Earth Integrated Heating and Cooling System

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Certificate

This is to certify that the B. Tech project titled “Earth Integrated Heating and Cooling System” prepared by

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Abstract

Excessive greenhouse gas emission is a serious concern worldwide and scientists are in search of more environment friendly alternatives of the existing technologies. The search is still on for more environment friendly and efficient heating and cooling systems. Ground coupled heat exchanger that exchanges energy with the ground for heating and cooling purposes is a new and emerging technology with immense potential. Extensive research on increasing the performance and reducing emissions from the same is being done. In this study we modeled two GCHE technologies namely earth air heat exchanger (EAHE) and ground source heat pump (GSHP) for a two storey building in New Delhi. The design calculations, assumptions have been clearly mentioned and other work including python code file and DWSIM model used for system modeling are provided in the annexure.The monthly COP of the GSHP system are in the range of 0.9-3.8 and those of EAHE system are in the range 3.696 - 12.019 making it more efficient than most conventional systems having an average COP of 3.5. The system can further be improved with the integration of solar panels for heating of circulating fluid or for providing the power supply to run the system.

Keywords: *ground coupled heat exchanger, earth air heat exchanger, ground source heat pump*

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List of Abbreviations

GCHP : `Ground Coupled Heat Exchanger
EAHE : Earth Air Heat Exchanger
GSHP : Ground Source Heat Pump
HVAC : Heating, Ventilation, and Air Conditioning
MERV : Minimum Efficiency Reporting Values
HEPA : High efficiency Particulate air

List of Symbols

ρ : density
k : thermal conductivity
$\boldsymbol{v}$: velocity
Cp: heat capacity
qt : surface heat flux
h_{tot} : combined heat transfer coefficient
T_s : radiant surface temperature
T_{ref}: room temperature
$\eta 0$: efficiency of the cell at 298 K.
ηel : electrical efficiency
β : silicon efficiency temperature coefficient
Tcell : temperature of one cell
S : heat absorber panel surface area
Qt : total heat collected in the storage tank
Qp : constant heat input from a circulation pump
Qu : useful energy gain delivered by the collector to the storage tank
Qs : stored energy as specific heat in the panel
$\dot{m}$: fluid's mass flow rate
C_P : specific heat capacity
T : temperature at any section along the pipe
r : inner radius of the pipe
h : inside heat transfer coefficient
x : length coordinate,
T_w : pipe wall temperature or earth temperature.
Q : ventilation rate/volumetric flow rate of air
P : number of occupants
A : area of the ventilation zone
R_p : air flow required per person
R_a : air flow required per unit area
f : friction factor,
L : total length of the tube,
D : diameter of the tube,
va: velocity of the air
P : principal investment
CF : Cash flow
i : interest rate

1. Introduction

Widespread industrialisation and an increase in the standard of living of people has led to increased greenhouse gas emissions which cannot be sustained by the planet's ecosystem. The high heat trapping potential of these gasses has caused an increase in average atmospheric temperatures worldwide, and other undesirable effects on climatic patterns. To escape the impact of resulting oppressive weather conditions, it is necessary to use heating and cooling systems to maintain thermal comfort indoors. This helps occupants of a conditioned space remain at ease, and ensures higher levels of alertness, agility and productivity, and maintain better health, all of which are essential to carry out daily activities efficiently.

Out of the total energy consumption in buildings, a large chunk (35%) is spent on heating, ventilation and air conditioning (HVAC) requirements [1]. High energy use by conventional HVAC systems and the corresponding CO_2 emissions, and heat pumping to the atmosphere further add to the already alarming rise of global temperatures, resulting in a decrease in energy demand for heating but a much larger increase for cooling. A Swiss study predicts heating demand to fall by 36-58% and cooling demand to increase by a whopping 223-1050% by 2100 [2]. The increase in cooling demand as a percentage of total households can be seen in Fig. 1. This constitutes a vicious climate change positive feedback loop which must be broken at all costs. All this coupled with the frequent breakdown of equipment, associated maintenance and the risk of leakage of toxic working fluids to the environment makes the operation of conventional HVAC systems simply inefficient and unsustainable.

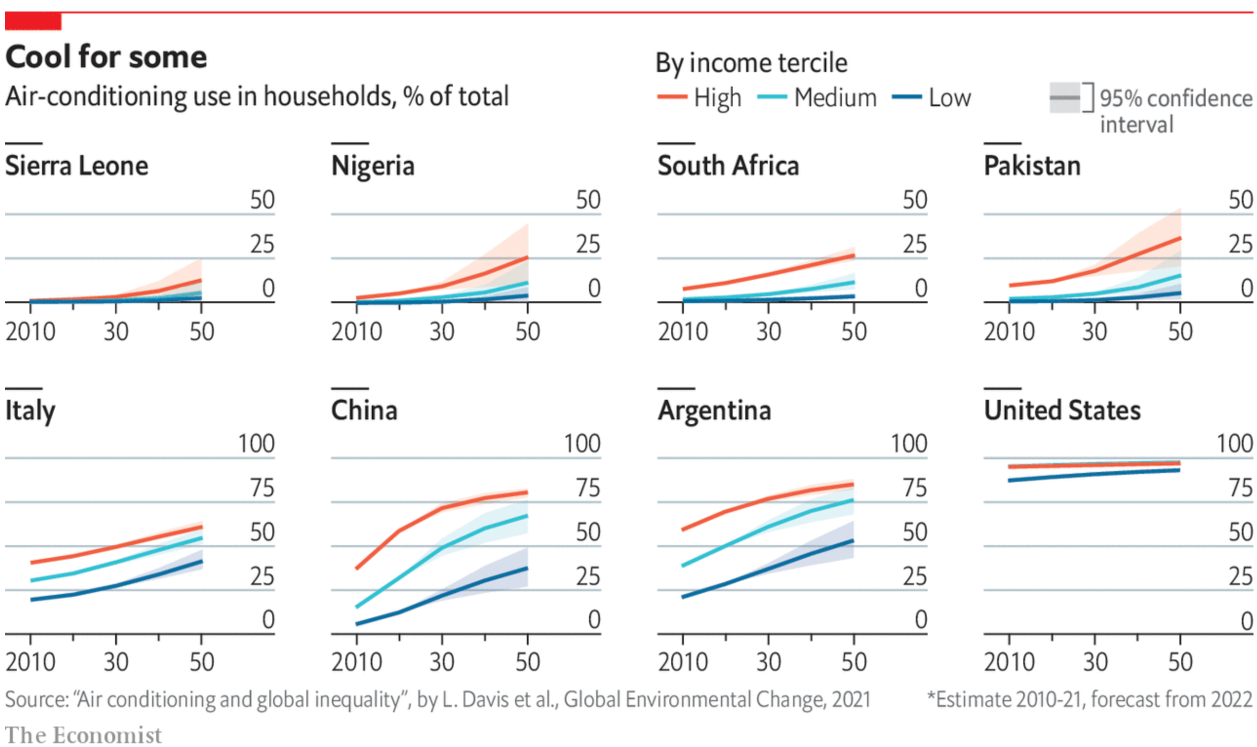


Fig. 1: Increase in Cooling Demand in Various Countries Adapted from [3]

Based on the above discussion, there is a dire need to adopt novel, energy efficient and environmentally benign technologies for heating and cooling of spaces. Some of these technologies exploit the phase change properties of substances, while some others work on the principle of energy storage in salt solutions. Another class of technologies based on geothermal energy is gaining widespread adoption and research interest. These systems can be coupled with renewable energy sources such as solar panels to reduce CO_2 emissions and grid dependence while meeting electricity demand for operating the system components.

Underground heat exchangers (geothermal based) that exchange energy with the ground for heating and cooling purposes are called Ground Coupled Heat Exchangers (GCHEs). The ground, whose temperature is relatively constant throughout the year, acts as a heat source in winters and as a heat sink in summers. Such systems have the advantage of being highly energy-efficient, environmentally friendly, easy to control, noise-free, cost-effective, providing good thermal comfort, having a stable capacity and requiring simple equipment. Disadvantages include high upfront costs and a lack of trained technicians and contractors [4]. However, the disadvantages are greatly outweighed by the advantages, because of which GCHEs have attracted widespread attention. Despite this, not a lot of research is available on the topic, especially those which compare two different technologies belonging to the class of GCHEs. Hence, in this study, we will be comparing the performance of two different GCHE technologies: Earth-Air Heat Exchangers (EAHEs) and Ground Source Heat Pumps (GSHPs) in a setting based out of New Delhi.

2. Literature Survey

2.1. Earth Air Heat Exchanger

The earth-air heat exchanger (EAHE) is a series of pipes buried underground at a particular depth through which fresh atmospheric air flows and gets cooled in summer and warmed in winter. The ambient air passing through the pipes exchanges heat with the pipe walls, which are in direct touch with the surrounding subsurface environment, transmitting heat via conduction and convection processes. The opposite end, which acts as an outlet, allows air to enter a structure. The thermal characteristics of the ground and the heat transfer between the pipes and the ground are closely connected to the performance of EAHE. Conduction, convection, and radiation are the heat transport processes in soil. Conduction occurs throughout the soil, although the solid and liquid elements carry the majority of the heat. With the exception of quick water penetration following significant rainfall, convection is typically modest.

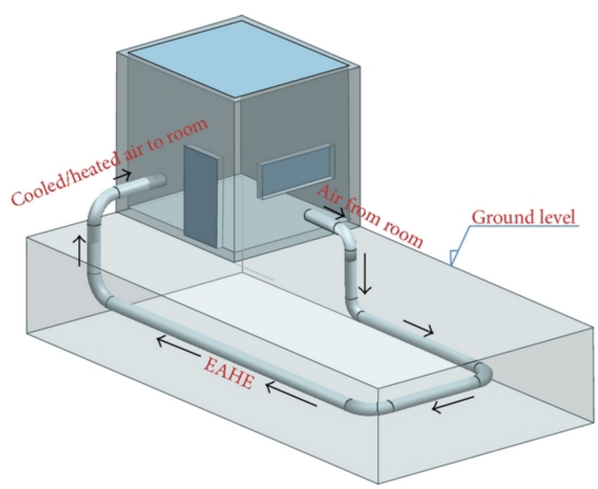


Fig. 2: Schematic of Earth-Air Heat Exchanger System. Adapted from [5].

Here for our design, we are building a system that works for a two-storey building. The main idea is to use the air present in the atmosphere to heat the environment inside the house in winters and cool it during summers. The material of the pipe to be laid underground depends on various factors. The comparison table is displayed below:

Table 1: Comparison Table for Tube material Selection Analysis. Adapted from [6].

S.no.	Material	Conductivity in W/mk	Resistance to corrosion	Cost
1	Copper	385	Medium	Very Costly
2	Aluminium	205	High	Costly
3	Brass	109	Medium	Costly
4	Iron	79.5	Low	Costly
5	Steel	50.2	High	Costly
6	PVC	0.19	High	Cheaper

PVC pipes are generally used as they are readily available, inexpensive, and easy to install. But the main disadvantage is it gets brittle over time and can crack, break, or even shatter. The presence of air compressor lubricants in the line, as well as heat from compressed air, hastens the deterioration of PVC. Due to the airborne, razor-sharp shrapnel, these failures, when paired with pressured air, have the potential to be lethal. While aluminum is non-corrosive and remains leak-free, unlike other pipe systems. Considering the high cost of producing compressed air, reducing compressed air consumption by eliminating leaks over the life of the system, the aluminum pipe becomes a more recommendable material.

