Ain Shams University
Faculty of Engineering
Mechanical Power Engineering Department
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GRADUATION PROJECT 2022-2023

Deploying Egypt's shallow geothermal potential for sustainable cooling and air conditioning in real estate developments

(Pilot Study)

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First and foremost, I always feel indebted to *ALLAH,*
The Most Kind and Most Merciful

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Summary:

The purpose of this project is to gather and analyse data on the potential for implementing sustainable air conditioning and cooling systems in Egypt using shallow geothermal energy. The report will examine various applications and discuss the methodology used for each application, while also assessing the environmental impact of geothermal cooling-heating systems based on the manufacturing of each component of ground source heat pump and assessing the electrical consumption of the unit.

The project aims to promote the use of heat exchanger systems in commercial and small industrial facilities in Egypt. For the considered climatic zones and for the different energy efficiency measures, primary energy savings, pay back periods and CO_2 emissions are evaluated as ground source heat pump (GSHP) systems, due to their high coefficient of performance (COP) and low CO_2 emissions are great substitute for fossil fuel to provide more comfortable coexistence of humans and the environment.

A dynamic simulation study in COMSOL environment has been conducted to build a mathematical model of Ground-source Heat exchangers to discuss the thermodynamic analysis of Cairo University case study as well as a semi-finite sink of ASU GEO-Cooling prototype through a steady state, and a transient solution, and investigate the feasibility of its practical development.

The report will also provide an overview of geothermal heating and cooling systems and discuss the importance of ground heat exchangers.

Introduction:

Geothermal energy can be used for different purposes, depending on the operating temperature of the source. Usually, in case of low and medium-enthalpy sources, geothermal energy is used directly, for space heating, domestic hot water, agricultural uses, etc. [1]. Conversely, in case of high-enthalpy geothermal resources, heat is more profitably converted into electricity [2,3] or in more complex cascade cycles [4]. Low or medium temperature geothermal energy can be converted into electric power only when innovative and expensive system layouts are considered [3,5]. Unfortunately, at a reasonable depth only low-enthalpy geothermal energy is usually available; only in some specific locations in the world, high-enthalpy geothermal resources are available even using low-depth wells [6].

1.1. Motivation

- ***Egypt's need for renewable energy in the HVAC sector:***

It has been reported that Egypt has embarked on a vision to encourage and support renewable energy-related industries in the Middle East and North Africa (MENA) mainly to address issues related to environment resulting due to large CO2 emissions. For example, a case study done for a Petrojet Company New Head Office Building in Cairo to assess the energy efficiency was found that about 64% of the electrical monthly calculated consumption of the building is due to conventional HVAC systems which represents about 180,000 (KWH/month).

Figure 2: Petro-jet Company new head administrative office building

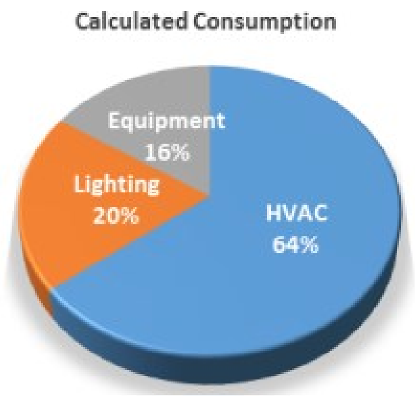


Figure 1: Energy use breakdown percentage

Buildings are responsible for one-third of world greenhouse gas (GHG) emissions (Robert & Kummert, 2012). Fossil fuels are largely responsible for GHG emissions' growth, which is leading to global warming, climate change, and environmental impacts (Liang, Wu, Lal, & Guo, 2013; Lising, 2012; Nejat, Jomehzadeh, Taheri, Gohari, & Majid, 2015; Pan & Garmston, 2012)

For the above-mentioned reasons, most of the studies on space heating and cooling systems driven by geothermal energy focus on Geothermal Heat Pumps (GHPs) and that such systems are going to become more and more profitable in the near future. Therefore, in this report, the motivation of the work done is based on the possibility of researching the geothermal heat potential in Egypt for space cooling purposes, to minimize the electrical consumption and maximize the economic performance of the system.

1.2. Background on Geothermal heating and cooling systems.

The economic viability of using geothermal energy is influenced by location and resources, initial expenses, discount rate, system efficiency, annual load, and demand, etc. (Gudmundsson & Lund, 1985). Yet, the substantial environmental and reliability advantages of geothermal energy over other energy sources must not be ignored (Rosen & Koohi-Fayegh, 2017).

For ground source heat pump systems, there are two basic cycles:

• ***Heating in cold seasons***:

Since ground temperature (T_g) is higher than atmospheric air temperature (T_a), ($T_g > T_a$), and T_g may be sufficient for heating or only preheating to conventional heating systems, based on the efficiency of the heat pump system to extract heat from the ground (Rosen & Koohi-Fayegh, 2017).

• ***Cooling in hot seasons***:

Now, $T_g < T_a$, and the ΔT enables cooling or precooling, enhanced by cooling mode heat pump operation to achieve greater efficiency (Rosen & Koohi-Fayegh, 2017). Geothermal-based heating and cooling systems consist of a heat pump, a ground heat exchanger (GHE) installed underground, and an air distribution system (Rosen & Koohi-Fayegh, 2017). The major cost depends on the ground-based heat exchanger, which must be sized depending on demand expectations and ancillary systems (e.g., a natural gas component for extremely cold temperatures). GHEs can be vulnerable to subsurface flow rates in permeable cases, as well as ground temperature, thermal properties of soil, and heat exchange coefficients, but can be designed optimally for a range of conditions (Rosen & Koohi-Fayegh, 2017). Fig. 3 shows that when geothermal energy is employed in a HVAC system, there is a potential of reducing the energy bill by half.

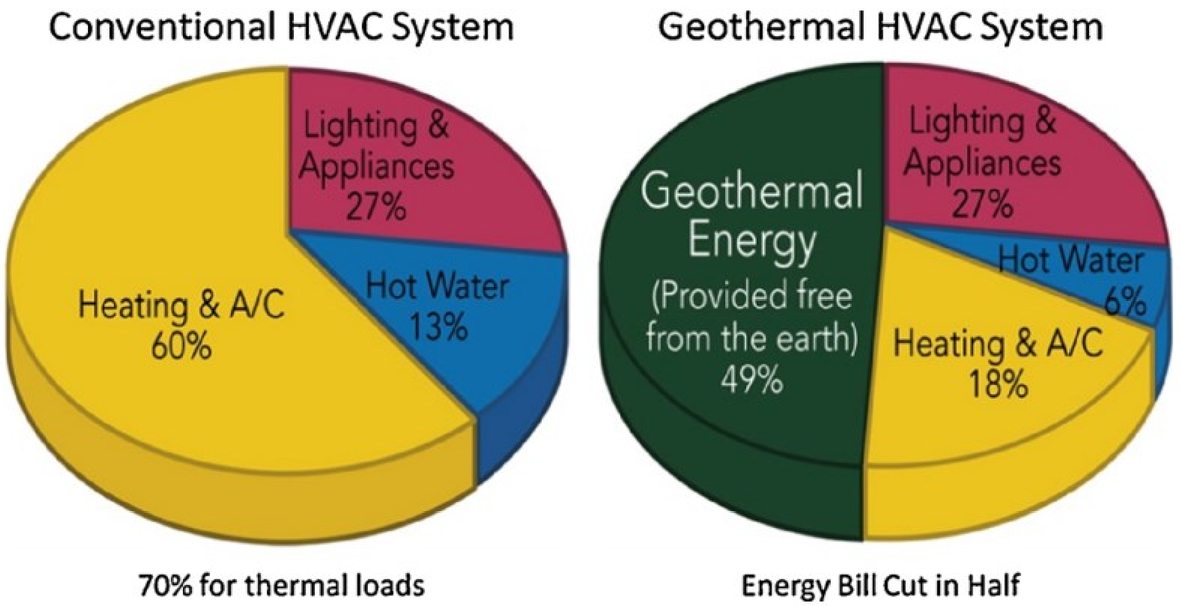


Figure 3: Geothermal versus conventional HVAC system (Anonymous, 2018b)

1.3. Aim and Objective

The objective of this feasibility study is to assess the technical feasibility, economic viability, and environmental impact of implementing ground source heat pump systems for cooling in various case studies in Egypt.

Additionally, the study aims to analyse the potential energy savings and financial benefits of integrating photovoltaic systems with ground source heat pump cooling systems. The study will provide recommendations and guidelines for the implementation of these systems in commercial and small industrial facilities in Egypt, with the goal of promoting their adoption and reducing reliance on fossil fuels.

By achieving these objectives, the study will contribute to the development of sustainable and renewable energy solutions for cooling and heating purposes, ultimately leading to a more environmentally friendly and energy-efficient future for Egypt.

1.4. Ground-source heat pump (GSHP)

All heat pumps function on the basis of a temperature difference (ΔT): the low-T medium is the heat source (TL) and the high-T medium is the heat sink (TH). A GSHP uses the ground as its heat source or sink (Rosen & Koohi-Fayegh, 2017), depending on the season (Deng, Feng, Fang, & Cao, 2018), and the GHE design mediates the heat exchange efficiency. A heating or cooling coil (air-based heat exchanger) mediates heat exchange between the heat pump and the space to be heated (e.g., through forced air circulation). When cooling (inserting heat into the ground), heat is exchanged from the cooling coil (low-temperature medium) to the refrigerant flowing in the GHE (high-temperature medium); when heating (heat removal from the ground), heat is exchanged with the refrigerant flowing in the GHE (low-temperature medium) to the heating coil (high-temperature medium) (Rosen & Koohi-Fayegh, 2017).

In this report, closed loop GSHP is the focus for the designing and implementation phases and in the methodology, different types of closed loop systems will be discussed.

Methodology:

The following Figure.4 shows the phase of the project for each case study:

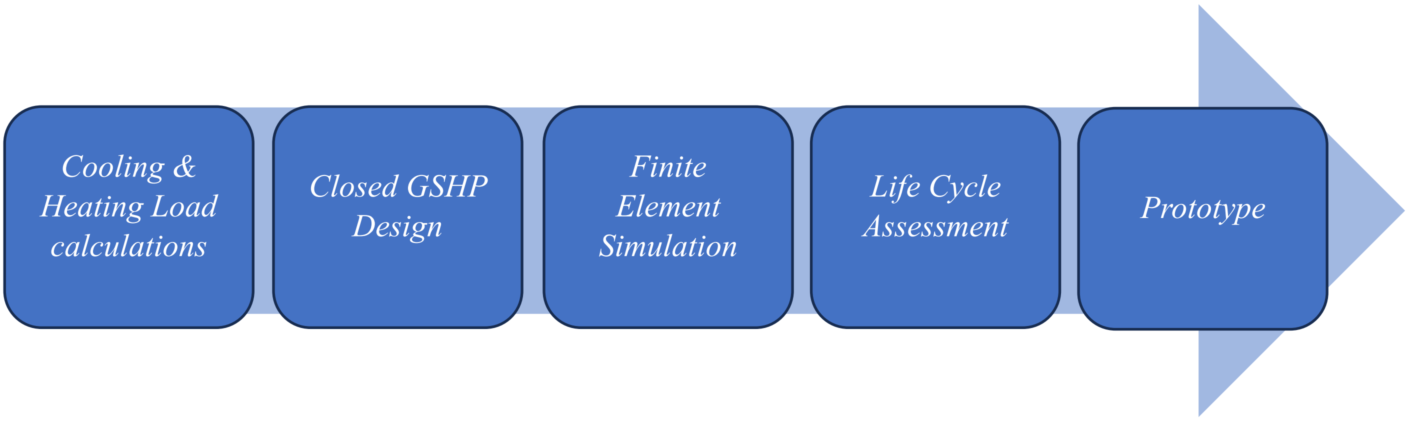


Figure 4: Flow chart of the project phases

2.1 Cooling & Heating Load Calculations:

2.1.1 Operation Theory

In the selection of cooling system, capacity needs to be determined according to building cooling load. That cooling load is not calculated in the reliability way cause selection of cooling system that is not suitable to building, increases cooling system cost and aggravation of indoor environment comfort conditions.

