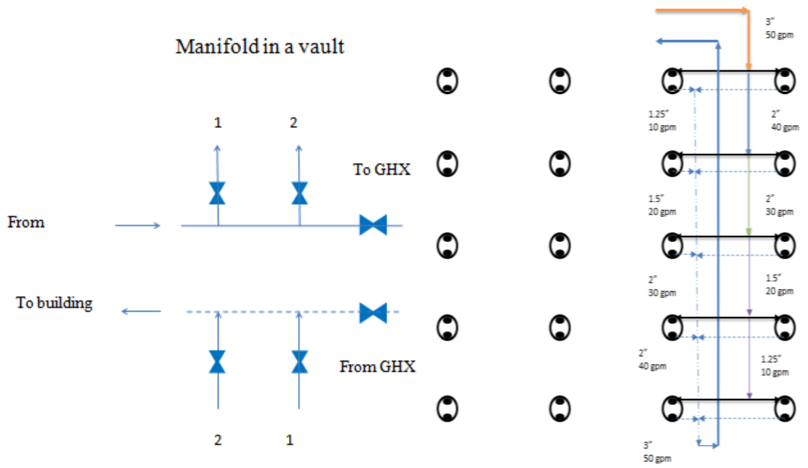

Figure 5. Layout of The Ground Boreholes Field.

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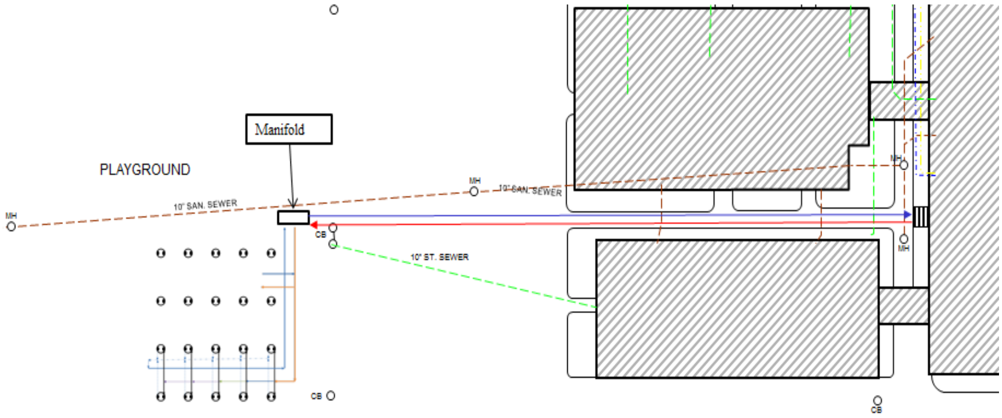

Figure 6.Design of the mechanical system inside the school.

Design of The Mechanical System

The design of the mechanical system inside the school building (piping routes and heat pump locations) is shown in Figure 9.

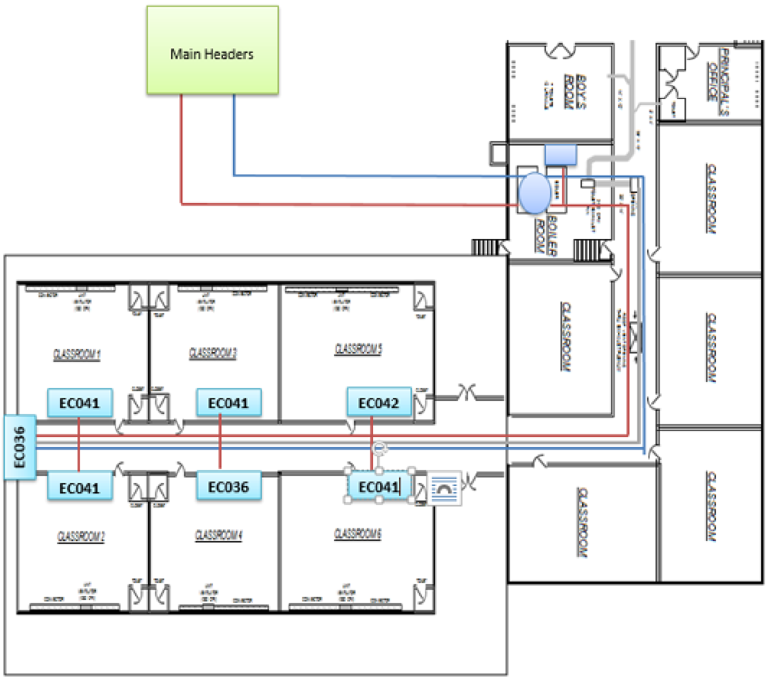

Figure 7. Layout of classes' boreholes and heat pump locations inside the school building.

Calculating the Head Loss.

The critical path is through the last heat pump, class 6 as shown in Figure 9. By Using E-PipeAlator 08.xls, the head loss is about 46.2 ft. The assumption of boreholes was 243 ft deep boreholes and 50 ft to the mechanical room. Piping head losses with valves and fitting losses were 18.6 ft. Heat pump losses were 16.4 ft, and the plate HX was 10.0 ft. Finally, the total losses were 45 ft [10,11].