Since the air blown inside is directly sucked from the atmosphere, it can be contaminated, hence we will need air filters. Some of the air filters are listed below.

Table 2: Comparison Table of Air Filters. Adapted from [7].

S.no.	Air filters	Efficiency	Pros	Cons
1	HEPA	99.97%	Captures large pollutants, cost effective, and changes in a few years.	Ineffective against fumes, gasses and odors. Mold sores settle on the filter.
2	UV light	99%	Kills molds, germs, exceptional indoor quality.	Costly. Ineffective to dust or allergens, gases, fumes, cigarettes. O2 to O3
3	Electrostatic	98%	Cost effective, reusable	Struggle to filter large particles, poor choice for people with respiratory issues

4	Washable	75%	Cost-effective, washable, Long lasting.	High maintenance, Complete drying after washing else source to pathogens.
5	Media filters	<20%	Low maintenance, large surface area, traps pollutants	Ineffective in filtering odours.
6	Spun glass	20%	Cheap, capture lint and dust.	Small surface area, ineffective against smaller pollutants, a poor choice for asthma patients, gets clogged.
7	Pleated	20-25%	Traps debris, large surface area, lasts longer, reusable.	Expensive, heat/cooling system will lose efficiency sooner

The MERV ratings play a vital role in choosing the best air filters. MERV stands for minimum efficiency reporting values.

Table 3: MERV Ratings for different types of filters. Adapted from [8].

MERV Rating	Air Filter will trap Air Particles size .3 to 1.0 microns	Air Filter will trap Air Particles size 1.0 to 3.0 microns	Air Filter will trap Air Particles size 3 to 10 microns	Filter Type ~ Removes These Particles
MERV 1	<20%	<20%	<20%	Fiberglass & Aluminum Mesh ~ Pollen, Dust Mites, Spray Paint, Carpet Fibres
MERV 2	<20%	<20%	<20%	
MERV 3	<20%	<20%	<20%	
MERV 4	<20%	<20%	<20%	
MERV 5	<20%	<20%	20% - 34%	Cheap Disposable Filters ~ Mold Spores, Cooking Dusts, Hair Spray, Furniture Polish
MERV 6	<20%	<20%	35% -49%	
MERV 7	<20%	<20%	50% - 69%	
MERV 8	<20%	<20%	70 - 85%	
MERV 9	<20%	Less Than 50%	85% or Better	Better Home Box Filters ~ Lead Dust, Flour, Auto Fumes,Welding Fumes
MERV 10	<20%	50% - 64%	85% or Better	
MERV 11	<20%	65% -79%	85% or Better	
MERV 12	<20%	80% - 90%	90% or Better	
MERV 13	Less Than 75%	90% or Better	90% or Better	Superior Commercial Filters ~ Bacteria, Smoke, Sneezes
MERV 14	75% - 84%	90% or Better	90% or Better	
MERV 15	85% - 94%	95% or Better	90% or Better	
MERV 16	95% or Better	95% or Better	90% or Better	
MERV 17 - HEPA 13	99.97%	99% or Better	99% or Better	HEPA & ULPA ~ Viruses, Carbon Dust, <.30 pm
MERV 18- HEPA 14	99.997%	99% or Better	99% or Better	
MERV 19 - UL5	99.9997%	99% or Better	99% or Better	
MERV20- U16	99.99997%	99% or Better	99% or Better	

Illustration Provided by LakeAir / www.lakeair.com

The MERV value is from 1 to 20. Higher MERV values correspond to a greater percentage of particles captured on each pass. The scale is designed to represent the performance of a filter when dealing with particles in the range of 0.3 to 10 micrometers. With a MERV 16 or above filter capturing more than 95% of particles over the full range is possible. We are using an air blower, a single pipe heat exchanger, air filters to enhance the air quality and pipes to be laid underground.

2.2. Ground Source Heat Pump (GSHP)

A ground source heat pump is a heating/cooling system that employs a heat pump to transfer heat to or from the ground, taking advantage of the earth's relatively constant temperatures throughout the seasons. It consists of three units whose functionalities are explained below.

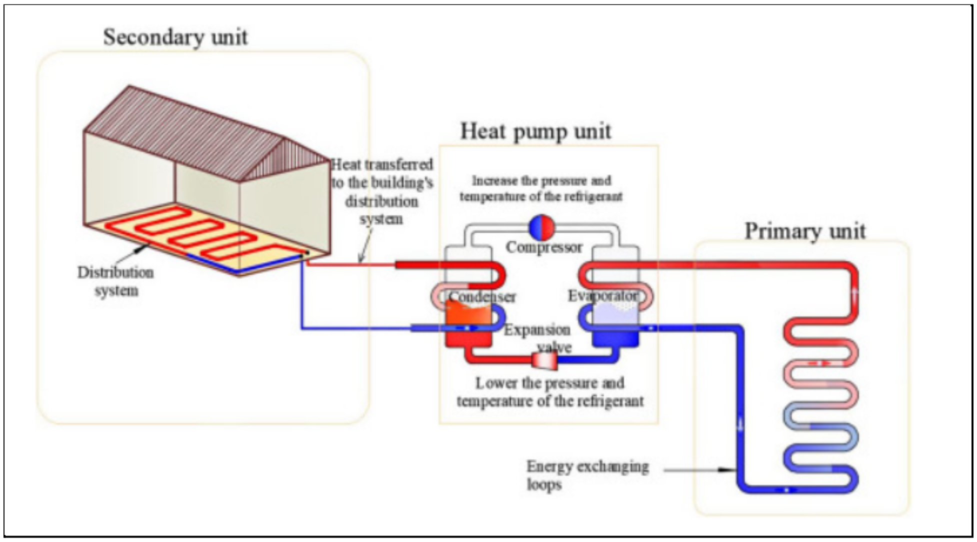


Fig. 3: Schematic of Ground Source Heat Pump (GSHP) System
Source: Adapted from [9].

1. <u>Heat Exchanger Unit:</u> Facilitates transfer of heat to or from the ground to the circulating fluid which is ultimately brought in thermal contact with the circulating fluid in the heat pump.

2. <u>Heat Pump Unit:</u> Extraction of heat from the circulating fluid in the heat exchanger and transferring it to the liquid in the secondary loop or the building loop

3. <u>Secondary Loop/Building Loop:</u> Comprises of a piping network for transfer of heat in and out of the building

2.2.1 Heat Exchanger

There are two types of GSHP systems: Vertical GSHP and Horizontal GSHP. Vertical GSHP consists of a vertical U-tube heat exchanger and is installed using deep drilled boreholes. It requires less outdoor space and is more energy efficient compared to horizontal GSHP systems. Considering the following advantages of the vertical GSHP system, we have selected it for our model. It consists of a <u>vertical U-tube Heat Exchanger</u>.

A U-tube Heat Exchanger consists of a circulating fluid, U- tube, grout and the surrounding ground. A U-bend configuration is used where the two vertical U-tube pipes are connected with 180 degree fitting. As the fluid descends, the temperature difference in the fluid and the earth mass causes heat flow. In our model, we have taken <u>water as the coolant</u>.

For designing the heat exchanger, we require certain physical properties such as thermal conductivity, viscosity and specific heat capacity. All these values have been taken from Process Heat Transfer by DQ Kern.

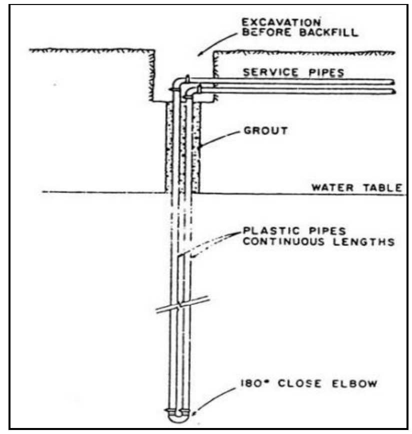


Fig 4: U-tube heat exchanger. Adapted from [9]

Table 4: Physical properties of water. Adapted from [10]

Density (ρ)	998.23 kg/m^3
Thermal Conductivity (k)	$6.15 * 10^{-7}$ kW/m ℃
Viscosity (ν)	$0.86 * 10^{-3}$ Kg/m.s
Specific Heat Capacity (C)	4.179 k J kg^{-1} K^{-1}

Table 5: U - tube dimension. Adapted from [10]

Tube OD	19.065 mm
Tube ID	15.392 mm
BWG	15
Length	6 m

2.2.2 Heat pump

A heat pump is a mechanical device that is used to transfer thermal energy between spaces using electricity much like a refrigerator. When employed in a GSHP system, in the winter season it is used to supply energy in the house for heating purposes and in the summer season it is used to remove heat from the house for cooling. It consists of four subcomponents namely evaporator, compressor, condenser and expansion valve along with a circulating fluid or refrigerant.

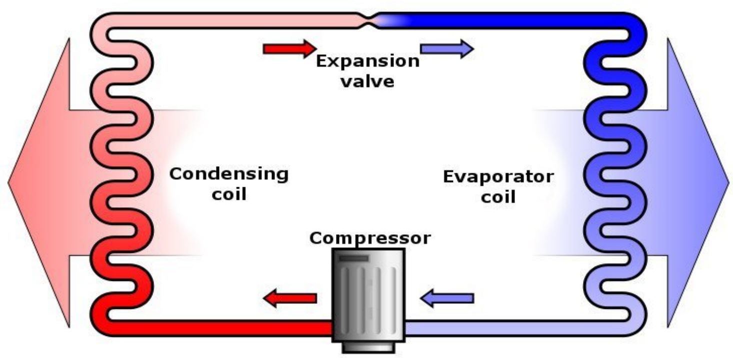


Fig 5: Heat Pump Schematic. Adapted from [11]

The heat pump cycle for space heating begins with extraction of heat from the heat carrier fluid which in our case is the working fluid of the heat exchanger by the refrigerant in the evaporator. This causes the refrigerant in the evaporator to boil, becoming a low pressure vapor which is then fed into the compressor. After compression, the refrigerant now at a higher temperature and pressure, passes into the condenser. At the condenser, the high pressure vapor transfers its energy to the secondary loop or the building loop and condenses into liquid form. From there it goes into the expansion valve which further reduces its temperature and pressure and the cycle repeats again. For space cooling, the working principle of the system is reversed. [17]

The performance indicator for a heat pump is its coefficient of performance or COP which is defined as the ratio of input power to the heat pump and output power.

$$COP = \frac{power\ input\ to\ the\ heat\ pump}{power\ output} \tag{1}$$

Taking environmental concerns into account, the choice of refrigerant for the heat pump is also very important. An ideal refrigerant should be one that does not contribute towards ozone layer depletion and has a minimum global warming potential. Table below visualizes the difference in properties of the earlier used R22 ($CHClF_2$) and R290 (propane) as a refrigerant.