Every building needs a specific amount of cooling to be comfortable, and obtaining this level of comfort is dependent on having the right-sized whole building air conditioning unit. Defining several factors for calculating the building cooling loads, including:

- **Daytime heat gain** – Building gains a certain amount of thermal heat daily.
- **Building Orientation** - the direction in which your building faces play a large role in daytime heat gain.
- **Levels of insulation from top to bottom** - insulation plays an important role in stopping heat transfer, so it’s critical to know how much and what type(s) of insulation you have.
- **Floor plan** - an open floor plan will conduct cool air very differently from one that has many closed rooms and walls.
- **Number and types of windows and doors** - insulated windows and doors have a large impact on retaining the cool air in home.
- **Number of occupants** - people generate heat, which will affect the cooling in home.
- **Square footage** - size relates to the amount of space your cooling needs to adequately cover.

2.1.2 Analytical Procedure

Cooling load is calculated to select HVAC equipment that has the appropriate cooling capacity to remove heat from the zone. A zone is typically defined as an area with similar heat gain, similar temperature and humidity control requirements, or an enclosed space within a building with the purpose to monitor and control the zone's temperature and humidity with a single sensor e.g., thermostat [7]. There is utmost 4 ways for energy (heat) transfer in a zone, these include:

- Solar transference (due to temperature difference);
- Air change load (infiltration or exfiltration);
- Machine load (heat dissipation via equipment's);
- From living organisms.

Cooling load calculation methodologies consider heat transfer by conduction, convection, and radiation. Methodologies include heat balance, radiant time series [8] cooling load temperature difference, transfer function [9] and soil-air temperature. Methods calculate the cooling load in either steady state or dynamic conditions and some can be more involved than others. These methodologies and others can be found in ASRAE handbooks, ISO Standard 11855, European Standard (EN) 15243, and EN 15255 [10]. ASHRAE recommends the heat balance method and radiant time series methods. Differentiation from heat gains. The cooling load of a building should not be confused with its heat gain. Heat gain refer to the rate at which heat is transferred into or generated inside a building. Just like cooling loads, heat gain can be separated into sensible and latent heat gains that can occur through conduction, convection, and radiation. Thermophysical properties of walls, floors, ceilings, and windows, lighting power density (LPD), plug load density, occupant density, and equipment efficiency play an important role in determining the magnitude of heat gains in a building. ASHRAE handbook of fundamentals refers to the following six modes of entry for heat gains [11]:

- Solar radiation through transparent surfaces.
- Heat conduction through exterior walls and roofs.
- Heat conduction through ceilings, floors, and interior partitions.
- Heat generated in the space by occupants, lights, and appliances.
- Energy transfer through direct-with-space ventilation and infiltration of outdoor air.
- Miscellaneous heats gains.

Furthermore, heat extraction rate is the rate at which heat is being removed from the space by the cooling equipment [12]. Heat gains, heat extraction rate, and cooling loads values are often not equal due to thermal inertia effects. Heat is stored in the mass of the building and furnishings delaying the time at which it can become a heat gain and be extracted by the cooling equipment to maintain the desired indoor conditions. Another reason is inability of the cooling system to keep dry bulb temperature and humidity constant.

Design of cooling loads assume steady periodic conditions (i.e., the design day's weather, occupancy, and heat gain conditions are identical to those for preceding days such that the loads repeat on an identical 24 h cyclical basis). Thus, the heat gain for a particular component at a particular hour is the same as 24 h prior, which is the same as 48 h prior, etc. [12].

After collecting all the cooling load components for each space, the heating load is calculated. Psychrometric calculations use thermodynamic properties to analyse conditions and processes involving moist air. By calculating these various saturations, we can determine the necessary airflows, entering and leaving air temperatures, and equipment loads of the zone. Once the individual space loads, psychrometric, and equipment loads are all calculated, the engine determines the final block loads of the zones, levels, and building [10].

Ensuring that you install the right size furnace, air conditioner, heat pump or complete HVAC system is critical to providing consistent indoor comfort for you and your family. It's also important for saving energy, which can be wasted by either a too-large or too-small system. Since energy savings translate to monetary savings, in today's economy you simply can't afford to ignore system sizing when selecting your new HVAC equipment.

2.1.3 HAP Software:

The calculation of space heat load using the transfer function method (TFM) consists of two steps. First, heat loss from exterior walls, roofs, and floors is calculated using conduction transfer function coefficients; and the solar and internal heat gains are calculated directly for the scheduled hour. Second, room transfer function coefficients are used to convert the heat gains to cooling loads, or heat losses to heating loads. Most of the widely adopted computer software programs for space load calculations, HAP software in our case, are based on the TFM.

Hourly Analysis Program (HAP) is a computer tool produced by Carrier, a company providing solutions for air conditioning, heating, and refrigeration. The aim of this program is to assist engineers in designing HVAC systems for commercial buildings. It presents two tools in one: estimation of the loads and designing system, and simulation of the energy use and calculation of energy costs. The program is thus split into two parts: HAP system design features and HAP Energy Analysis Features (HAP Carrier 2005).

In the first part, HAP can perform the following tasks:

- To calculate design cooling and heating loads for spaces, zones, and coils.
- To determine the required airflow rates for spaces, zones, and system.
- To size cooling and heating coils.
- To size air circulation fans.
- To size chillers and boilers.

During the energy analysis, HAP executes the following tasks:

- To simulate an hour-by-hour operation of all heating and air conditioning systems.
- To simulate an hour-by-hour operation of all plant equipment.
- To simulate an hour-by-hour operation of non-HVAC systems.
- To calculate the total energy use and energy costs based on the previous simulations.

To generate tabular and graphical reports of hourly, daily, monthly, and annual data. [13]

2.2 Closed Loop Ground Source Heat Pump Design:

2.2.1 Vertical vs Horizontal GSHP:

Design of ground heat exchangers is complicated by the variety of geological formations and properties that affect thermal performance. Proper identification of materials, moisture content, water movement is an involved process and cannot be economically justified for every project.

For constructing a closed loop ground source heat pump for any building, there are several factors that affect the selection for the orientation of GSHP, these factors are:

I. Economic Evaluation
II. System Performance
III. Accessed Area for digging and implementation.

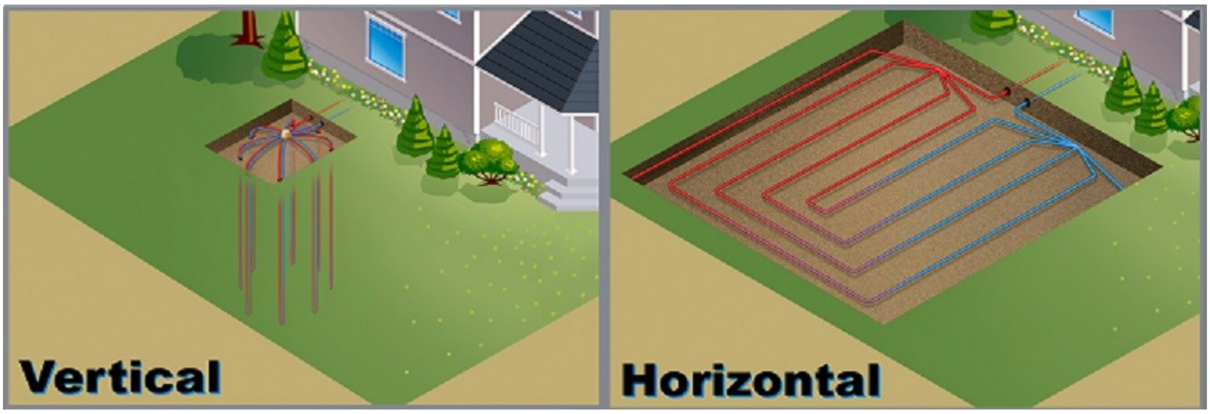


Figure 5: Vertical vs Horizontal GSHP setup

In this report, the design will be constructed using:

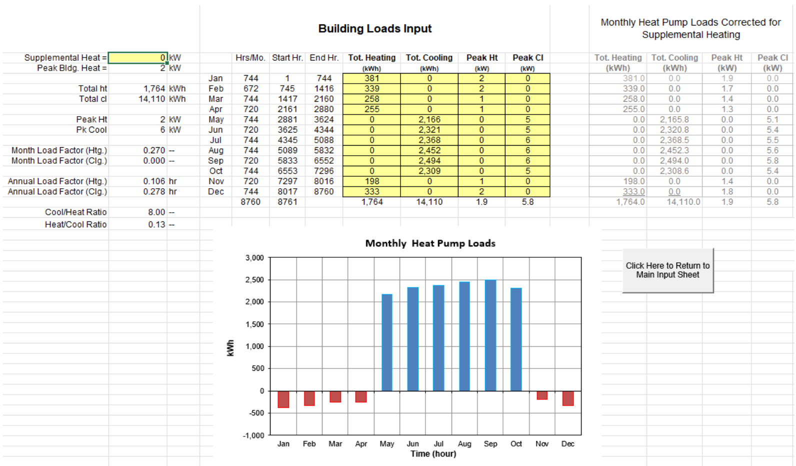
a. GHX_Design_Toolbox to check the ground response on the underground system.
b. ASHRAE Manual book to validate GHX_Design_Toolbox software by addressing the heat transfer governing equations

2.2.2 GHX_Design_Toolbox:

With the collaboration of GEB (Geothermal Energy Capacity Building in Egypt) co-funded by the Erasmus + Programme of the European Union, GHX_Design_Toolbox software has been given an access to Ain Shams university to design and implement the closed loop systems for every application written in this report.

2.2.2.1 Procedure in GHX_Design_Toolbox software:

First, cooling and heating loads are entered as an input data before defining the closed loop system parameters as shown in Figure. 6



Supplemental Heat =	0	kW
Peak Bldg. Heat =	2	kW
Total ht	1,764	kWh
Total cl	14,110	kWh
Peak Ht	2	kW
Pk Cool	6	kW
Month Load Factor (Htg.)	0.270	--
Month Load Factor (Clg.)	0.000	--
Annual Load Factor (Htg.)	0.106	hr
Annual Load Factor (Clg.)	0.278	hr
Cool/Heat Ratio	8.00	--
Heat/Cool Ratio	0.13	--

Building Loads Input / Monthly Heat Pump Loads Corrected for Supplemental Heating

	Hrs/Mo.	Start Hr.	End Hr.	Tot. Heating (kWh)	Tot. Cooling (kWh)	Peak Ht (kW)	Peak Cl (kW)	Tot. Heating (kWh)	Tot. Cooling (kWh)	Peak Ht (kW)	Peak Cl (kW)
Jan	744	1	744	381	0	2	0	381.0	0.0	1.9	0.0
Feb	672	745	1416	339	0	2	0	339.0	0.0	1.7	0.0
Mar	744	1417	2160	258	0	1	0	258.0	0.0	1.4	0.0
Apr	720	2161	2880	255	0	1	0	255.0	0.0	1.3	0.0
May	744	2881	3624	0	2,166	0	5	0.0	2,165.8	0.0	5.1
Jun	720	3625	4344	0	2,321	0	5	0.0	2,320.8	0.0	5.4
Jul	744	4345	5088	0	2,368	0	6	0.0	2,368.5	0.0	5.5
Aug	744	5089	5832	0	2,452	0	6	0.0	2,452.3	0.0	5.6
Sep	720	5833	6552	0	2,494	0	6	0.0	2,494.0	0.0	5.8
Oct	744	6553	7296	0	2,309	0	5	0.0	2,308.6	0.0	5.4
Nov	720	7297	8016	198	0	1	0	198.0	0.0	1.4	0.0
Dec	744	8017	8760	333	0	2	0	333.0	0.0	1.8	0.0
	8760	8761		1,764	14,110	1.9	5.8	1,764.0	14,110.0	1.9	5.8

Figure 6: Loads Input data in GHX_Design_toolbox

Second, according to the selection of closed loop GSHP, soil parameters as well as other parameters are being defined.