Circulating Pump Capacity Calculation

By using Equation (14) and Equation (15), the pump worked at 60 gpm and 45 ft of water was 1.1 HP or 1.0257 watts.

Individual Boreholes per Classroom

Following the same procedure for designing central ground heat exchanger by using the same software to estimate the number of bores and the total cooling and heating length for each class individually, then using E-pipeAlator to calculate the pressure loss for each loop. The results are shown in the table below in Tables (4 and 5).

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Table 4. The total cooling and heating length for each class.

No. Class	***No. bores***	***Bore depth(ft)***	***Cooling length(ft)***	***Heating length (ft)***	***Flow rate (gpm)***
1	2	236	473	377	6.5
2	2	194	387	349	6.4
3	2	235	471	374	6.4
4	2	193	385	342	6.2
5	3	187	561	443	7.5
6	2	226	467	382	6.8
Corridor	2	226	449	382	5.5
Total	15	1,497 ft	3,193	3,193 ft	Avg = 6.47

***** The additional four bores to handle the ventilation loads by using a water-water heat p pump.*

Layout of the Ground Boreholes Field

The layout of the classes (2, 3) which have two boreholes.

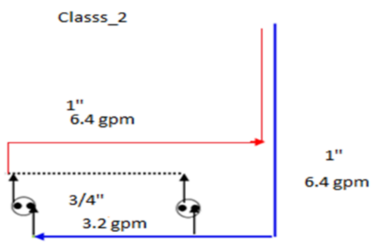

Figure 90.Layout of the classes 2 and 3.

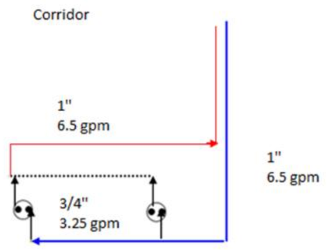

Figure 81. Layout of class 1 and corridor.

layout for class 4

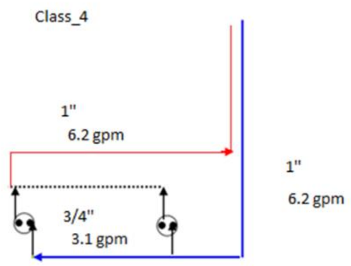

Figure 10.layout for class 4.

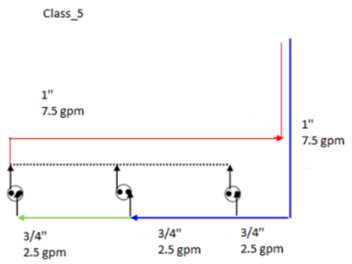

Figure 11.layout for class 5.

Layout for class 6

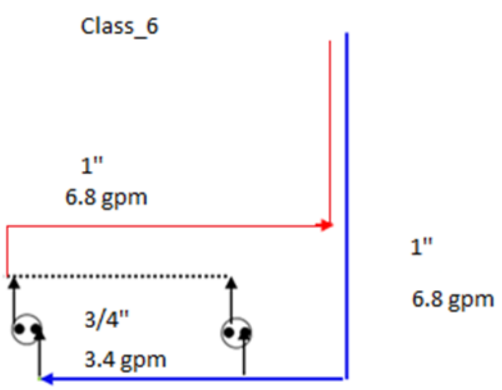

Figure 12. Layout for class 6.

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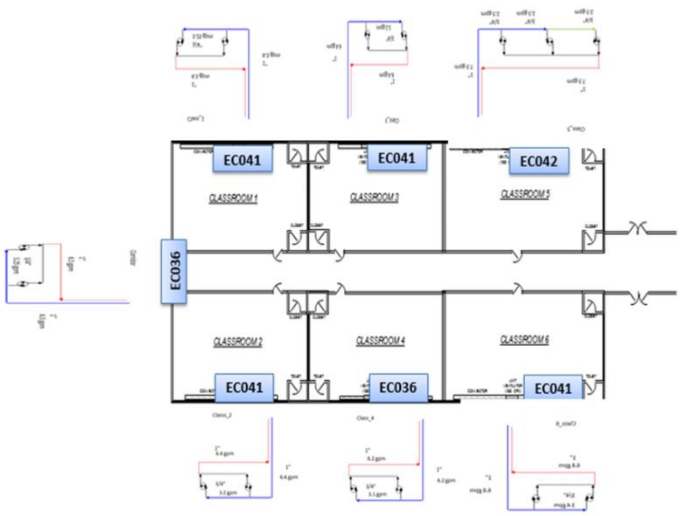

Figure 13. Layout of classes' boreholes and heat pump locations inside the school building.

Head Loss Calculations

In our system, we have three kinds of losses including minor and major losses, heat pump losses and plate heat exchanger losses. **E-pipealator** code was used to calculate the major and minor losses through the borehole pipes, connections, and heat pump borehole pipes as illustrated in the layout below. For heat pump losses the specification data sheet was used at fluid flow and pressure drop to get the FGH pressure drop. For plate heat exchanger losses, we assume that head losses 10 m. [11]

Table 5. Total head losses.