Table 6: Characteristics of R22 & R29. Adapted from [12]

	R22	*R290*
ODP	0.055	0
GWP	1300	3
Molecular mass (kg/kmol)	86.5	44.103
Normal boiling point (°C)	–40.8	–42.1
Critical temperature (°C)	96.0	96.7

2.2.3 Radiant Heating/Cooling System

A popular category of heating, ventilation and air conditioning system (HVAC) are radiant heating and cooling systems in which heat is exchanged by both convection and radiation. The total amount of heat removed by a radiant surface can either be calculated separately as cited in Chapter 6 of ASHRAE Handbook, HVAC Systems and Equipment (ASHRAE 2012a) or separately using a combined heat transfer coefficient as recommended by ISO 11855 [36]. For the design of radiant systems the latter is generally used and the formulae for the same is given below.

$$q_t = h_{tot}\left|T_s - T_{ref}\right| \tag{2}$$

Where, q_t is surface heat flux

h_{tot} : combined heat transfer coefficient

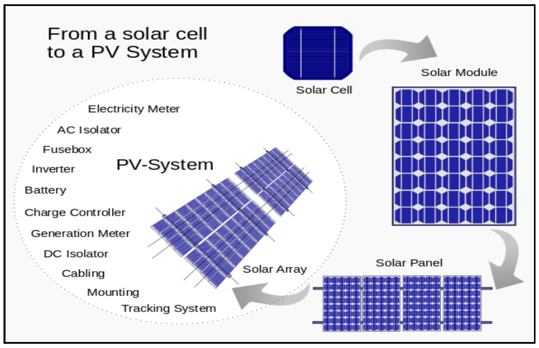
T_s : radiant surface temperature

T_{ref} : room temperature

2.3 Solar Panel

2.3.1 Introduction

A solar panel which is also known as a photovoltaic module is a device which generates the direct current (D.C) electricity that uses the sunlight as an energy source. The assembly of solar cells which are mounted in a specific frame for installation is called solar panels or PV panels and the assembly of these panels is called arrays.



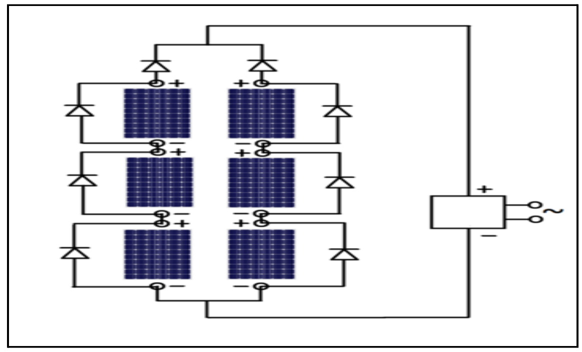


Fig 6: Solar cell to PV system Adapted from [13]

Fig 7: Connection of Panels with diode Adapted from [13]

2.3.2 Integration of solar panels :

Integrated solar is when solar electricity is built in as part of the original structure rather than as a separate module. Solar energy can be collected in two ways: thermally or through photovoltaic cells. Integrated solar panels are weatherproof and as tough as regular roof tiles, but they also feature photovoltaic cells for electricity generation. During the heating season, mounted solar panels will collect energy and store it in seasonal heat storage.

2.3.3 Performance and efficiency analysis of solar panels :

Solar cells convert solar energy into electricity with a very low efficiency of less than 20%, with more than 80% of the energy being dumped into the environment.There are two types of solar panels which are crystalline and non-crystalline panels made of silicon. Poly-crystalline panels are cheaper and easier to manufacture whereas mono-crystalline panels are most efficient but production cost is high. Around 80% of total solar panels are based on crystalline technology which are mono-crystalline and poly-crystalline silicon solar panels and non crystalline technology based on thin-film solar panels.

Efficiency of the solar cells are defined as the electrical and thermal efficiency the formula for the same is,

$$\eta_{el} = \eta_0[1+ \beta(T_{cell} - 298\ K) \quad (3)$$

Where, η_0 is the efficiency of the cell at 298 K, η_{el} is electrical efficiency, β is silicon efficiency temperature coefficient and Tcell is temperature of one cell.

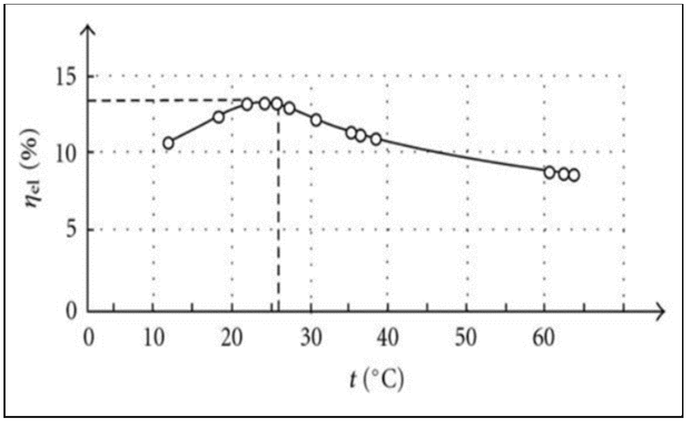


Fig 8: Efficiency w.r.t Temperature plot
Adapted from [14]

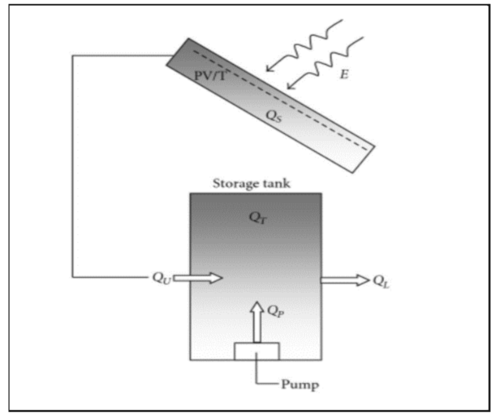


Fig 9: Energy flow in the system
Adapted from [15]

Thermal efficiency is defined as follows,

$$\eta_T = (Q_U + Q_S)/E\ .S \quad = \quad (Q_T + Q_L - Q_P + Q_S)/E\ .S \quad (4)$$

Where, S : heat absorber panel surface area

Qt : total heat collected in the storage tank

Qp : constant heat input from a circulation pump

Qu : useful energy gain delivered by the collector to the storage tank

Qs : stored energy as specific heat in the panel

Efficiency of solar panels also depends on the angles so when we integrate the solar panels then it should be considered that solar panels should be put in the right direction with an optimum angle so we can get the maximum sunlight on the arrays and then can get maximum efficiency.

So here basically we use some terms of angles like Tilt angle which decides the optimum angle to get max sunlight on the solar panels. , Azimuth angle defines the direction of the sun for the solar panels and it varies throughout the day and other angles are angle of altitude, angle of latitude

Energy produced by a solar panel is calculated by,

$$E = A \times r \times H \times PR \quad (5)$$

Where, A = solar panel area(m^2)
r = solar panel efficiency
H = Annual average solar radiation
PR = performance ratio
(default value = 0.75)

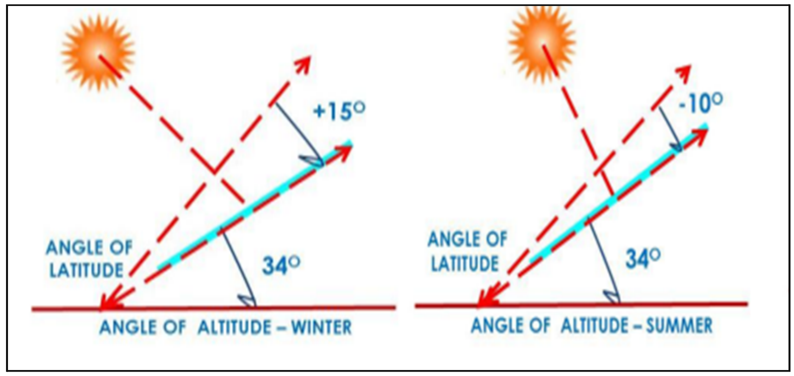


Fig 10: Angle of Altitude and Latitude in Different Seasons. Adapted from [16]

2.4. Data collection

2.4.1 Ground Temperature Data for New Delhi

The monthly distribution of the earth's temperature in New Delhi at a depth of 1.5 m can be seen in Fig. 11. This gives the limiting temperature that the working fluid can reach when exchanging heat with the ground.

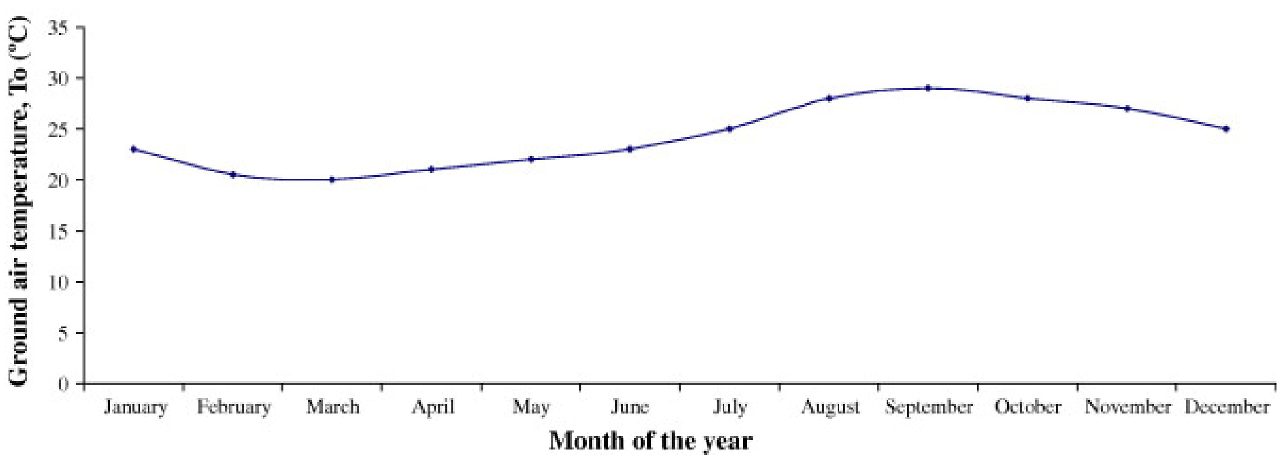


Fig 11: Monthly Average Ground Temperature for New Delhi at a Depth of 1.5 m. Adapted from [18]

2.4.2 Climatic Data for New Delhi

The climatic data for New Delhi is shown in Table 7. It helps understand the annual weather patterns in New Delhi, and will primarily be used in calculations pertaining to the operation of the earth air heat exchanger (EAHE).

Table 7: New Delhi Climatic Data. Adapted from [19]

Month	High / Low (°C)
January	20° / 8°
February	24° / 11°
March	30° / 16°
April	37° / 23°
May	40° / 27°
June	39° / 28°
July	35° / 28°
August	34° / 27°
September	34° / 25°
October	33° / 21°
November	28° / 14°
December	22° / 9°

3. Problem Statement

Earth experiences climate change from extreme winters to extreme summers. Pertaining to the irregular climate change in past few decades humans found it difficult to sustain. Excessive usage of air conditioners is the leading cause of atmospheric degradation. In this report we propose an environment friendly method for heating and cooling of houses in winters and summers respectively. In our problem we propose to build a model based on GCHE or Ground Coupled Heat Exchanger technology. This technology has two types of systems EAHE (Earth Air Heat Exchanger) and GSHP (Ground source heat pump). Our main aim is to build an efficient system that can be used during extreme climatic conditions in India while keeping cost in mind. Specifically, we are proposing this model for a two storey residential building in New Delhi occupied by 4 people. The area of each storey is 300 sq ft, giving the total area of the building as 600 sq ft.