I. For Vertical GSHP:

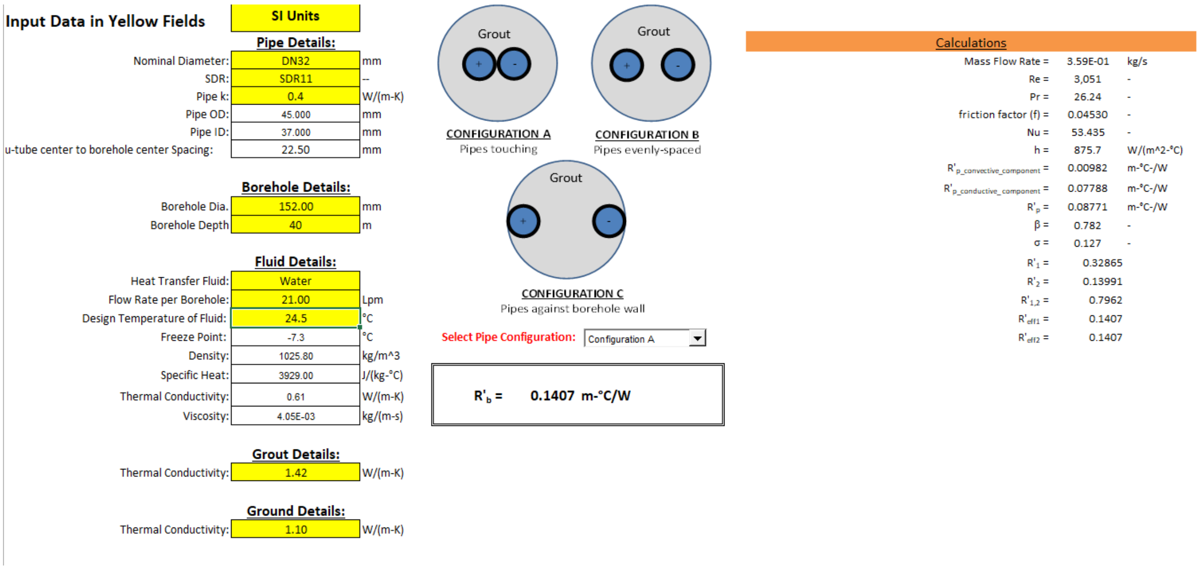


Figure 7: Pipe and Grout Parameters Input Data

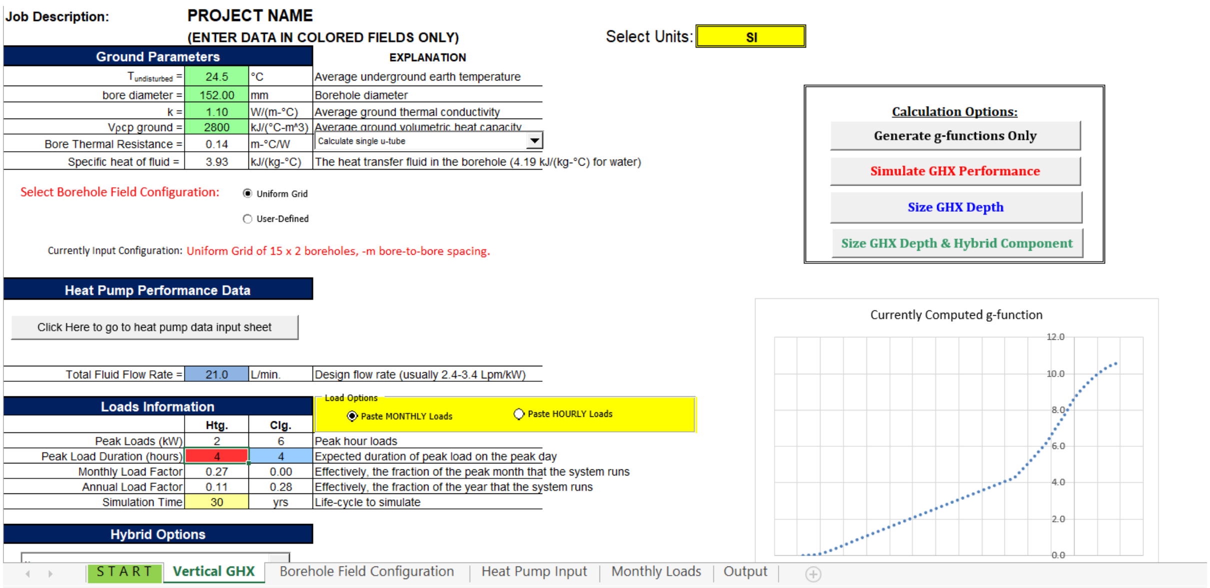
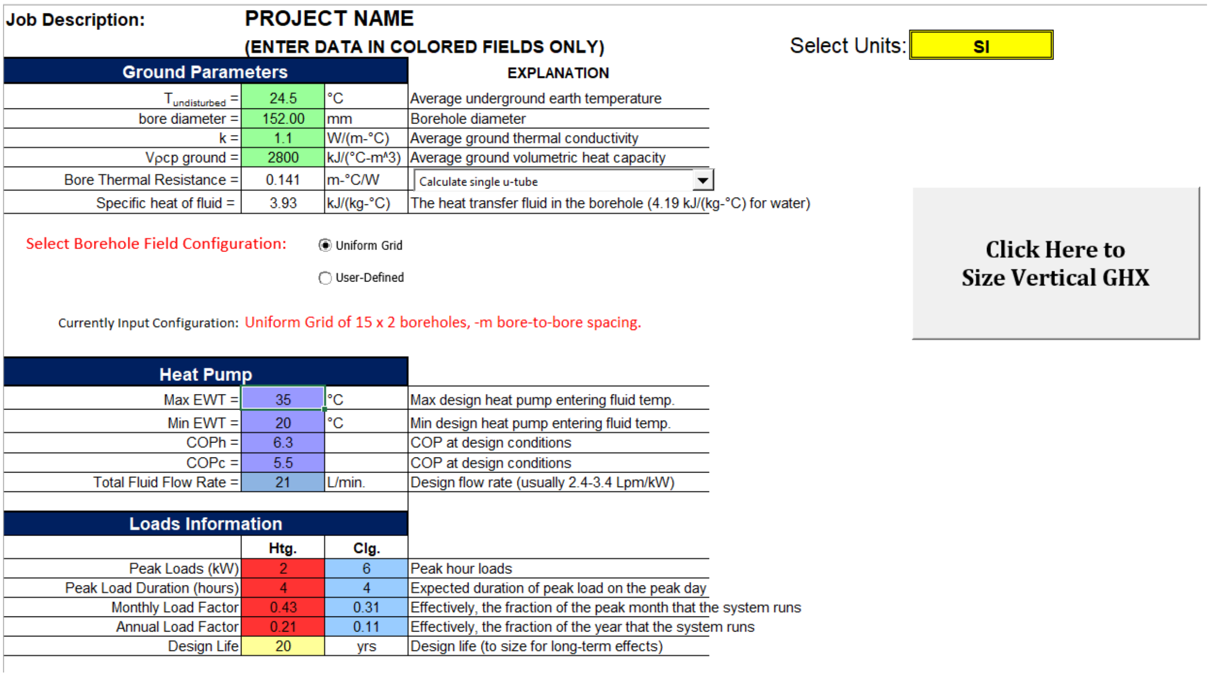


Figure 8: Ground Parameters Input Data



Job Description: **PROJECT NAME**

(ENTER DATA IN COLORED FIELDS ONLY)

Select Units: **SI**

Ground Parameters			EXPLANATION
$T_{undisturbed}$ =	24.5	°C	Average underground earth temperature
bore diameter =	152.00	mm	Borehole diameter
k =	1.1	W/(m-°C)	Average ground thermal conductivity
Vρcp ground =	2800	kJ/(°C-m^3)	Average ground volumetric heat capacity
Bore Thermal Resistance =	0.141	m-°C/W	Calculate single u-tube
Specific heat of fluid =	3.93	kJ/(kg-°C)	The heat transfer fluid in the borehole (4.19 kJ/(kg-°C) for water)

Select Borehole Field Configuration: Uniform Grid / User-Defined

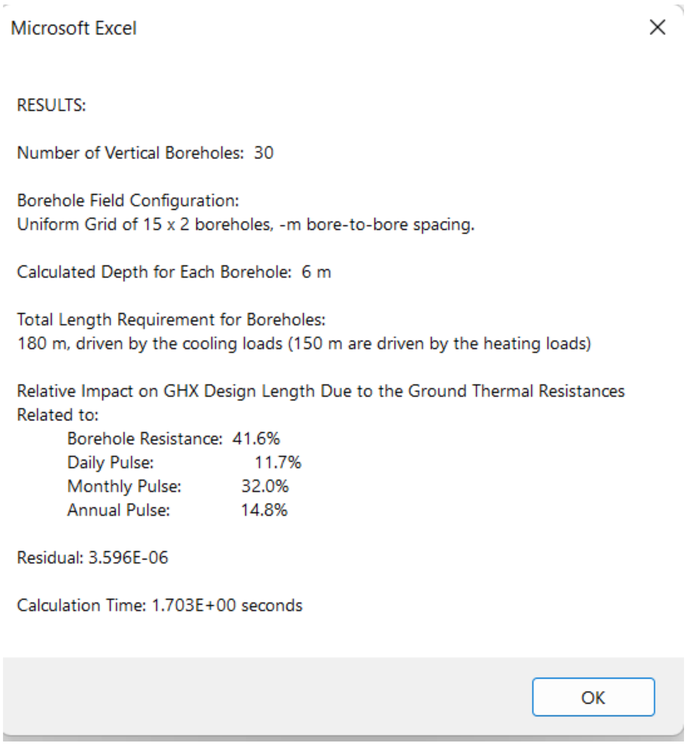
Currently Input Configuration: Uniform Grid of 15 x 2 boreholes, -m bore-to-bore spacing.

Click Here to Size Vertical GHX

Heat Pump			
Max EWT =	35	°C	Max design heat pump entering fluid temp.
Min EWT =	20	°C	Min design heat pump entering fluid temp.
COPh =	6.3		COP at design conditions
COPc =	5.5		COP at design conditions
Total Fluid Flow Rate =	21	L/min.	Design flow rate (usually 2.4-3.4 Lpm/kW)

Loads Information	Htg.	Clg.	
Peak Loads (kW)	2	6	Peak hour loads
Peak Load Duration (hours)	4	4	Expected duration of peak load on the peak day
Monthly Load Factor	0.43	0.31	Effectively, the fraction of the peak month that the system runs
Annual Load Factor	0.21	0.11	Effectively, the fraction of the year that the system runs
Design Life	20	yrs	Design life (to size for long-term effects)

Figure 9: Fluid Specifications Input Data



Microsoft Excel

RESULTS:

Number of Vertical Boreholes: 30

Borehole Field Configuration:
Uniform Grid of 15 x 2 boreholes, -m bore-to-bore spacing.