Class	Major-Minor losses	Heat Pump Losses	Plate HX[11]	Total head losses ft	Flow rate gpm
1	18.7	5.8	10	34.5	6.5
2	16.5	5.8	10	32.5	6.4
3	18.6	5.8	10	34.4	6.4
4	16.4	6.5	10	32.9	6.2
5	16.1	11	10	37.1	7.5
6	18.1	9.1	10	37.2	6.8
Corridor	18.1	9.1	10	37.2	6.5

Pump Sizing

The pump work in horsepower (hp) and (kw) is calculated by using Equations (14 and 15) and shown in Table (6).

Table 6. Pump sizing

Column1	Flow Rate	Head Looses	Pump	W pump	Pump Size
Class	gpm	ft	hp	kw	inches
1	6.5	34.5	0.16179	0.15087	5.5
2	6.4	32.5	0.15007	0.13994	5.5
3	6.4	34.4	0.15884	0.14812	5.5
4	6.2	32.9	0.14717	0.13723	5.5
5	7.5	37.1	0.20075	0.18720	6
6	6.8	37.2	0.18251	0.170191	6
Corridor	6.5	37.2	0.17445	0.16268	6
				1.09625	

Ventilation Air System Design

The ventilation air system design was based on the peak loads which are 43 kBtu/hr for heating and 48 kBtu/hr for cooling. Also, the monthly load factors for cooling and heating are 0.3 and 0.4 respectively. The design gave that the total flow rate is 10 gpm and 4 vertical boreholes with a total length of 205 ft for each borehole [12,14].

Discussion

The design of the geothermal system for the school was carefully tailored to the local climate and the facility's cooling requirements. The selected heat pumps (HPs) were sized to efficiently handle the cooling load, with

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backup heating systems incorporated to provide supplemental heat during colder periods when temperatures drop below specific thresholds. While this approach introduces additional costs, it ensures the building remains comfortable year-round. The design strategy prioritized effective cooling while preparing for heating demands, achieving a balanced and efficient solution. To optimize the performance of the Ground Heat Exchanger (GHX), we applied knowledge to select the most suitable design parameters. All assumptions made during the design process were intended to maximize thermal efficiency and avoid potential economic losses resulting from inefficient operation or mismatched GHX capacity. By focusing on thermal performance, the design ensures that the system operates at peak efficiency, minimizing long-term operational and maintenance costs. The results of our calculations for both central and individual GHX configurations revealed no significant difference in the total heating and cooling lengths of the GHX. This indicates that the system can be effectively optimized to meet both heating and cooling needs without requiring significant adjustments for either mode. This finding demonstrates the system's ability to maintain consistent thermal performance throughout seasonal variations while minimizing energy consumption and costs.

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المستخلص

تم تصميم نظام تدفئة وتهوية وتكييف هواء بمضخة حرارية أرضية لمدرسة XYZ Indianapolis باستخدام الطاقة المتجددة. تُعد الطاقة الحرارية الأرضية إحدى الطاقات المهمة التي أثبتت نفسها في أنظمة تكييف الهواء في جميع أنحاء العالم. تم تصنيف التصميم في ثلاث نقاط على النحو التالي. باستخدام مبادل حراري عمودي للبئر ومضخة حرارية وجميع معداتها مثل الغرفة الميكانيكية والأنابيب وتصميم التركيبات لتغطية الحمل الإجمالي للمدرسة، كانت النتائج 20 بئرًا ومعدل تدفق إجمالي 60 جالونًا في الدقيقة (3 جالونًا في الدقيقة / بئر) من محلول البروبيلين بنسبة 20٪ وتباعد 20 قدمًا مع 1.1 حصان و 6.5 في حجم المكره. بالنسبة لتصميم البئر الفردي، كانت النتائج 15 بئرًا واستهلاك إجمالي للطاقة 1.1 كيلو واط. بالنسبة لتصميم التهوية، يبلغ الحمل الأقصى للتدفئة 42779 وحدة حرارية بريطانية / ساعة، وللتبريد، يكون 47779 وحدة حرارية بريطانية / ساعة. يوجد أربعة آبار، ومضخة كهربائية بقدرة 0.5 حصان، بمعدل تدفق 10 جالون/دقيقة، مع انخفاض ضغط الماء بمقدار 40 قدمًا، ومروحة بقدرة 10 أحصنة مدمجة مع مضخة حرارية تعمل بتقنية الماء إلى الماء. الحمل الإجمالي الذي افترضناه لإمدادات التدفئة في الصف السادس والممر هو 10% و25% على التوالي. حُسبت المقاومة الحرارية لكل حالة وفقًا لمعدل تدفق السوائل. صُمم حمل التهوية للمدرسة بشكل منفصل عن أحمال التدفئة والتبريد، وهو متصل بمضخة حرارية تعمل بتقنية الماء إلى الماء، مما يضمن كفاءة عالية في تشغيل النظام. وأخيرًا، صممنا هذا المشروع باستخدام الطاقة الحرارية الأرضية، وهي أنظف من الطاقة التقليدية.

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