4. Methodology, Results and Discussion

4.1 Determining Heating and Cooling Loads

Since the project is based on providing thermal comfort inside a house, the first step after defining the problem statement and project scope is to determine the heating and cooling loads for the house in question. A simulation software called 'DesignBuilder', widely used by engineers for building design and energy simulations, was employed for the purpose. A mockup of a two storey building (area of each storey = 300 sq ft) having a window wall ratio (WWR) of 30% and housing 4 occupants was created, with the geographical area set to New Delhi. The total space volume is 6889.77 cubic ft. The U value for the roof is 0.25 W m^{-2} K^{-1}, and that for the walls is 0.352 W m^{-2} K^{-1}. Solar panels of area 300 sq ft were added to the roof to see if they could fulfill the heating and cooling requirements. The simulation was then run to obtain the results.

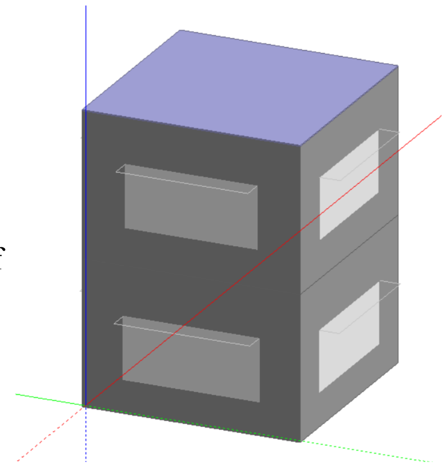
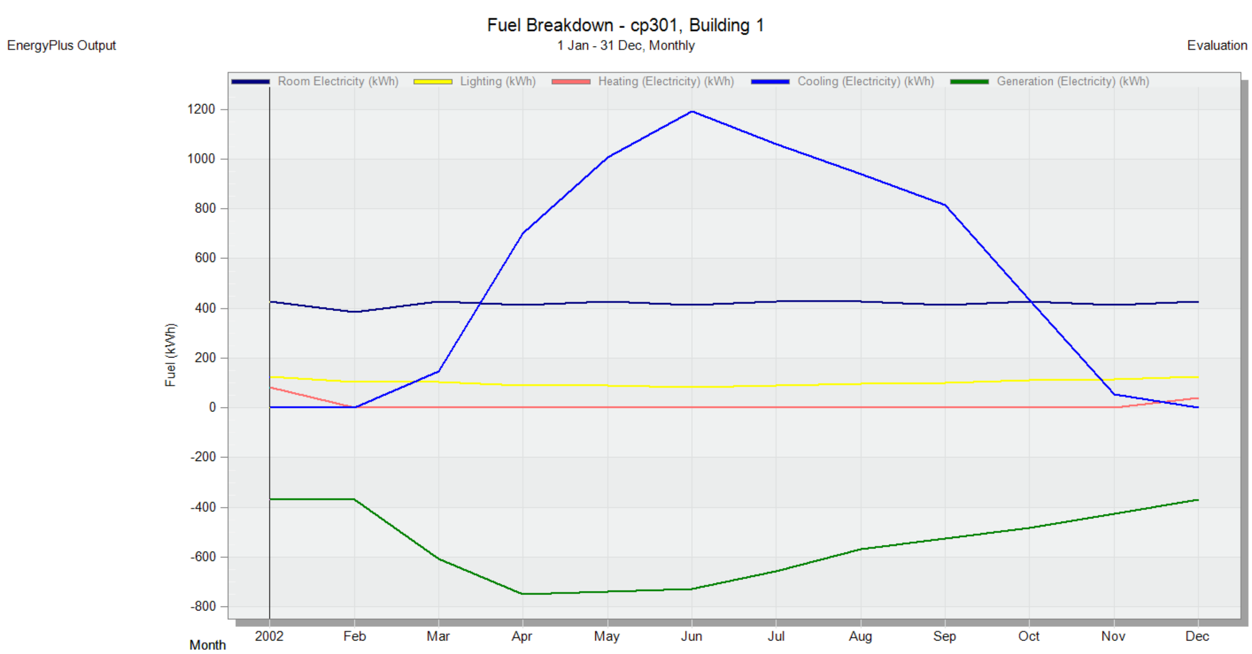
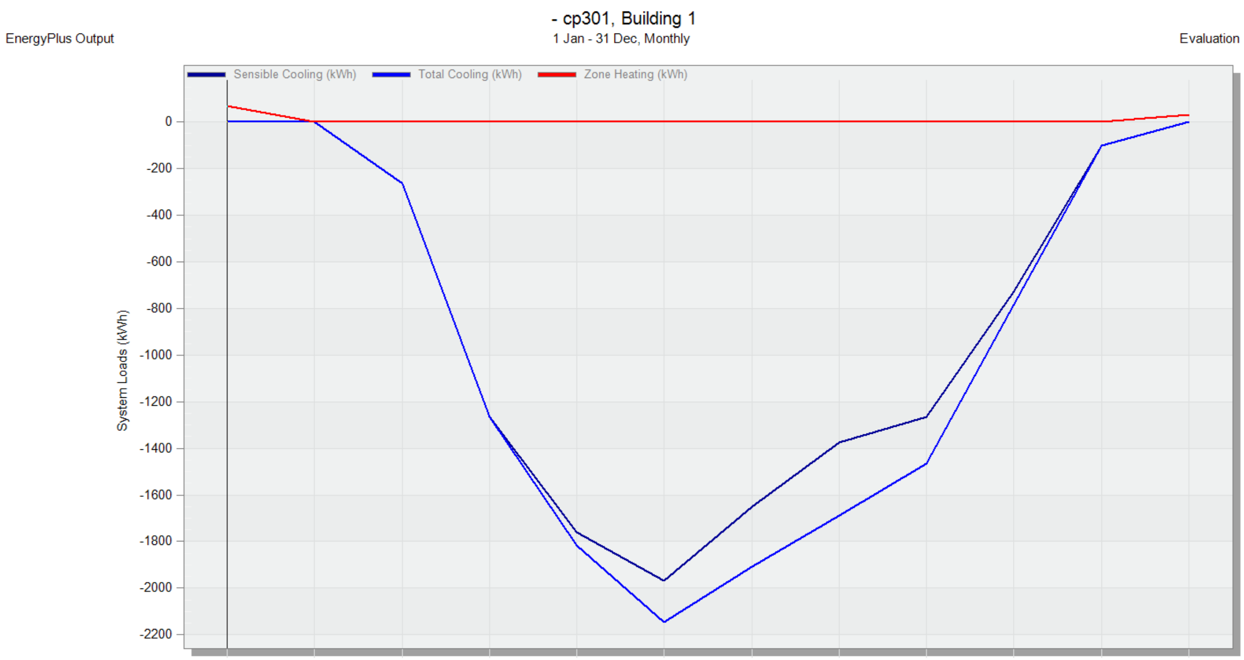


Fig 12 : Target Building Mockup



	2002	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
Room Electricity (kWh)	427.57	386.19	427.57	413.77	427.57	413.77	427.57	427.57	413.77	427.57	413.77	427.57
Lighting (kWh)	125.56	105.20	103.71	88.16	90.61	80.78	88.64	95.74	99.26	111.52	115.71	123.41
Heating (Electricity) (kWh)	82.01	2.46	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	38.73
Cooling (Electricity) (kWh)	0.00	0.43	146.05	702.06	1008.55	1192.63	1060.49	938.51	814.11	435.45	55.25	0.00
Generation (Electricity) (kWh)	-367.39	-370.32	-605.84	-750.26	-740.67	-729.56	-658.36	-567.77	-525.99	-481.65	-425.07	-368.74

Fig 13: Electricity Consumption/Generation plot



Month	2002	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
Sensible Cooling (kWh)	0.00	-0.78	-262.90	-1263.71	-1760.26	-1967.11	-1650.95	-1375.62	-1266.19	-727.20	-99.46	0.00
Total Cooling (kWh)	0.00	-0.78	-262.90	-1263.71	-1815.38	-2146.73	-1908.89	-1689.32	-1465.40	-783.81	-99.46	0.00
Zone Heating (kWh)	69.71	2.09	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	32.92

Fig 14: Heating and Cooling Loads

4.2 Heat Transfer Governing Equation

The next step was to derive a governing equation for the heat transfer operation. To do this, an energy balance is carried out on a section of the horizontal heat exchanger pipe.

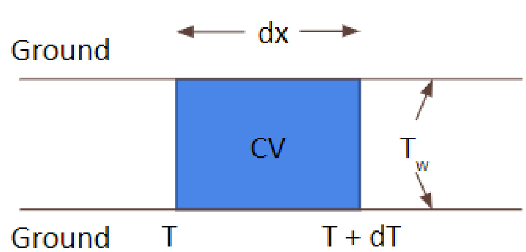


Fig 15: Pipe Section

In deriving the equation, we assume:

1. Steady state
2. Uniform flow
3. Constant earth temperature
4. Temperature of pipe walls equals earth temperature
5. Heat transfer resistance is only due to convection

The energy balance on the control volume (CV) gives:

$$\dot{m}C_P\frac{T+dT-T}{dx} = 2\pi rh\,dx(T_w - T) \tag{6}$$

where $\dot{m}$ is the fluid's mass flow rate, C_P is its specific heat capacity, T is its temperature at any section along the pipe, r is the inner radius of the pipe, h is the inside heat transfer coefficient, x is the length coordinate, and T_w is the pipe wall temperature = earth temperature. Upon integrating (6), using the initial condition that $T = T_{in}$ (inlet temperature) at x = 0, and rearranging, we get the relation for outlet temperature (T_{out}) as (please refer to the Annexure for the complete derivation):

$$T_{out} = T_w + (T_{in} - T_w)e^{-\frac{2\pi rhx}{\dot{m}C_P}} \tag{7}$$

This equation can also be used for vertical heat exchanger pipes if variation of the earth's temperature with depth is neglected. The value of Earth's temperature in New Delhi for each month can be read from Fig. 11 to be used as T_w in (7).

4.3. Earth Air Heat Exchanger (EAHE)

4.3.1. Heat Transfer Equation Accuracy

Before using the derived equation (7) for calculations specific to our case, it is essential to gauge its accuracy if possible. For this, we compare the equation's output with results of an experiment carried out at Ajmer. Here, an EAHE having an inner diameter of 0.15 m and length of 23.42 m is used for heating and cooling of air (density = 1.225 kg/m^3, specific heat capacity = 1006 J kg^{-1} K^{-1}). The air velocity v (and hence mass flow rate) is varied from 2 to 5 m/s. The months of January (inlet air temperature = 20.6 ºC and ground temperature = 26 ºC) and March (inlet air temperature = 43.7 ºC and ground temperature = 30 ºC) are considered [20, 21]. The only missing parameter is the inside heat transfer coefficient h which can be calculated using the correlation h = 2.8 + 3v (8), given by A. Chel et al [18]. The results are compared in Tables 8 and 9.