Calculated Depth for Each Borehole: 6 m

Total Length Requirement for Boreholes:
180 m, driven by the cooling loads (150 m are driven by the heating loads)

Relative Impact on GHX Design Length Due to the Ground Thermal Resistances
Related to:
Borehole Resistance: 41.6%
Daily Pulse: 11.7%
Monthly Pulse: 32.0%
Annual Pulse: 14.8%

Residual: 3.596E-06

Calculation Time: 1.703E+00 seconds

OK

Figure 10: Vertical GSHP Results

II. For Horizontal GSHP:

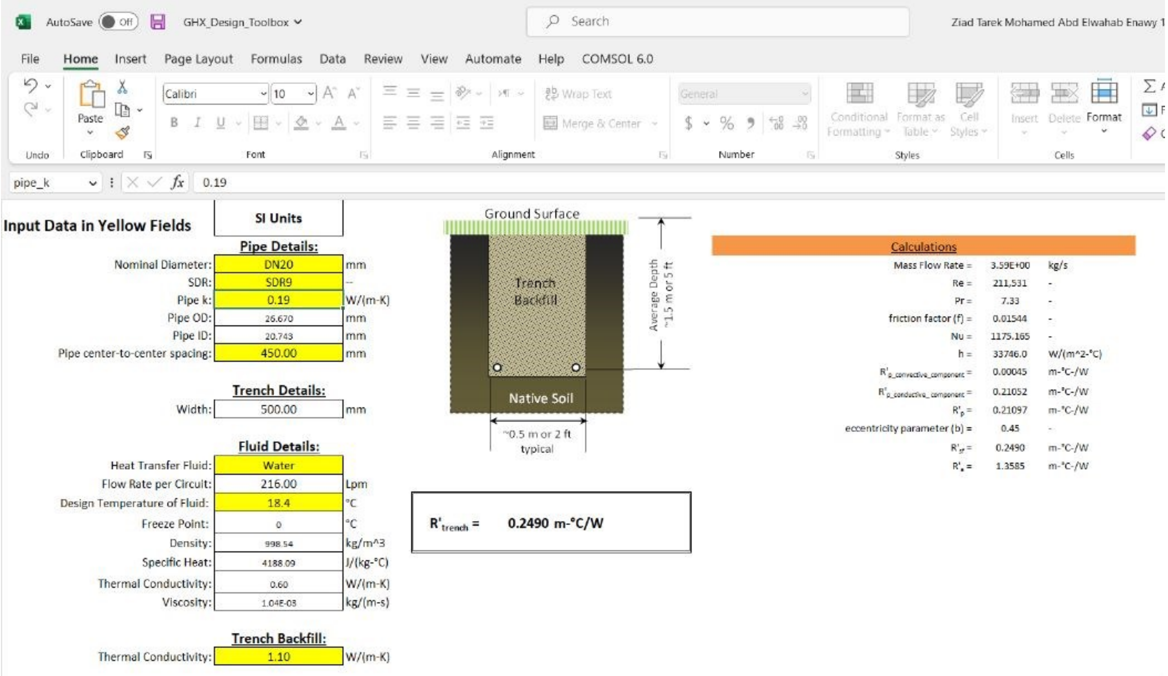


Figure 11: Pipe Parameters and Configuration Input Data

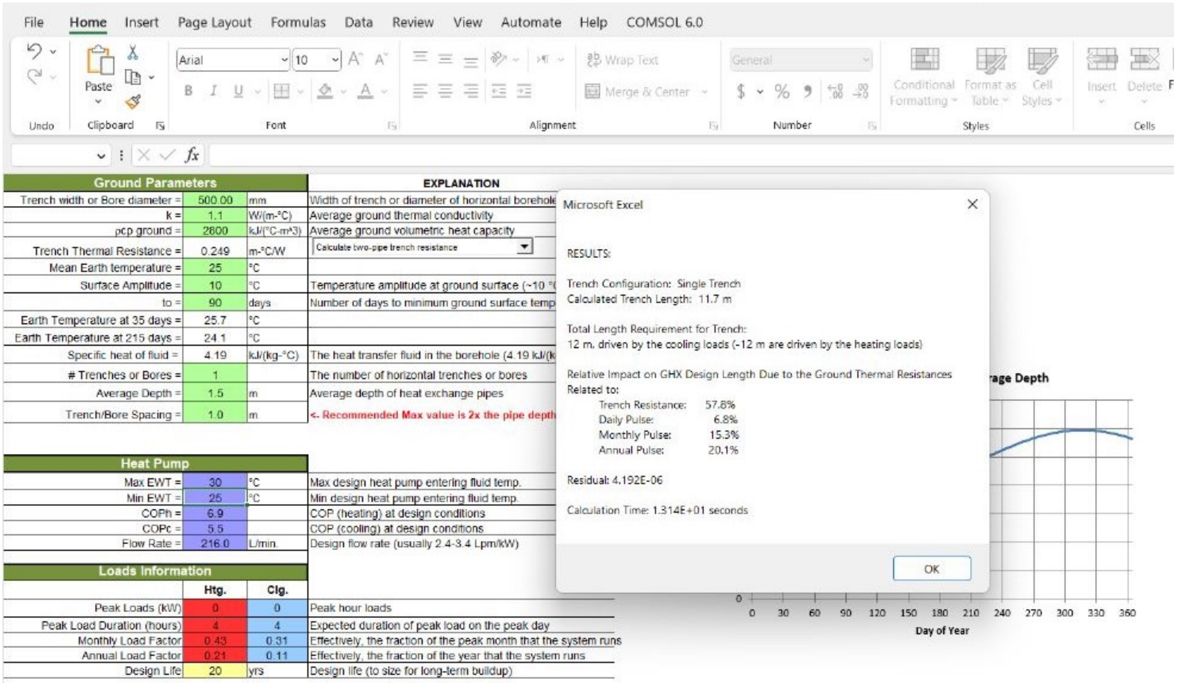


Figure 12: Ground Parameters and Horizontal GSHP Results

2.2.3 Governing Equations:

2.2.3.1 For vertical GSHP

Closed-loop ground heat exchanger design uses a simple steady-state heat transfer equation per unit length:

$$q = \frac{L_{bore}\,(t_g - t_w)}{R_{ov}} \qquad \textbf{eq.(1)}$$

Where:

- q = Heat transfer rate to/from ground
- L_{bore} = Ground heat exchanger bore length
- t_g = Ground temperature
- t_w = Average water-loop temperature
- R_{ov} = Overall resistance of ground and bore, ft·h·°F/Btu (m·K/W)

For Design Optimization: (solve for the vertical heat exchanger bore length)

$$L_{bore} = \frac{q \times R_{ov}}{(t_g - t_w)}$$

Where:

- ***The heat rate (q)*** is fixed by the building heating and cooling requirements.
- ***The ground temperature (t_g)*** is fixed by the earth.
- ***The overall resistance (R_{ov})*** is constrained by the thermal properties of the ground, the design of the heat exchanger, and the heat rate to and from the ground.

Target:

Design optimization is between the average water-loop temperature (t_w) and the heat exchanger length (and cost).

From Eq. (1):

It can be transformed to represent the variable heat rate of a ground heat exchanger by using a series of constant heat rate pulses as suggested by Ingersoll et al. (1954).

The thermal resistance (R) of the ground per unit length is calculated as a function of time, which corresponds to the time over which a particular heat pulse occurs.

Note:

- ***In cooling mode***, a lower value for t_w results in more efficient heat pump operation but a longer and more expensive ground loop.
- ***In heating mode***, a higher value for t_w results in improved heat pump operation but a longer and more expensive ground loop.

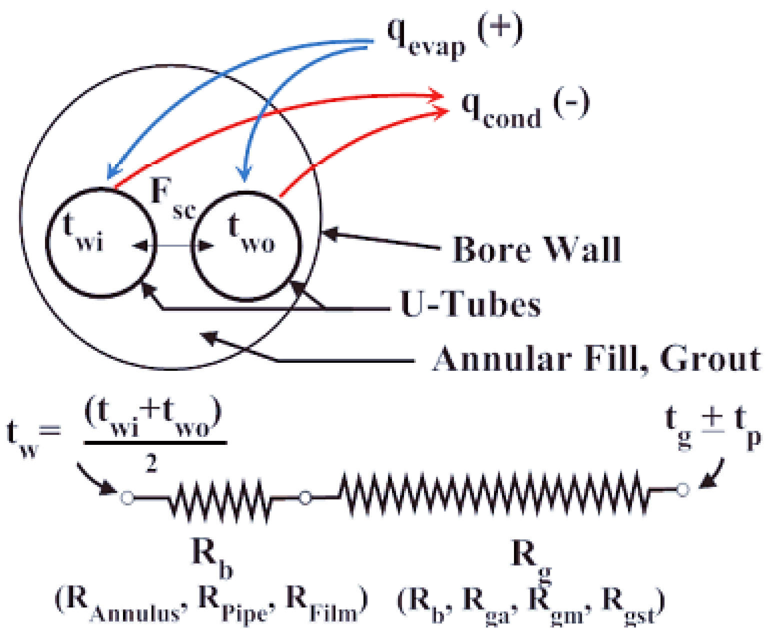


Figure 13: Schematic and Thermal Network for U-Tube Ground Heat Exchanger

From Figure (4):

The ***bore resistance*** $(\boldsymbol{R_b})$ that accounts for the thermal resistance of the tube wall (R_t), the ***film resistance*** between the fluid and tube $(\boldsymbol{R_{film}})$, and the ***resistance of the fill or grout material*** $(\boldsymbol{R_{annulus}})$ in the annual region between the tube(s) and the bore wall.

From Equation (2) & (3):

A minimum of three heat pulses are included: an average annual pulse, an average monthly pulse preceding the design day, and a short-term pulse that is typically the maximum pulse during the design day of one to six hours in length.

Resulting Equation for ground heat exchanger bore Length:

$$L_{cooling} = \frac{q_a\, R_{ga} + q_{cond}\left(R_b + PLF_m\, R_{gm} + F_{sc}\, R_{gst}\right)}{t_g - \frac{ELT + LLT}{2} + t_p} \qquad \boldsymbol{eq.(2)}$$

$$L_{heating} = \frac{q_a\, R_{ga} + q_{evap}\left(R_b + PLF_m\, R_{gm} + F_{sc}\, R_{gst}\right)}{t_g - \frac{ELT + LLT}{2} + t_p} \qquad \boldsymbol{eq.(3)}$$

Where:

- F_{sc} = short-circuit heat loss factor between supply and return tubes in bore
- $L_{cooling}$ = required bore length for cooling, ft (m)
- $L_{heating}$ = required bore length for heating, ft (m)
- PLF_m = part-load factor during design month
- q_a = net annual average heat transfer to the ground, Btu/h (W)

- R_{ga} = effective thermal resistance of the ground—annual pulse, h·ft·°F/Btu (m·K/W)
- R_{gst} = effective thermal resistance of the ground—short-term pulse, h·ft·°F/Btu (m·K/W)
- R_{gm} = effective thermal resistance of the ground—monthly pulse, h·ft·°F/Btu (m·K/W)
- R_b = thermal resistance of bore, h·ft·°F/Btu (m·K/W)
- t_g = undisturbed ground temperature, °F (°C)
- t_p = long-term ground temperature penalty caused by ground heat transfer imbalances, °F (°C)
- ELT = heat pump entering liquid temperature, °F (°C)
- LLT = heat pump leaving liquid temperature, °F (°C)

Notes:

- The sign convention for Equations (2) and (3) assumes the energy balance is done on the heat pumps; therefore, $\boldsymbol{q_{evap}}$ **is positive**, $\boldsymbol{q_{cond}}$ **is negative**.