Table 8: Experimental vs Calculated Results for Heating in January
Data Source: [20]

	2 m/s	3.2 m/s	4 m/s	5 m/s
Exit temperature (Experimental)	25.4	25.1	24.9	24.7
Exit temperature (Calculated)	25.42	25.24	25.17	25.11
% difference	0.076%	0.567%	1.092%	1.664%

Table 9: Experimental vs Calculated Results for Cooling in January
Data Source: [21]

	2 m/s	3.2 m/s	4 m/s	5 m/s
Exit temperature (Experimental)	31	32	32.5	33.7
Exit temperature (Calculated)	31.47	31.92	32.10	32.26
% difference	1.527%	0.242%	1.228%	4.287%

The percentage error between calculated and experimental results is less than 5% indicating a high degree of accuracy, which means that the equation together with the heat transfer coefficient correlation is good for deployment in real world scenarios.

4.3.2. Building Ventilation Requirements

Before we use the heat transfer equation for our case, its parameters must be determined.

The mass flow rate of air, its velocity and heat transfer coefficient will depend on the ventilation requirements of the building. This is given by:

$$Q = R_pP + R_aA \tag{9}$$

where Q is the the ventilation rate/volumetric flow rate of air, P is the number of occupants (4 here), A is the area of the ventilation zone (600 sq ft here), R_p is the air flow required per person (5 cfm/person for residential area) and R_a is the air flow required per unit area (0.06 cfm/sq ft for residential area). The values of R_p and R_a are obtained from the ASHRAE 62.1 standards [22]. Using (9), the volumetric flow rate of air is obtained as 56 cfm or 0.02643 m^3/s. A metric called Air Changes per Hour (ACH) is used to gauge the ventilation of a given space. It is given by:

$$ACH = \frac{60 \times Q}{Space\ Volume} \tag{10}$$

Equation (10) gives an ACH of 0.49, which is greater than the 0.35 recommended by ASHRAE for residential areas. Thus, we can safely restrict the operation of our EAHE to this flow rate. The operation can be sustained by the MB840-D blower made by Oriental Motor, having an impeller diameter of 3.15 in and priced at USD 76. Fig. 16 shows the blower, and Table 10 its specifications [23]. It will be operated at a 50 Hz single phase voltage of 200 V, corresponding to an input power of 28 W.

Fig. 16 : MB840-D Blower. Adapted from [23]

Table 10: Blower Specifications
Source: [23]

Model	Voltage	Frequency	Current	Input	Speed	Max. Air Flow		Max. Static Pressure		Noise Level	Capacitor
	VAC	Hz	A	W	r/min	m^3/min	CFM	Pa	inH_2O	dB (A)	µF
MB840-D	Single-Phase 200	50	0.14	28	2800	1.6	56.5	152	0.61	55	2.5
	Single-Phase 200	60	0.18	32	3200	1.8	63.5	221	0.886	58	
	Single-Phase 220	60	0.18	35	3350	1.8	63.5	226	0.906	59	
	Single-Phase 230	50	0.15	35	2850	1.6	56.5	157	0.63	55	
	Single-Phase 230	60	0.18	36	3350	1.8	63.5	226	0.906	59	

4.3.3. Other Calculation Parameters

For this problem, we have chosen a heat exchanger pipe having inner diameter 0.15 m and length 30.48 m. Based on this, the heat transfer area comes out to be 14.363 m^2, the air velocity is 1.4956 m/s, and the inside heat transfer coefficient is 7.2868 W m^{-2} K^{-1}. The mass flow rate can be calculated by multiplying the volumetric flow rate with the density. The density of the air, however, will vary with temperature, and so will the viscosity. These properties will be used in heat exchanger calculations corresponding to every month of the year, and looking up the values every time or reading them from graphs can be tedious. To circumvent this issue, simple correlations have been developed with the help of scipy.optimize library in Python. The training datasets for both properties include values at -10, 0, 10, 20, 30, 40, 50 and 60 ºC and 1 bar, while testing datasets include values at 5, 15, 25 and 35 ºC and 1 bar (data source: The Engineering Toolbox [24, 25]).

A 2nd order curve of the form $aT^2 + bT + c$ is fit to the density training data. The R^2 and Root Mean Square Error (RMSE) for the training data are 0.999975 and 0.000464 respectively, and those for the testing data are 0.999707 and 0.000779 respectively. The equation parameters are shown in Table 11, and the curve fit in Fig. 17.

Table 11 : Constants of 2nd Order Density Curve

a	1.38690477e-05
b	-4.69226191e-03
c	1.29223214e+00

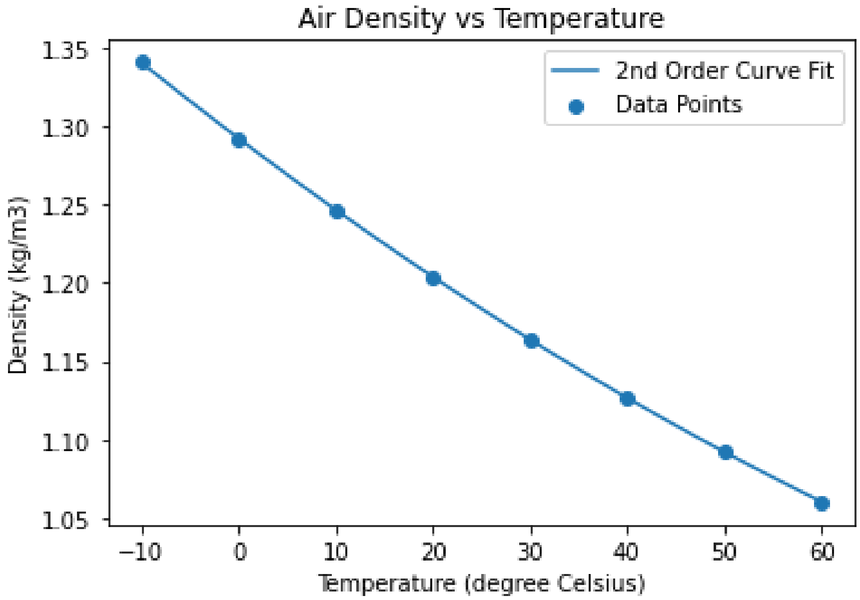


Fig. 17: Air Density vs Temperature with Curve Fit

Similarly, a line of the form aT + b is fit to the viscosity training data. The R^2 and RMSE for the training data are 0.999756 and 1.708e-8 respectively, and those for the testing data are 0.998698 and 1.936e-8 respectively. The equation parameters are shown in Table 12, and the curve fit in Fig. 18.

Table 12 : Constants of Linear Viscosity Fit

a	4.76666667e-08
b	1.71533333e-05

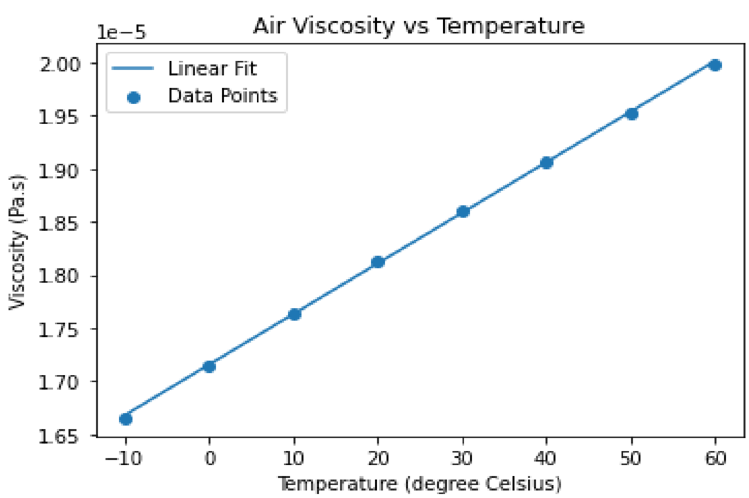


Fig. 18: Air Viscosity vs Temperature with Line Fit

Since the correlations perform very well, they can be used for calculation purposes without any worries (please refer to the Annexure for the complete correlation derivation codes). The last parameter to be considered is the inlet air temperature, which not only appears in the heat transfer equation but also decides the density and viscosity values to be used. This is given by the climatic data for New Delhi.

Now that we have collated all the data, we can proceed to calculations.

4.3.4. Heat Exchanger Calculations

For calculation of various aspects of EAHE operation, a simple Python code was developed, a snippet of which is shown in Fig. 19. Important libraries are imported, arrays for maximum and minimum inlet temperatures and ground temperatures are created, other properties and parameters are defined, mass flow rates are calculated with the help of the density correlation, and the heat transfer equation is implemented to calculate T_{out}. Based on the results for T_{out}, heat duty, effectiveness, COP, etc. can be calculated (snippet shown in Fig. 19, please refer to the Annexure for the complete code).

```python
import pandas as pd
import numpy as np
import math
import matplotlib.pyplot as plt
import matplotlib.ticker as mticker

Tw=np.array([23,20.5,20,21,22,23,25,27.5,28.5,27.5,26.5,25]) # deg C
Tin_min=np.array([8,11,16,23,27,28,28,27,25,21,14,9]) # deg C
Tin_max=np.array([20,24,30,37,40,39,35,34,34,33,28,22]) # deg C

air_flow=0.02643 # m^3/s
Cp=1005 # J kg^-1 K^-1

blower_power=28 # W

pipe_ID=0.15 # m
pipe_length=30.48 # m
area=14.363 # m^2

v=air_flow/(math.pi*pipe_ID**2/4) # m/s
h=2.8+3*v # W m^-2 K^-1

m_dot_min=air_flow*(1.38690477e-05*Tin_min**2-4.69226191e-03*Tin_min+1.29223214e+00)
m_dot_max=air_flow*(1.38690477e-05*Tin_max**2-4.69226191e-03*Tin_max+1.29223214e+00)

Tout_min=Tw+(Tin_min-Tw)*np.exp(-math.pi*pipe_ID*h*pipe_length/(m_dot_min*Cp))
print('Minimum outlet temperatures:')
print(Tout_min)
print('\n')

Tout_max=Tw+(Tin_max-Tw)*np.exp(-math.pi*pipe_ID*h*pipe_length/(m_dot_max*Cp))
print('Maximum outlet temperatures:')
print(Tout_max)
```

Fig. 19: Python Code Snippet

The outlet air temperature (in °C) and exchanger heat duty (in W) results are shown in Table 13. The annual range of outlet air temperature is 19.841-28.678 °C, which is much narrower than the ambient temperature range of 8-40 °C. Further, as can be seen in Fig. 20, the outlet air temperature at any given instant for a particular month will lie within a very narrow band, enabling tight temperature control for all months. The width of this band varies in the range 0.216-0.575 °C. Thus, the EAHE does an impressive job in maintaining thermal comfort in the house throughout the year and offsetting the impact of oppressive weather.

Table 13: Outlet Temperatures and Heat Duties

Month	Ground Temp	Min Temp	Max Temp	Outlet Temp for Min Inlet Temp	Outlet Temp for Max Inlet Temp	Heat Duty for Min Inlet Temp	Heat Duty for Max Inlet Temp
Jan	23	8	20	22.350	22.886	478.574	92.302
Feb	20.5	11	24	20.102	20.627	300.338	106.409
Mar	20	16	30	19.841	20.339	124.557	298.699
Apr	21	23	37	21.073	21.501	60.986	468.337
May	22	27	40	22.175	22.545	150.670	522.398
June	23	28	39	23.173	23.490	150.227	465.673
July	25	28	35	25.104	25.320	90.136	294.396
Aug	27.5	27	34	27.482	27.710	15.067	191.910
Sept	28.5	25	34	28.375	28.678	106.094	162.386
Oct	27.5	21	33	27.256	27.680	199.390	162.856
Nov	26.5	14	28	25.993	26.552	391.601	45.068
Dec	25	9	22	24.314	24.889	508.923	91.752

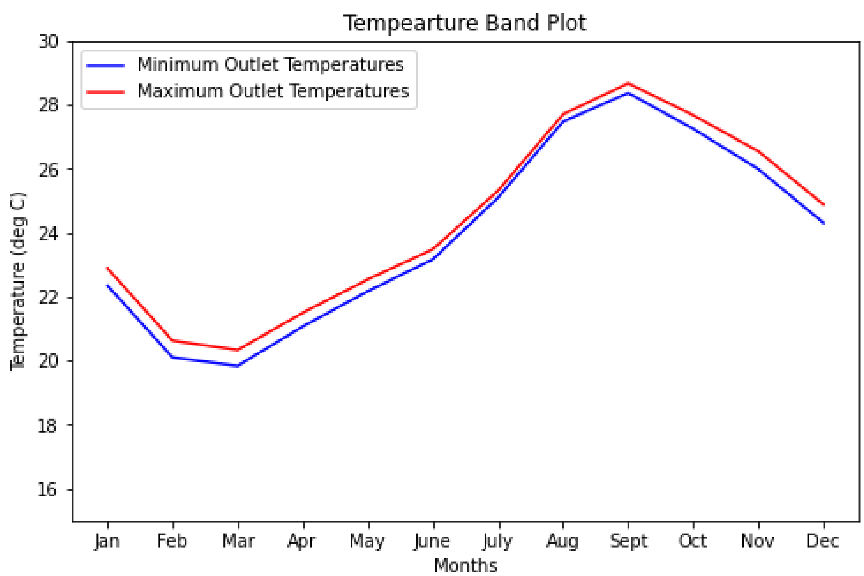


Fig. 20: EAHE Outlet Temperatures vs Month Plot

Pressure drop for the system is given by:

$$P = f\frac{L}{D}\rho\frac{va^2}{2} \qquad (11)$$

Where, f is the friction factor, L is the total length of the tube, D is the diameter of the tube, ρ is the density of the air and va is the velocity of the air.

Friction factor is the measure of the resistance to the fluid flow. It depends on the Reynolds number and varies for laminar and turbulent flow.

$$f = \frac{64}{Re} \qquad \text{(for Laminar flow Re < 2300)} \qquad (12)$$

$$f = (1.82 log(Re) - 1.64)^{-2} \qquad \text{(for turbulent flow Re > 2300)} \qquad (13)$$

Where Reynolds number is given by the following expression:

$$Re = \frac{\rho vaD}{\mu}$$

(14)

Based on the correlations, the following values of air density and viscosity are obtained, which can then be used to calculate the pressure drop. The results are shown in Tables 14 and 15.

Table 14: Viscosity and density data with respect to temperature.

Month	Ground Temp	T avg	Viscosity(Pa.s)	Density(kg/m3)
Jan	23	15.175	0.00001801	1.213
Feb	20.5	15.551	0.00001791	1.222
Mar	20	17.9205	0.00001803	1.212
Apr	21	22.0365	0.00001822	1.196
May	27	21.5875	0.0000182	1.197
June	23	25.5865	0.00001839	1.182
July	25	26.552	0.00001844	1.178
Aug	27.5	27.241	0.00001847	1.175
Sept	28.5	26.6875	0.00001845	1.177
Oct	27.5	24.128	0.00001832	1.187
Nov	26.5	19.9965	0.00001813	1.204
Dec	25	16.657	0.00001797	1.218

Table 15: Pressure drop data with respect to temperature

Month	Reynolds no.	friction factor	Pressure drop
Jan	15109.62909	0.02809	7.743497473

Feb	15306.72697	0.028	7.77595714
Mar	15080.42596	0.02811	7.742622519
Apr	14726.16026	0.02829	7.689334396
May	14754.66923	0.02827	7.69032298
June	14419.24307	0.02844	7.639618813
July	14331.48156	0.02849	7.627151283
Aug	14271.76502	0.02852	7.615738243
Sept	14311.55447	0.0285	7.623351481
Oct	14535.56659	0.02838	7.655749788
Nov	14898.25483	0.0282	7.716142085
Dec	15205.6828	0.02805	7.764344111

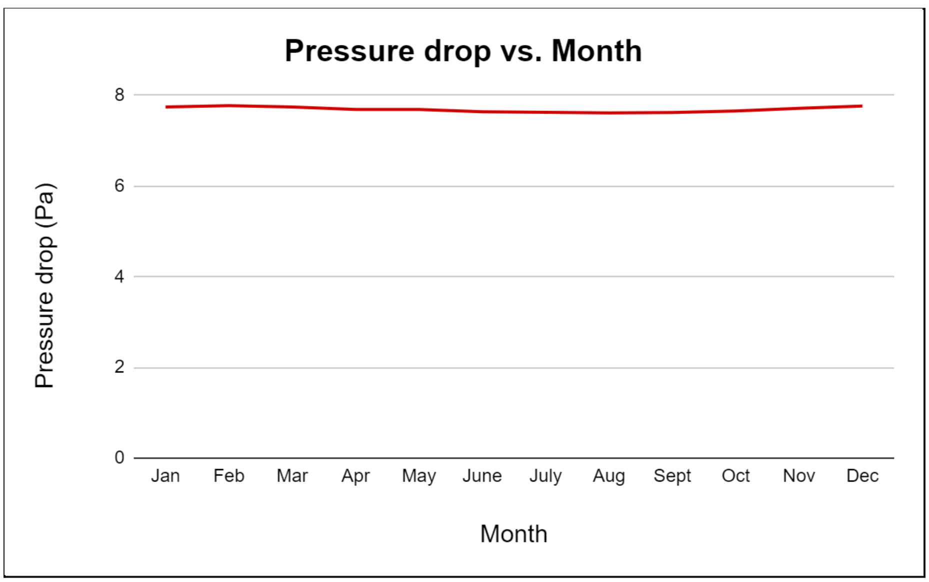


Fig. 21: Pressure drop v/s Month plot

Not only is the EAHE effective in maintaining thermal comfort, but also in efficiently using the available area for heat exchange with the ground, and performing better than conventional heating-cooling systems in terms of energy efficiency. This is indicated by high values of effectiveness, given by:

$$\text{Effectiveness} = (T_{out} - T_{in})/(T_w - T_{in}) \qquad (15)$$

and Coefficient of Performance (COP), given by:

$$COP = Heat\ Duty/Blower\ Power\ Consumption \quad (16)$$

in Table 16. The annual range of effectiveness is 0.957-0.97, with a very high average of 0.964, indicating that the outlet air temperature nearly equals the ground temperature which is the limiting value. The monthly average COP is in the range 3.696-12.019, with an annual average of 8.153. This means that our EAHE is much more energy efficient than most conventional systems having an average COP of 3.5.

Table 16: COP and Effectiveness

Month	COP for Min Inlet Temp	COP for Max Inlet Temp	Average Monthly COP	Effectiveness for Min Inlet Temp	Effectiveness for Max Inlet Temp
Jan	17.092	3.296	10.194	0.957	0.962
Feb	10.726	3.800	7.263	0.958	0.964
Mar	4.448	10.668	7.558	0.960	0.966
Apr	2.178	16.726	9.452	0.963	0.969
May	5.381	18.657	12.019	0.965	0.970
June	5.365	16.631	10.998	0.965	0.969
July	3.219	10.514	6.867	0.965	0.968
Aug	0.538	6.854	3.696	0.965	0.968
Sept	3.789	5.799	4.794	0.964	0.968
Oct	7.121	5.816	6.469	0.963	0.967
Nov	13.986	1.610	7.798	0.959	0.965
Dec	18.176	3.277	10.726	0.957	0.963

From Fig. 22, it can also be seen that the COP is highest for the peak summer and winter months which is desirable as maximum heat exchange with the ground must take place to ensure thermal comfort. In some months, the average COP is much less than in the peak months due to the difference between inlet air and ground temperatures being relatively smaller, thus reducing the scope for heat exchange. However, it is still higher than the average of 3.5 for conventional systems.

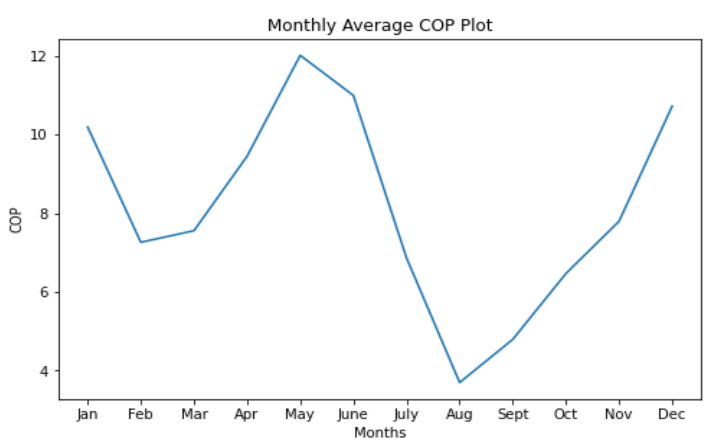


Fig. 22: Average COP for each Month plot

Thus, it can be concluded that the EAHE is effective in maintaining a comfortable environment inside the house and is an environmentally friendly option when compared to other systems due to its high energy efficiency.

From the above analysis for our EAHE system COP was in the range of 0.538-18.657. Analysis by comparing our model with the existing one gave not so surprising results. In an EAHE system used in China where the temperature ranges from -0.5 to 14.3 °C it was found that the COP and Q decrease along with the growth of inlet air temperature but increase along with the pipe length. The COP and Q of EAHE could reach up to the maximum of 15.8 and 2.5 KW respectively when the inlet air temperature is -0.5°C. Whereas the COP decreases to 2.7 and Q to 0.4 KW respectively, when the inlet air temperature is 14.3°C. Within our model's temperature range at a maximum temperature of about 40°C in May the COP was 18.657. This is much higher than the COP from the existing model in China. Whereas for a minimum temperature of about 8°C in January the COP was 17.096. Even though at this temperature which is quite near to 14.3°C the COP is way higher in comparison. Thus we can say we successfully modeled an EAHE system to fulfill our goals.

4.3.5 Cost Analysis

As previously discussed we use HEPA air filters for our EAHE system. The price of the HEPA filter is Rs 6000/- per piece.

Following are the specifications of the air blower we plan to use, based on the discussion in section 4.3.2.