  $\boldsymbol{q_a}$ **is positive** if the annual amount of heat removed from the ground in heating (q_{evap} × operating time) is greater than the heat added to the ground in cooling (q_{cond} × operating time).

  $\boldsymbol{t_p}$ **is positive** for a long-term rise in ground temperature.

- ***In the cooling mode***, the optimal trade-off between system efficiency and ground-loop length typically occurs when the *maximum value for the heat pump ELT is 20°F to 30°F (11°C to 17°C)* **greater than** the undisturbed ground temperature (t_g).

- ***In the heating mode***, the optimum value for the *ELT is typically 8°F to 15°F (5°C to 8°C)* **less than** the undisturbed ground temperature (t_g)

Optimum Liquid Flow Rates of heat pumps:

For closed-loop systems, they are typically in the 2.5 to 3.0 gpm/ ton (***2.7 to 3.2 L/min·kW***) range. The following estimates can be used with good accuracy for the heat pump LLT. These values assume water is the fluid; values will be 3% to 5% higher for typical antifreeze solutions used with GSHPs.

- **For a flow rate of 3.0 gpm/ton (3.2 L/min·kW)** the LLT will be approximately 10°F (5.6°C) higher than the ELT in cooling and 6°F (3.3°C) lower than the ELT in heating.

- **For a flow rate of 2.5 gpm/ton (2.7 L/min·kW),** the LLT will be approximately 12°F (6.7°C) higher than the ELT in cooling and 7.2°F (4°C) lower than the ELT in heating.

- **For a flow rate of 2.0 gpm/ton (2.15 L/min·kW),** the LLT will be approximately 15°F (6.7°C) higher than the ELT in cooling and 9°F (5°C) lower than the ELT in heating.

Fluid	Temperature, °C	10 L/min			20 L/min			40 L/min		
		25 mm	32 mm	40 mm	25 mm	32 mm	40 mm	32 mm	40 mm	50 mm
Water	20	10030	7769	6293	20129	15657	12616	31342	25165	20080
20% Propylene glycol	0	2625	2011	1666	5315	4080	3238	8138	6570	5300
20% Propylene glycol	10	3750	2925	2314	7577	5844	4689	11767	9465	7543
20% Propylene glycol	30	7030	5484	4350	14022	10916	8820	21774	17482	13964
30% Propylene glycol	0	1500	1188	925	3053	2316	1898	4729	3786	3058
30% Propylene glycol	10	2343	1828	1481	4749	3639	2903	7258	5902	4689
30% Propylene glycol	30	4968	3839	3054	9951	7718	6252	15506	12471	9989
25% Methyl alcohol	0	3093	2376	1944	6220	4852	3908	9678	7795	6218
25% Methyl alcohol	10	4499	3565	2869	9160	7057	5694	14186	11358	9072
25% Methyl alcohol	30	8343	6490	5183	16736	1301	10383	25953	20823	16614

To estimate loop water flow: L/min ≈ q (kW) ÷ [0.0692 × Δt (°C) × No. of Parallel U-Tubes]

Table 1: Reynolds Numbers in DR 11 HDPE Pipe for Various Pipe Diameters and Flow Rates

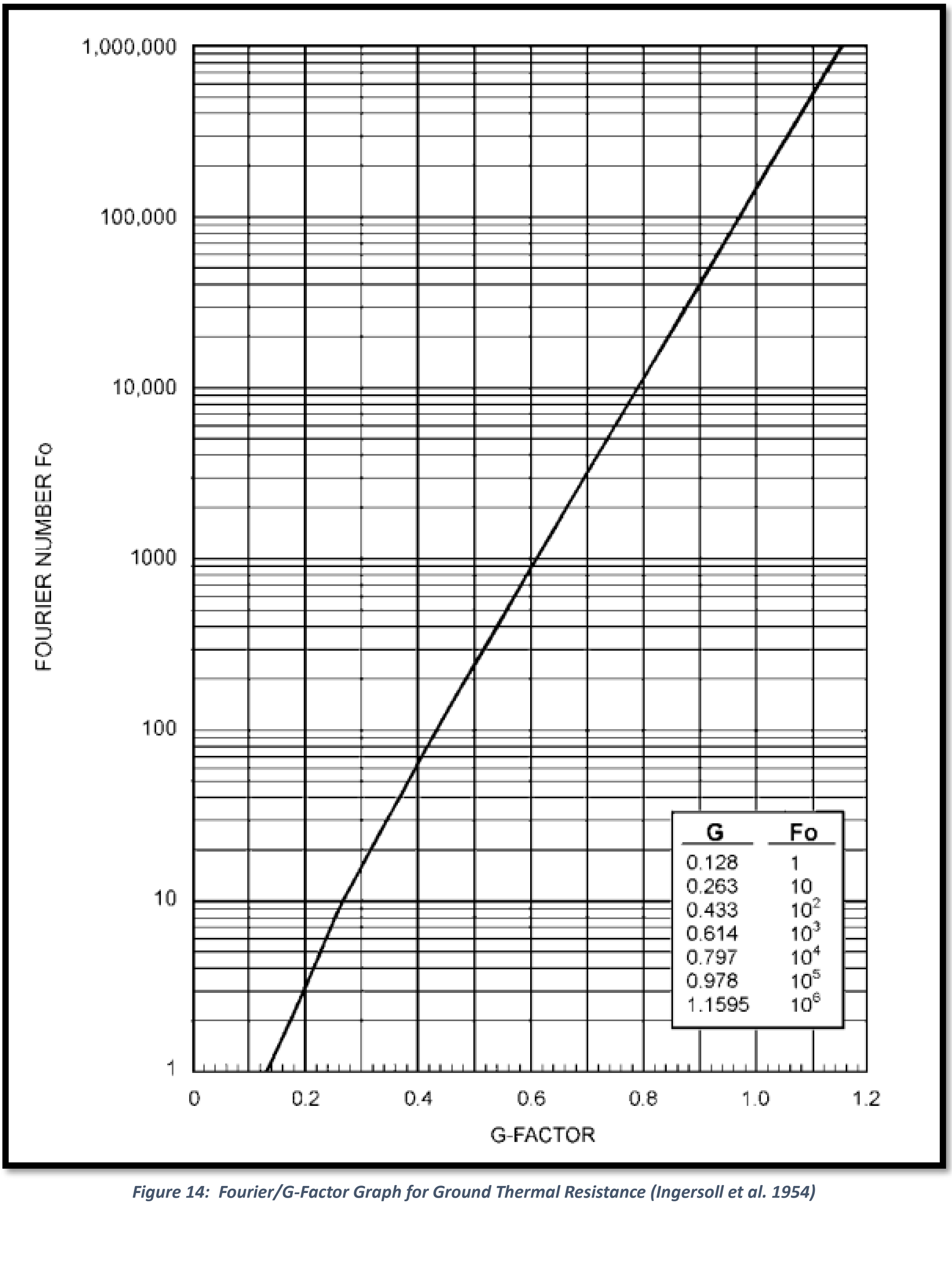


Figure 14: Fourier/G-Factor Graph for Ground Thermal Resistance (Ingersoll et al. 1954)

2.3 Finite Element Simulation:

For two case studies, a 3D model was built using Autodesk Fusion 360 and imported to COMSOL Multiphysics to create a numerical model and validate it against analytical and experimental work.

This work aims to simulate the model through two studies:

i. ***Stationary [Steady-State solution]***
ii. ***Time-dependent (aiming for 24hr & 30 days simulation time) [Transient solution]***

2.3.1 CFD Methodology for Cairo University Case Study

A 3D symmetrical model was designed and imported to COMSOL Multiphysics to simulate a single U-tube borehole for a Vertical GSHP in certain working conditions. Soil properties from Table 2 were applied to the soil geometry. Uniform soil property was applied to the model. Model Builder Physics and Materials are found in Appendix A.

In the finite element simulation, it was assumed that the soil properties are function to the change in ground temperature throughout the simulation time. The soil domain dimensions used in the simulation considering it as a cylinder of 5 m radius and 45 m depth. It was also assumed that negligible ground water movement was present within the soil domain. Borehole resistance associated with this geometry was calculated in the determination of total piping length required for Cairo University rock laboratory using GHX_Design_Toolbox. The inlet flow temperature and the flow velocity are considered as an input for heat transfer in fluids and solids, and laminar flow respectively. The input data can be found in the Results section.

Table 2: Cairo University Soil properties

Property	*Value*	*Unit*
Thermal Conductivity	$k(T)$	$W/(m.k)$
Density	$\rho(T)$	kg/m^3
Heat Capacity	780	$J/(kg.k)$

2.3.2 CFD Methodology for Ain Shams University prototype

A full 3D model was designed and imported to COMSOL Multiphysics to simulate the prototype (horizontal ground heat exchanger configuration) with soil domain dimensions in certain working conditions found in the prototype section. Soil properties same as Table 2 except *heat capacity = 830 J/(kg.k)* and applied to the soil geometry. Uniform soil property was applied to the model.

In the finite element simulation, it was assumed that the soil properties are function to the change in ground temperature throughout the simulation time. It was also assumed that negligible ground water movement was present within the soil domain. Borehole resistance associated with this geometry was calculated in the determination of total piping length required for the prototype using GLD software. This model was developed by coupling the governing equations of both heat and mass transfer in the soil and fluid flow in horizontal ground heat exchangers. The heat transfer mechanism in this model was primarily the heat conduction in the soil, the pipe walls, and partly in the carrying fluid, as well as the heat convection in the carrying fluid The input data can be found in the Results section.

2.4 Life Cycle Assessment:

2.4.1 Introduction

At present, the importance for the subsistence of life on the planet is professionally, theoretically, and even people in the street in the public eye, to cope with high emissions of substances harmful to the environment associated with the action of people. For this reason, the reduction of emissions has become the battleground in the fight for the preservation of the environment [14-16].

Industry is in constant innovation, production and application of new technologies that contribute to one's comfort, but paradoxically, this increases the damage to the environment. To cut back on risks and environmental damages, there are effective methods, which identify the weaker factors of each process, and that must be developed. One of these methods is LCA which due to the systematic, objective, and global nature constitutes a more appropriate methodology for environment order [17,18]. The intense industrial activity and manufacturing processes require a high consumption of energy and have a significant influence on greenhouse gases (GHG) emissions, which has a negative impact on the preservation of resources and the environment, due to its contribution to global warming. These impacts include of GHG emissions, such as carbon dioxide (CO2), the main worldwide polluting gas, and other gases like methane, nitrous oxide and chlorofluorocarbons which can be measured in units of CO2 equivalent to (CO2-eq) [19,20].

LCA has become a highly important tool for providing in-depth analyses of this kind, for instance in studies concerned with the replacement of fossil fuels by renewables in electricity production, and a significant option in the process of transition towards a low emission production economy. It has used the Life Cycle Assessment used as a methodology which assesses environmental impacts caused by products, processes, or systems.

According to ISO 14040 standards, LCA is defined as the collection and evaluation of the inputs and outputs for determining possible environmental impacts of a product, process, or system during its life cycle. Thus, LCA is a tool for the analysis of the environmental burden of products in all phases of its life cycle, from the extraction of resources, production of materials, pieces, and the product itself, until the use of the mentioned product and residue management after being discarded, whether re-purposing, recycling, or final disposal [21]. The main parts of the LCA are the following:

a) Discuss the purpose and definition of the scope of application of this approach.
b) Make an inventory of the inputs and outputs of the system.
c) Assess all types of impacts on the environment.
d) Interpret the results and evaluate the impacts.