1. Model: MB840-D by Oriental motor
2. Power: 28W
3. Impeller diameter: 3.15 in
4. Cost: Rs 5769.73/-

The model we are proposing is chosen to be at New Delhi in location. Since the system is working with suction pumps and Heat exchanger the electricity costs need to be estimated. The electricity tariffs for New Delhi are:

Table 17: Electricity tariffs for domestic usage in Delhi. Adapted from

Sr. No.	CATEGORY	FIXED CHARGES	ENERGY CHARGES				
1	DOMESTIC						
1.1	INDIVIDUAL CONNECTIONS		0-200 Units	201-400 Units	401-800 Units	801-1200 Units	>1200 Units
A	Upto 2 kW	20 Rs./kW/month	3.00 Rs./kWh	4.50 Rs./kWh	6.50 Rs./kWh	7.00 Rs./kWh	8.00 Rs./kWh
B	> 2kW and ≤ 5 kW	50 Rs./kW/month					
C	> 5kW and ≤ 15 kW	100 Rs./kW/month					
D	>15kW and ≤ 25 kW	200 Rs./kW/month					
E	> 25kW	250 Rs./kW/month					
1.2	Single Point Delivery Supply for GHS	150 Rs./kW/month	4.50 Rs./kWh				

<u>Electricity:</u> (28 W air blower)

Now, the annual electricity consumption for the air blower is 81.76 kWh/year and the cost of grid electricity in India is Rs 3/kWh from table 17. Also, fixed charges per year is Rs 240/kW (Rs 20/kW/month). Therefore the annual electricity cost is given by,

Annual electricity cost = Rs (245.28/year + 23476.8/year) = Rs 23722.08/year

Now, total annual maintenance cost is equal to 10% of total cost which comes out to be Rs 2372.208/year. Hence the total annual cost is calculated as shown.

Total Annual cost = 23770.26 + 2377.026 = Rs 26094.288/year

Now simple payback period is given by,

$$Simple\ payback\ period\ =\ \frac{Initial\ Investment(P)}{Annual\ net\ cash\ flow(CF)} \qquad (17)$$

Where, P is the initial investment

CF = annual uniform cash inflow(R) - Annual uniform cast outflow(M)

Now, P = cost of air filters + cost of air blower + Total pipe cost + Cost of installation. Therefore,

P = 6000 + 5769.73 + 15802.63 + 4823.58 = Rs 32395.94/-

$$\text{Discounted payback period} = \frac{ln[\frac{Cf}{Cf - P*r}]}{ln[1+r]} \qquad (18)$$

Where r = i x (1 - tax benefits)

The suitable annual interest rates considered for EAHE are:

1. 4% interest rate normally offered by government sectors in India.
2. 7–8% is the interest rate normally offered by government banks.
3. 10–12% is the interest rate offered by private banking sectors.
4. 12–16% is the interest rate for any other private source

Using equations (17) and (18) the payback period can be calculated with a knowledge of P, CF and r.

4.4 Ground Source Heat Pump (GSHP)

For providing thermal comfort inside the two storey building in question, we apply a different approach i.e. a ground source heat pump or GSHP system which primarily consists of three units as explained in section 2. We will now look at the design calculations of each unit one by one.

4.4.1 Heat Exchanger Analysis

For the GSHP system, since we are using vertical ground heat exchangers, we have designed a Borehole field with the following specifications:

Table 18: Borehole field specifications

Number of Boreholes	20
Depth of Borehole	3 m
Drilling diameter	120 mm
Shank space	95.325 mm
Distance between U-tube legs	96 mm

For our model, we have taken the Outer diameter 19.065 mm and Inner diameter 15.392 mm of the U-tube heat exchanger. Based on this, we calculated the heat transfer area as follows:

Outer CSA = $\pi * d_o * L = \pi * (19.065 * 10^{(-3)}) * 1 = 0.06\ (m^2)$
Inner CSA = $\pi * d_i * L = 0.048\ (m^2)$

The mass flow rate of water in the pipe is taken as 0.25 kg/s. The velocity of water comes out to be:

$$u = mass\ flow\ rate\ / (p * Inner\ CSA) = 1.346\ \text{m/s}$$

The convective heat transfer coefficient of water is calculated using this expression [10]:

$$h_i = \frac{4200(1.35+0.02t)u^{0.8}}{di^{0.2}} \qquad (19)$$

Where, u_t is the velocity of water in tube, d_i is the inner diameter of tube and t is the tube thickness.

For the sake of simplicity, we've used the ground surface temperature (Table-7) for different months as the vertical heat exchanger's inlet temperature. We have assumed that temperature does not vary with depth at any point below the ground level. Therefore, we directly used the equation (7) derived in section (4.2) to calculate the outlet temperature of the heat exchanger. Using inlet and outlet temperature, we calculated the heat exchanger duty (Q), which is given by:

$$Q = m * C * \Delta T \qquad (20)$$

Table 19: Heat Exchanger Outlet temperature and Heat duties

Month	**L(m)**	**Mass flow (kg/sec)**	**v (m/sec)**	**Tin (deg C)**	**Tout (deg C)**	**Q (KW)**
Jan	6	0.25	1.346	15	22.97	8.327
February	6	0.25	1.346	17	20.99	4.169

March	6	0.25	1.346	26	20.02	-6.248
April	6	0.25	1.346	30	21.03	-9.371
May	6	0.25	1.346	35	22.05	-13.53
June	6	0.25	1.346	38	23.05	-15.619
July	6	0.25	1.346	35	25.03	-10.416
August	6	0.25	1.346	32	26.02	-6.248
September	6	0.25	1.346	28	27	-1.045
October	6	0.25	1.346	25	26.99	2.079
November	6	0.25	1.346	22	25.99	4.169
December	6	0.25	1.346	19	24.98	6.248

Now we can move ahead with the calculations of the heat pump unit.

4.4.2 Heat Pump Analysis

For calculating the power requirement to the compressor in the heat pump we developed a DWSIM model consisting of the main four subcomponents namely evaporator compressor, condenser and expansion valve. The DWSIM model is shown below.

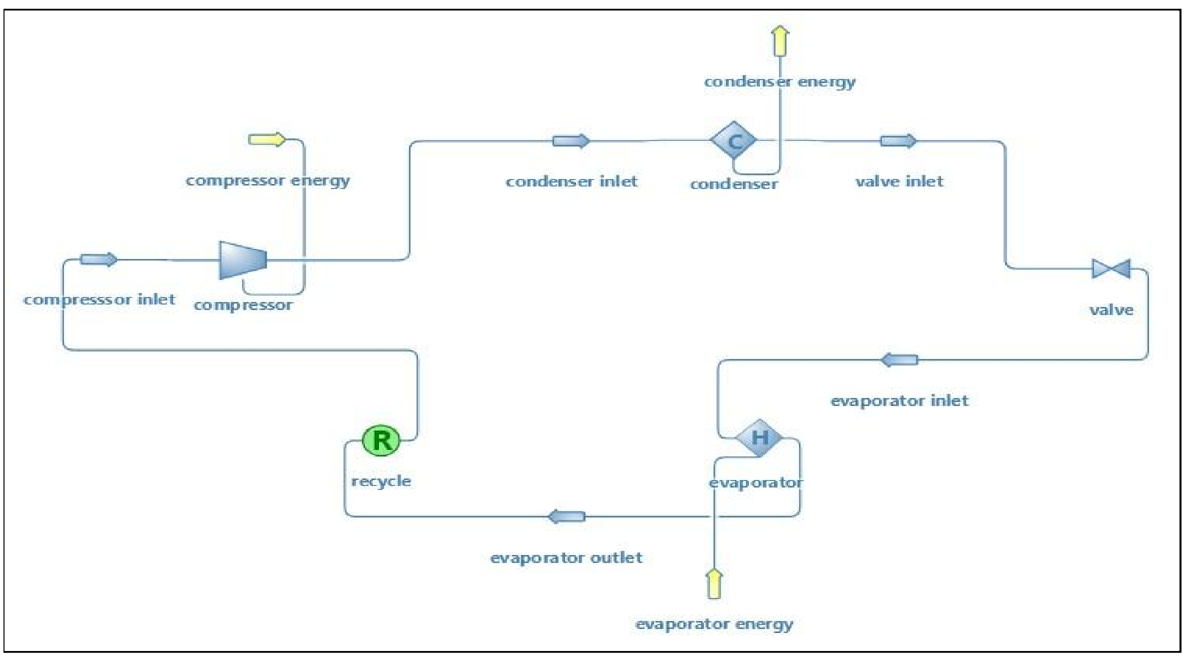


Fig. 23: DWSIM model of heat pump for GSHP system

Taking a closer look at the simulation we can see that energy is fed into the evaporator and compressor and is released from the condenser. The energy input to the evaporator comes from the heat exchanger unit, i.e., the heat load of each month that we have calculated in section 4.4.1. We assume there is no heat loss and heat is transferred directly from the circulating fluid of the heat exchanger (water) to the refrigerant in the heat pump.

In this DWSIM model we have carefully chosen propane as the refrigerant which has zero ozone depletion potential and negligible global warming potential. We have assumed 100% efficiency of the evaporator and taken pressure drop as 7 Pa (after much research on maximum allowable pressure drop from literary sources). For the compressor we assumed adiabatic process and 100% adiabatic efficiency.

The energy input to the evaporator was taken as the heat duty for the heat exchanger unit as calculated in section 4.4.1. Coefficient of performance is calculated using equation (1) as Qcompressor/Qcondensor where Qcomp is the power that needs to be supplied to the compressor and Qcondensor is the output power of the heat pump.

Master property table for the month of January is shown below.

Tabel 20: DWSIM Model Master Property Table for January

Master Property Table				
Object	evaporator energy	condenser energy	compressor energy	
Energy Flow	8.327	11.7632	3.43619	kW

The results obtained for each month are given in the table below.

Table 21: DWSIM Heat Pump Model Results

Month	Qevaporator (KW)	Qcompressor (KW)	Qcondensor (KW)	COP
January	8.327	3.43619	11.7632	3.42
February	4.169	3.37861	7.54761	2.23
March	-8.327	3.28062	-5.04638	1.54
April	-9.371	3.27281	-6.09819	1.86
May	-13.53	3.24169	-10.2883	3.17
June	-15.619	3.22605	-12.3929	3.84
July	-10.416	3.26499	-7.15101	2.19
August	-7.292	3.28837	-4.00363	1.22
September	-6.248	3.29618	-2.95182	0.9
October	2.079	3.36523	5.44423	1.62
November	4.169	3.37861	7.54761	2.23
December	6.248	3.41511	9.66311	2.83

The obtained results are shown graphically for better visualization.

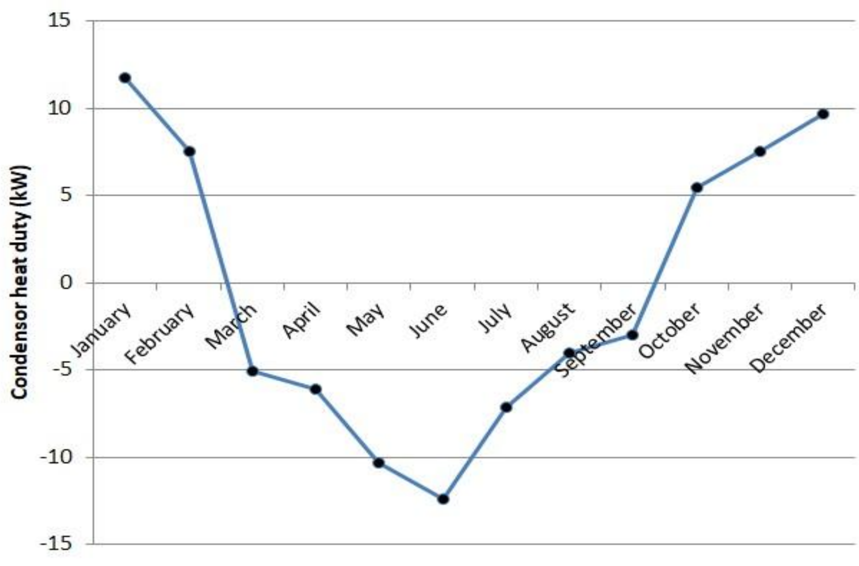


Fig. 24: Condenser heat duty vs. month plot

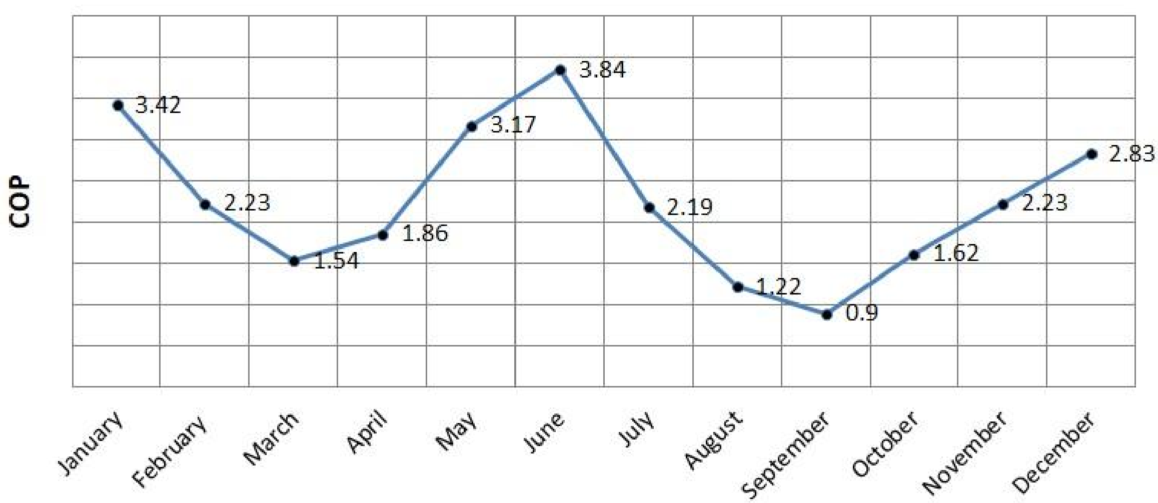


Fig. 25: COP vs. month plot for GSHP

The sign of heat duty of the condenser obtained from the simulation model can easily be analyzed. For winter months like January, February, October,November and December, Qevaporator is positive meaning that heat is supplied to the heat pump, Qcompressor is positive indicating that power needs to be supplied to the heat pump, and Qcondensor is positive which means that energy is being released by the heat pump. This heat can further be used to heat up the circulating fluid in the secondary loop which can further be transferred to the building. For the remaining months, Qcondensor is negative which means that heat is being extracted from the circulating fluid in the secondary loop by the heat pump which can ultimately be used for cooling.Hence, the results are as required i.e, heating in winter months and cooling in summer months. The COP of each month is also satisfactory with monthly COPs lying in the range of 0.9 -3.8 and average annual COP of 2.25. Hence we can move forward with our calculations.

4.4.3 Radiant Heating/Cooling Unit Analysis

Now that we have completed the calculations of the heat exchanger unit and heat pump unit we are left with the calculation for the secondary loop or building loop consisting of a radiant heating/ cooling unit. For this we have to consider heat transfer by convection and radiation both. For simplicity we will be using the concept of combined heat transfer coefficient as recommended by ISO 11855. The combined heat transfer coefficient as given in ASHRAE HVAC Systems and Equipment (2012) for floor/wall/ceiling heating/cooling is 8.29 W/m^2 K. If we multiply equation (2) by the area of the radiant surface on both sides we can calculate the total heat transferred by the radiant system.

Now, in this case we assume that all the energy being supplied by the condenser of the heat pump is absorbed by the circulating fluid in the secondary loop (water) and there are no power losses. For the sake of simplicity, we again assume that the energy being supplied to the circulating fluid in the secondary loop is transferred to the interior of the two story house via the radiant heating/ cooling system. With these assumptions we can say that the total heat transferred by the radiant surface is equal to the power output of the condenser and by dividing by the combined heat transfer coefficient and radiant surface area we can obtain the temperature difference between the walls and room temperature.

For calculating the total radiant surface temperature we assume a piping network in the ceiling and two walls opposite to each other on both floors. The building has a square base of 300ft^2(27.87 m^2) making the dimensions of the base 17.32 ft × 17.32 ft (5.28m × 5.28m). The total height of the building is 7.5m making the height of each floor 3.75m. Hence the total radiant surface area of the entire building is given by,

$$A = (27.87\ m^2 \times\ 2\ ) + (\ 5.28\ m \times 3.75m) \times 4 = 134.94\ m^2$$

The results after dividing total heat transferred by radiant surface area and combined heat transfer coefficients (as described above) for each month are shown in the table below.

Table 22: Difference between wall surface temperature and room temperature

Month	|Qcondensor| (KW)	h_{tot} (W/m^2 K)	A(m^2)	|ΔT|
January	11.7632	8.29	134.94	10.52
February	7.54761	8.29	134.94	6.75
March	5.04638	8.29	134.94	4.51
April	6.09819	8.29	134.94	5.45
May	10.2883	8.29	134.94	9.2
June	12.3929	8.29	134.94	11.08
July	7.15101	8.29	134.94	6.39
August	4.00363	8.29	134.94	3.58
September	2.95182	8.29	134.94	2.64
October	5.44423	8.29	134.94	4.87
November	7.54761	8.29	134.94	6.75
December	9.66311	8.29	134.94	8.64

The temperature differences obtained give a rough idea of how much heating or cooling takes place. The temperature differences are appreciable and we can say that the modeled GSHP system can be used to provide thermal comfort in the two story building to an appreciable extent.

4.4.4 Electricity Cost Analysis

From section 4.3.5, Cost of grid electricity in India = Rs 3/kWh and annual electricity consumption of the GSHP system = 29084.85072 kWh/year

Therefore, Annual electricity cost = Rs (3*29084.85072) = Rs 87254.55 /year

Table 23: Heat Pump Electricity consumption

Month	Qcomp(KW)	time	total power
Jan	3.43619	744	2556.52536
February	3.37861	672	2270.42592
March	3.28062	744	2440.78128
April	3.27281	720	2356.4232
May	3.24169	744	2411.81736
June	3.22605	720	2322.756
July	3.26499	744	2429.15256
August	3.28837	744	2446.54728
September	3.29618	720	2373.2496

October	3.36523	744	2503.73112
November	3.37861	720	2432.5992
December	3.41511	744	2540.84184
Qtotal	39.84446		29084.85072

5. Conclusion

We have developed two heating and cooling systems: EAHE and GSHP. Both these Earth integrated systems are viable techniques for preheating the air/water in winter and cooling it in summer. We must calculate the building's heating/cooling load requirement, geometrical constraints, and perform cost analysis in order to design these systems. In the EAHE, air is the heating/cooling medium and in the GSHP, heating/cooling medium. Both the systems are quite energy efficient compared to conventional heating-cooling systems, as indicated by their high COP values. They are able to bring significant temperature differences and we can therefore say that our modeled systems provide thermal comfort in the building. Simulation results show that GSHP can provide 11.08 °C indoor temperature reduction, for the summer peak month (June) in New Delhi. For the same month, from our calculations, the EAHE has the potential for reducing the indoor temperature by 15.51°C. The EAHE also has a higher average COP of 8.153 as compared to 2.25 for the GSHP. Thus, it can be concluded that the EAHE is more efficient than the GSHP system.

6. Future Scope

This study is entirely based on a theoretical analysis of the two heat exchanger systems. It could be taken a step further by relaxing the assumption of constant ground temperature for a given month and taking into account its variability as a result of heat exchange with the working fluid and other natural phenomena at work. Currently, the benefits and performance of solar panels have been analyzed, however, we would like to explore their proper integration with our systems to determine their viability and how much of the electrical energy demanded by the system equipment can be fulfilled. Also, in this study, the determination of heating and cooling loads and heat exchanger performance evaluation have been treated as separate problems. So, we would like to combine the two into a comprehensive whole with the help of a suitable simulation software to get a theoretical understanding of the thermal conditions which would prevail indoors when the heat exchangers are in action. Lastly, a good way to advance the study would be to implement the systems in a real setting to gain better insights, see how our assumptions hold up and how fluctuations in prevalent conditions influence the operation of our systems.

We can also implement the integration of solar panels in the designing of a system of houses called R-2000 houses. Houses which are made on R-2000 technology include features like upgraded insulation, roofing and vapor barrier system that make it 30% more efficient in comparison to a normal house. Non-crystalline panels, which is a Thin-Film solar panel, can be the future of the solar industry because they are not completely made with silicon so they are economical or easy to manufacture.

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8. Annexure

8.1. Heat Transfer Governing Equation Derivation

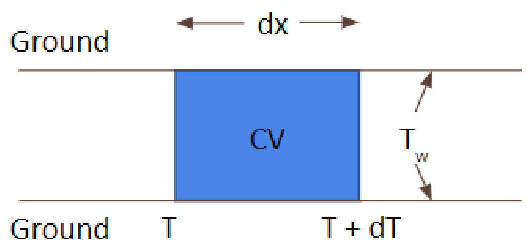

Fig. 26: Horizontal Heat exchanger Pipe-cross section

Energy balance on control volume (CV) gives:

$$\dot{m}C_P \frac{T+dT-T}{dx} = 2\pi rh\, dx(T_w - T)$$

$$\frac{dT}{T_w - T} = \frac{2\pi rh\, dx}{\dot{m}C_P}$$

On integrating, we get:

$$-\ ln(T_w - T) = \frac{2\pi rh\, dx}{\dot{m}C_P} + C$$

Boundary Condition: T = T_{in} at x = 0, hence C = $-\ln(T_w - T_{in})$

On substituting, we get:

$$ln\left(\frac{T_w - T}{T_w - T_{in}}\right) = -\frac{2\pi rh\, dx}{\dot{m}C_P}$$

$$\frac{T - T_w}{T_{in} - T_w} = exp\left(-\frac{2\pi rh\, dx}{\dot{m}C_P}\right)$$

On rearranging, we get the final equation:

$$T_{out} = T_w + (T_{in} - T_w)e^{-\frac{2\pi rhx}{\dot{m}C_P}}$$

8.2. Codes for Air Density and Viscosity Correlation Derivation

Google Drive link to the Python code file for air density correlation:

https://drive.google.com/file/d/1q0qfx3Wt3mjQGgcZJ2fLeJAX4y5-dAd1/view?usp=sharing

Google Drive link to the Python code file for air viscosity correlation:
https://drive.google.com/file/d/1HTJh1VWCcVUxDsAafVIGGvnYR6gwycSP/view?usp=sharing

8.3. Code for Heat Exchanger Calculations

Google Drive link to the Python code file:
https://drive.google.com/file/d/1-mlIRDo72sjFQv9SNbFaqfg75b7bMEqO/view?usp=sharing

8.3. Code for DWSIM Model

Google Drive link:
https://drive.google.com/file/d/14LQJ07JDVyA0sJ_4LuoMYqm5XT_kLHcY/view?usp=sharing