There are LCA studies and works include environmental issues about energy productions systems, but few comparatives between different systems that cover the same demands and are considered renewable. One of them is LCA comparative of wood pellets and wood split logs for residential heating which provides information on the impacts generated by the combustion of the wood and its by-products in three types of places, a pellet boiler, a waterproof stove and a traditional fireplace [22], other study is LCA Comparative of electric generation by different wind turbine types which shows us that most environmental impacts are associated with the manufacture of fundament, tower and nacelle [23] and last example is

LCA comparative of fixed and single axis tracking systems for photovoltaics to understand the environmental differences between both systems [24]. These studies have used different software and different methods of analysis, which gives us information to contrast with the results of this studies.

2.4.2 LCA Report Scope Objective

The present work deals with the environmental impacts caused by a water-to-air heat pump as it is considered as a renewable energy system. Emissions produced by the processes during extraction of materials, manufacture, operation, and end-of-life stage of the system has been considered. So, the scope is focusing on applications of geothermal heat pumps in different projects.

Goal / Scoping:	Evaluate LCA comparing potential environmental impacts of geothermal heat pumps relative to conventional HVAC system
Application:	Basis for decisions on geothermal heat pumps installation
Functional Unit:	One unit of a heat pump (e.g., 1 ***KWhth*** of the process)
System Boundaries:	- Geothermal Heat pumps operational energy - All manufacturing processes contributing significantly to the life cycle impacts are considered. - Scope is based on ***Cradle-to-gate***

2.4.3 Life Cycle Inventory Analysis:

It is to quantify the environmental inputs and outputs as everything is measured by defining what flows in and out of the system.

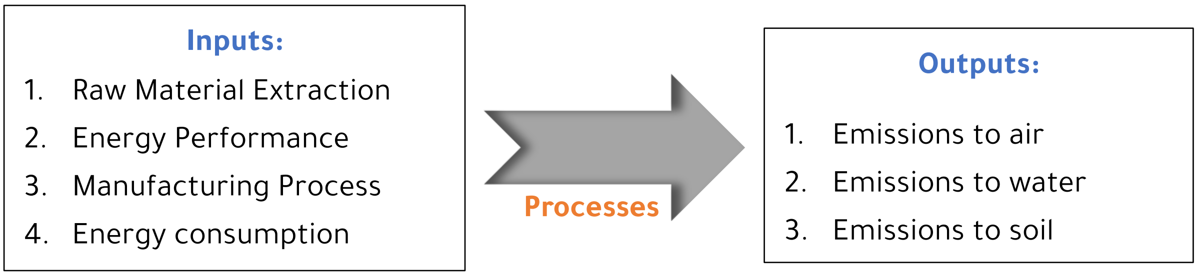


Figure 15: LCA Inventory Analysis (Inputs & Outputs)

2.4.4 Flow Diagram:

To acquire knowledge about the system, a flow diagram of each process related to the heat pump is show in Fig. 2. The figure shows a diagram of the life cycle of a heat pump, from the extraction of materials to its end of life. In some cases, it consists of disambiguation of some elements, and in others, the transfer to landfill. The inputs and outputs of both materials and energy, occurs throughout the cycle, being essential in the study a rigorous collection of these quantities. One of the main points of the LCA methodology consists of an inventory of the major inputs and outputs. To achieve this objective, it has been used various sources among which are the manufacturer's catalogues, information in the literature and databases of environmental data of the SimaPro. In addition, the following databases from *Ecoinvent 3.0*, *EU & DK Input Output Database*, *Industry data 2.0*, *USL CI* and *Methods* have been consulted. These databases offer a significant amount of data relating to resource consumption and emissions during manufacturing. The most import raw materials which are involved in the processes of the cycle of life have been considered. [25,26].

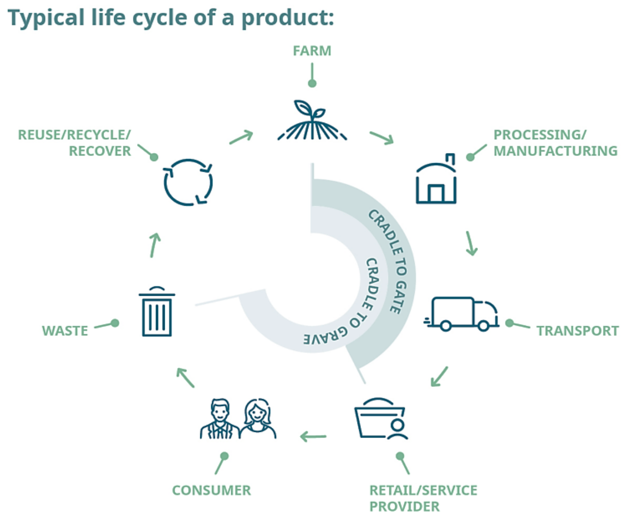


Figure 16: Product Life Cycle

- ***Flow Diagram for the process:***

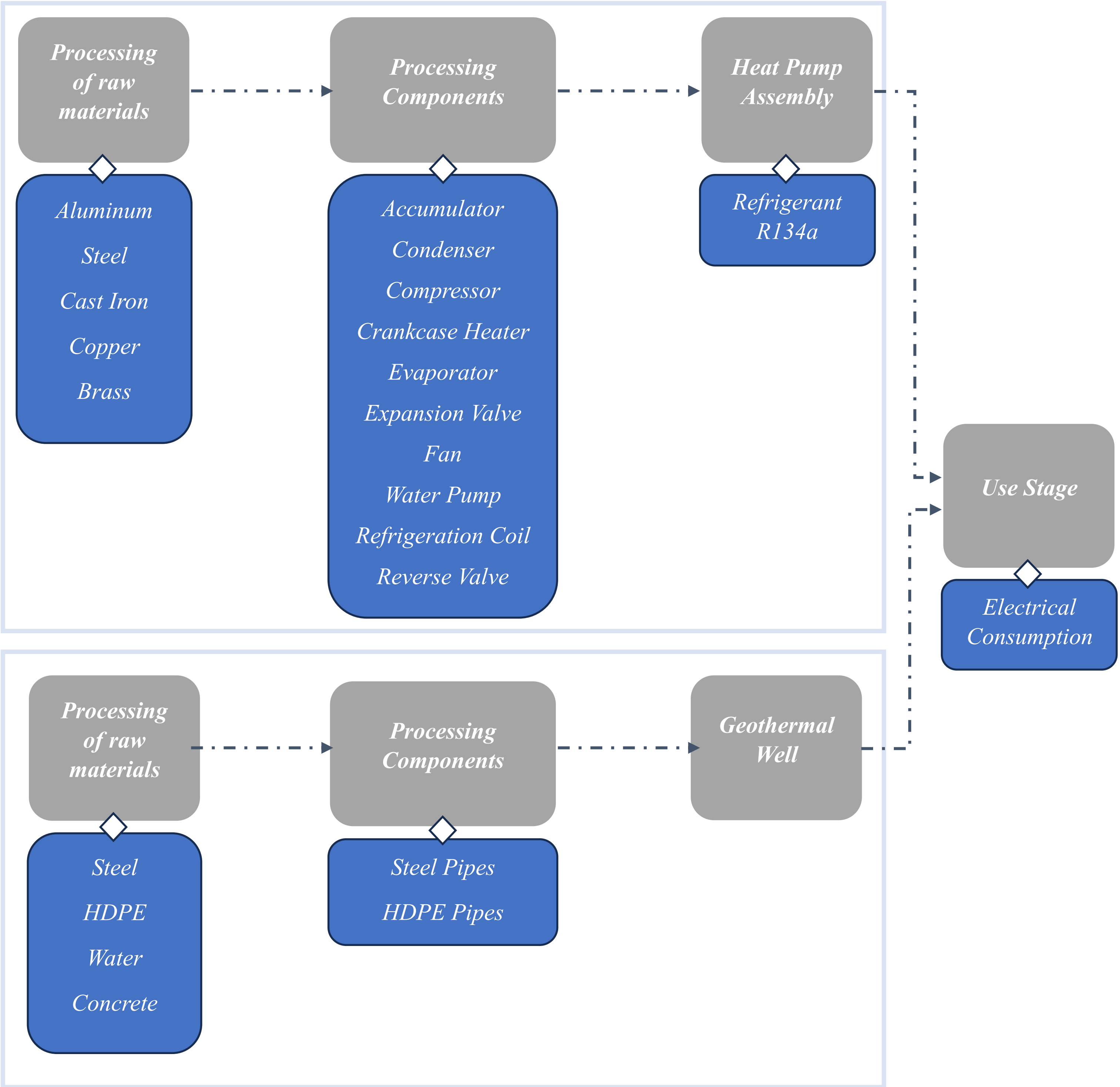


Figure 17: Process flows of the heat pump.

2.4.5 LCA Impact Assessment Method:

In this section, it has been analysed and quantified the results of the inventory. This process will allow to obtain environmental indicators from the list of emissions and consumed resources caused by the heat pump system during its life cycle so it can become easier to comprehend. For this transformation it has been used one method which is ***Environmental Footprint v3.0*** and the choice of this method is to obtain a final impact value.

Environmental Footprint v3.0 was performed for those impact categories:
Climate change, ozone depletion, Ionizing radiation, Photochemical ozone formation, Particulate matter, Human toxicity (non-cancer/cancer), Acidification, Ecotoxicity freshwater, land use, and water use.

- ***Method Results:***

It has been obtained global results and proceed to compare the different impact categories.

- ***Characterisation:*** In this method damage categories can be studied, and they are measured in different units.
- ***Weighting:*** The calculation of values of normalization is based on emissions data measured in various European countries, and then carry out an extrapolation at European level to estimate the total European emissions per year/inhabitant. Fig. 39.
- ***Single score:*** In this step, the relative importance of each category of impact is determinate. *The unit called the Eco-point indicator (Pt) is used*. It should be noted that the absolute value is not very relevant, because the main objective is to compare the relative differences between products or components or processes.

2.5 Prototype

This prototype was built to discuss the ability of designing a horizontal ground source heat pump that aims to operate a 1/8 hp dispenser giving a high performance to the system instead of using conventional electric grid.

2.5.1 Simulation Parameters

To conduct this research, a state-of-the-art transient 3D finite element model was developed at Ain Shams university by using COMSOL Multiphysics, a widely used finite element simulation software. As explained in Section 2.3

2.5.2 Prototype System Configuration

The following Figure.58 shows the prototype schematic design:

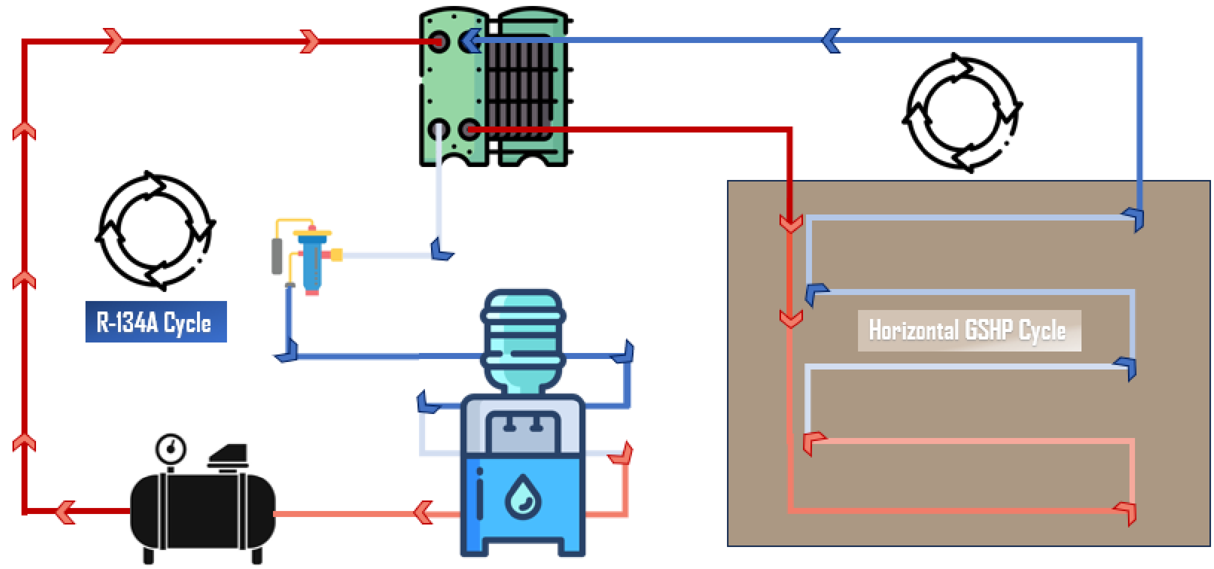


Figure 18: Prototype schematic diagram

The prototype consists of the following components:

- ***Dispenser:*** containing a compressor and an expansion valve where the refrigerant R-134A flows through.
- ***Plate Heat Exchanger:*** where the hot refrigerant exchanges heat with cold water flowing out from the soil (in our case Cooling Mode).
- ***Soil in a wooden box:*** where HDPE pipes are installed in a horizontal configuration as explained in detail in the following section and these pipes are connected to a circulating pump.

For the selection of the prototype's plate heat exchanger and circulating pump, calculations are made and found in Appendix A.

2.5.3 Prototype GLD parameters and results

2.5.3.1 Parameters and Specifications

Program used to design horizontal ground source heat pump is GLD software (Ground Loop Design). In this program, the following parameters are entered for the design of the prototype fluid temperature difference as an output:

1) Monthly Load calculation:

Since the prototype goal is to operate 1/8 hp dispenser and by assuming the heat pump system will work in cooling mode during summer only, then the monthly load data will be as following:

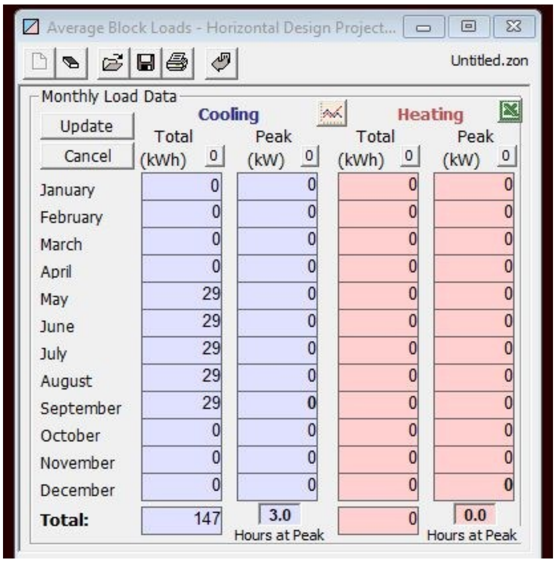


Figure 19: Monthly Load data - GLD software

2) Pump power calculation:

From calculations, the water flow inside the pipe required an input power of 0.1 kW and selecting a pump of motor efficiency 85%. Details and screenshots in Appendix A.

3) Heat pump specifications:

The following Table.3 shows the required capacity, power, COP, and flow rate needed to design the system *in cooling mode only*. Screenshots in Appendix A.

Table 3: Heat pump specifications at design temperature and flow rate - GLD software

Parameters	*Value*	*Unit*
Capacity	0.1	kW
Power	0.07	kW
Flow Rate	16.5	$L/min.$
Coefficient of performance (COP)	1.4	-
Partial Load Factor	0.7	-

4) *Fluid specifications:*

It was determined that the fluid flowing in pipes is water only since the prototype is built in a hot region. From the following Table.4, Fluid specifications are clarified:

Table 4: Prototype Water Fluid Specifications – GLD Software

Parameters	Value	Unit
Design Temperature	25.0	$°C$
Specific heat (Cp)	4.182	$kJ/(kg.k)$
Density (ρ)	999.6	kg/m^3

5) *Pipes specifications:*

The following Table.5 shows the HDPE pipes specifications:

Table 5: Prototype Pipe Specifications – GLD Software

Parameters	Value	Unit
Pipe Resistance	0.103	$m.k/W$
Pipe Size	1 in. (25 mm)	mm
Inner Diameter	28.4	mm
Outer Diameter	33.5	mm
Pipe Type	SDR13.5	-
Flow Type	Laminar	-

6) *Soil specifications:*

According to GLD software, the ground temperature is embedded on the software and can't be edited since the software used is a demo, the following Table.6 shows the soil specifications:

Table 6: Prototype Soil Specifications – GLD Software

Parameters	Value	Unit
Ground Temperature	16.7	$°C$
Thermal Conductivity	1.3	$W/(m.k)$
Thermal Diffusivity	0.058	m^2/day
Ground Temperature Corrections at given depth:		
Regional Air Temperature swing	20.4	$°C$
Coldest/Warmest Day in the year (1 - 365)	25 / 225	-

2.5.3.2 GLD Trench and Pipe Configuration

For determining the optimum inlet and output flow temperature, certain pipe configuration is selected as follows:

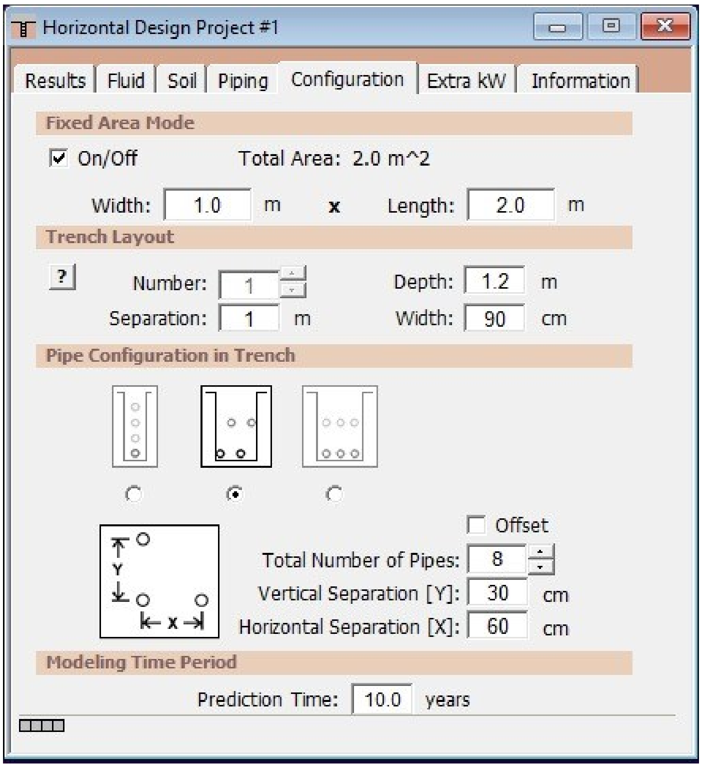


Figure 20: Prototype Pipe configuration from GLD software

From this calculation, it was determined that the dimensions of the box that will contain the soil, will be $(1.2 \times 2.4 \times 1.2)\ m$. From Figure.68, A diagram of the GLD pipe configuration parameters is shown. As for the pipe configuration, it has changed slightly from the GLD software since the wooden box required a certain amount of soil to be able to handle the box from collapsing.

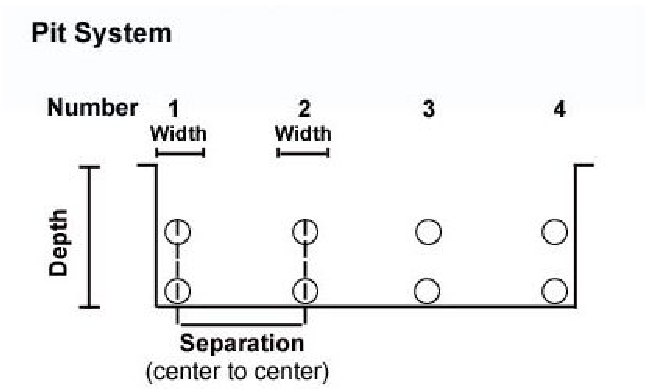


Figure 21: GLD pipe configuration diagram

2.5.3.3 Prototype GLD Results

Results shows that the prototype system will be able to output a temperature difference of $7.5°C$, where:

$$\Delta T = (T_{out} - T_{in})_{heat\ pump\ perspective} = 43.5 - 36 = 7.5°C$$

Considering that water is heated and enters the soil with a temperature of $43.5°C$ and the ground temperature is $16.7°C$. (default in GLD software)

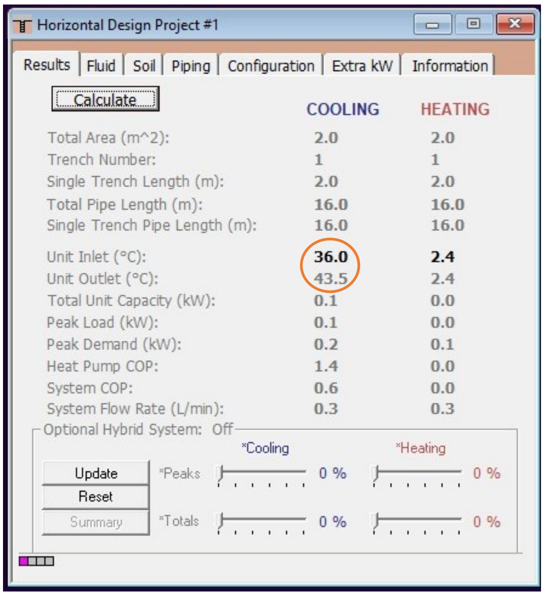


Figure 22: Prototype GLD Results

2.5.4 Prototype Experimental Setup

2.5.4.1 Experimental Pipe Configuration

As for building the prototype, the box used about more than $500\ kg$ of soil filling the box to a height of $0.75\ m$ while setting the U-shape trench pipe of dimensions $(2\ m\ length \times 0.566\ m\ in\ width)$ creating a horizontal 4-loop GSHP with a vertical spacing of $18\ cm$ between each trench pipe measured from the pipe centreline. A CAD is drawn by Autodesk Fusion 360 to illustrate the previous dimensions in Figure.23.

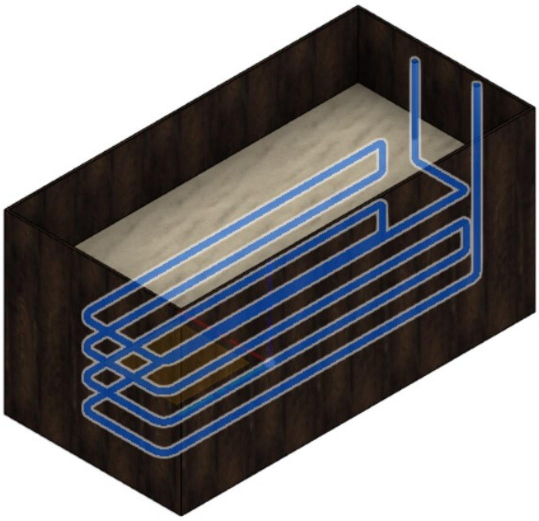


Figure 23: Prototype CAD Model

Prototype Dimensions	
Vertical Spacing	*18 cm*
Pipe Size	*1-inch (25 mm)*
No. of Loops	*4*
Loop Size $(L \times W)$	*(2 × 0.566) m*
Box Sizing $(L \times W \times H)$	*(1.2 × 2.4 × 1.2) m*
Soil Height	*0.75 m*

2.5.4.2 Experimental Prototype Parameters

Same input data as GLD software in the parameters and specifications section in case of monthly load calculations, pump power calculations, heat pump, fluid, pipes, and soil specifications except that the initial ground temperature of the soil is $25\ °C$, flow rate of the circulating pump is $1\ L/min.$, and the initial inlet fluid temperature to the soil is $33.5\ °C$.

2.5.4.3 Experimental Prototype Instrumentation

Various instruments were used to measure power consumed from dispenser's compressor, and temperature of water and soil at various locations as illustrated in Fig. 24. A diaphragm pump (capacity 13.8 W, maximum discharge 1.2 L/min., volts: 24 VDC) is used to circulate the water in the pipes buried in the soil.

To measure the power consumed from dispenser's compressor, a digital Avometer is used for measuring the current, and since the compressor is connected to the electric grid (220 V AC), the power consumed is determined.

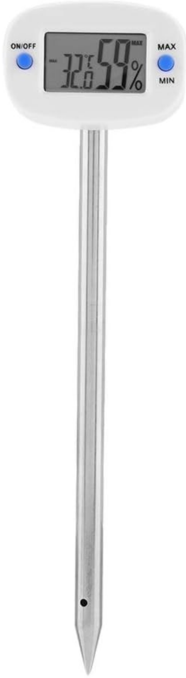
To measure the temperature of the inlet and output fluid temperature from the soil, thermocouple type K sensors were installed each at the inlet and outlet of water inside the pipes. To measure the temperature of the soil, a Digital Soil Hygrometer Meter Temperature Tester was used and positioned at the centre of the box as well as thermocouples are used at various depth in the soil.

All the sensors were connected to Arduino-UNO which is supplied by power from a laptop where data is being recorded each hour.

2.5.4.4 Experimental Prototype Photos

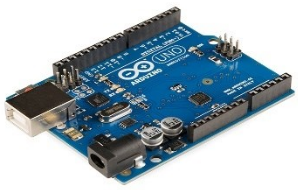
a) Multi-meter Avometer

b) Thermocouple type K



e) Soil Hygrometer

c) Diaphragm pump



d) Arduino UNO

Figure 24: Devices used in experimentation.

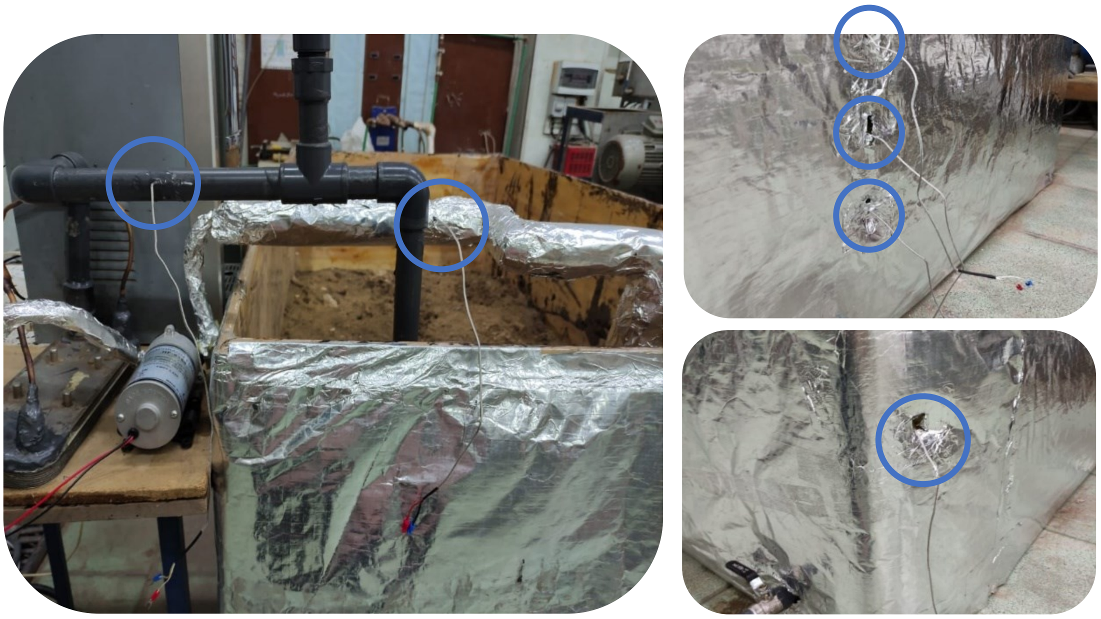


Figure 25: Sensors at various points of the prototype

a) dispenser components b) plate heat exchanger + diaphragm pump c) Laptop setup

Figure 27: Prototype Instrumentation Setup

Figure 26: Prototype from different angles

2.5.4.5 Prototype sensors configuration

The prototype is being monitored by 7 thermocouples type K (5 to measure the soil and 2 to measure inlet and outlet of the heat pump) and 1 digital soil hygrometer. The following schematic figure shows the position of each sensor:

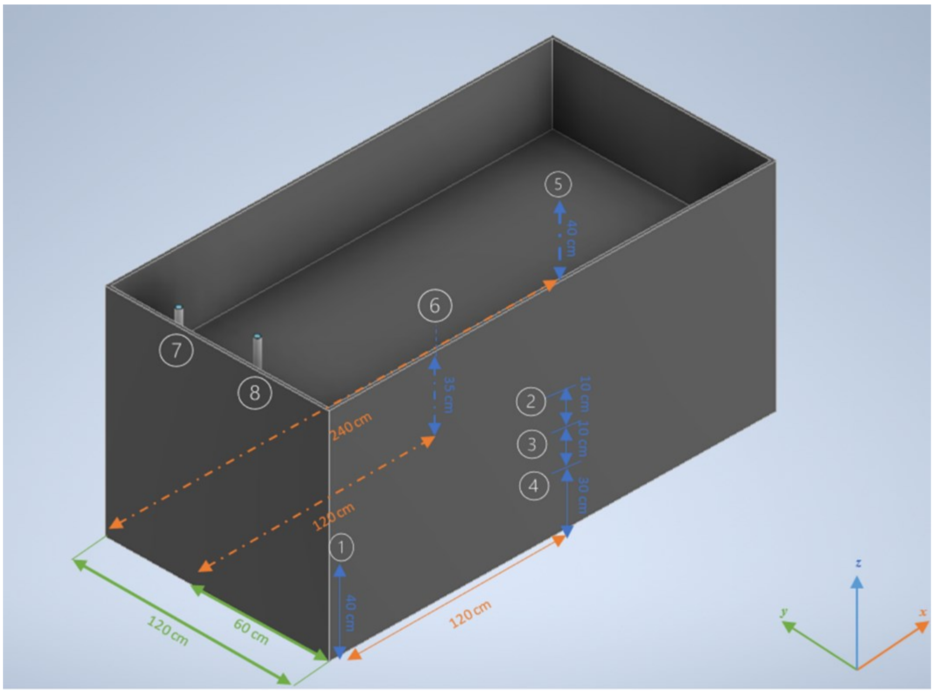


Figure 28: Prototype sensor configuration

Pin No.	Sensor Type
1	Thermocouple Type-K for soil
2	
3	
4	
5	
6	Digital Soil hygrometer
7	Thermocouple Type-K for inlet and outlet of pipes
8	

Results:

3.1. Cairo University Case Study:

3.1.1 Input Data:

The computer laboratory of the rock engineering department in Cairo university was selected to design a full cooling and heating system using GCHP.

P.O.C	***Data***
Location:	- The department of Mining, Metallurgical and Petroleum Engineering Department (Building No. 32). - It is located on the ground floor.
Site Geological Data:	Lat.: 30.0244932, Long.: 31.2099024
Lab Area	$Area\ (m^2)\ \approx\ 47.25\ m^2$
Lab Activities:	As a part of the Rock Engineering Laboratory, it is used in the following activities: - Processing and analysing of testing results. - Technical meetings between lab members. - Teaching activities such as practical training sessions in different mining program courses such as the fundamentals of rock mechanics.
Future Lab Activities:	Furthermore, this room is planned to be used in the teaching activities of the diploma modulus as indicated in the surveying reports on CU facilities and equipment.
Operational Conditions:	- ***Six*** *people &* ***Six*** *computer units.* - ***Demand Temperature*** $= \mathbf{24\ °C}$ *according to ASHRAE standard design.* - ***Operating hours:*** *7:00 am to 10:00 pm (16 hours)*
Software used	HAP

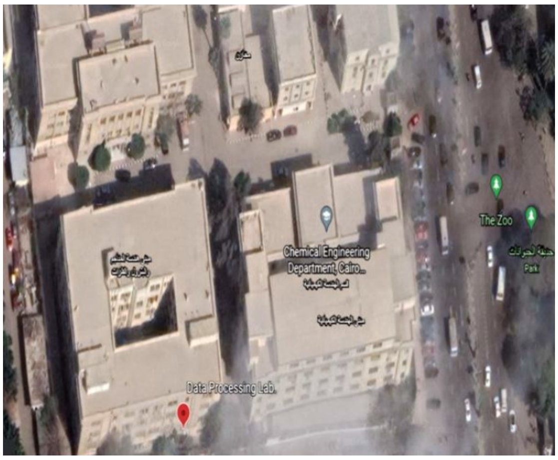


Figure 30: Lab Site from Google

Figure 29: Visit to the lab (Inside)

3.1.2 Results:

3.1.2.1 Cooling and Heating Load Calculations

Time	January	February	March	April	May	June	July	August	September	October	November	December
0:00	1.6	1.5	1.3	1.2	4.6	5.0	5.1	5.3	5.1	4.5	1.2	1.5
1:00	1.7	1.6	1.4	1.3	4.5	4.9	5.0	5.1	5.0	4.4	1.3	1.6
2:00	1.8	1.7	1.5	1.4	4.3	4.8	4.9	5.0	4.9	4.2	1.4	1.7
3:00	1.9	1.8	1.6	1.5	4.2	4.7	4.8	4.9	4.7	4.1	1.5	1.8
4:00	2	1.9	1.6	1.5	4.2	4.6	4.7	4.8	4.6	4.0	1.6	1.9
5:00	2.1	1.9	1.7	1.6	4.1	4.6	4.7	4.7	4.6	3.9	1.7	1.9
6:00	2.1	1.9	1.6	1.5	4.2	4.7	4.7	4.8	4.7	4.0	1.6	2
7:00	1.9	1.7	1.4	1.3	3.7	4.1	4.2	4.3	4.2	3.8	1.4	1.8
8:00	1.6	1.4	1.2	1.2	3.7	4.2	4.3	4.5	4.6	3.9	1.1	1.5
9:00	1.3	1.2	1	0.9	4.2	4.2	4.3	4.6	4.8	4.5	0.8	1.2
10:00	1	0.9	0.7	0.7	4.3	4.7	4.8	4.9	5.2	4.8	0.5	0.9
11:00	0.7	0.6	0.5	0.5	4.4	4.8	4.8	5.2	5.3	5.1	0	0.6
12:00	0.4	0.3	0	0.3	4.8	5.1	5.2	5.4	5.6	5.3	0	0.3
13:00	0	0	0	0	4.9	5.1	5.2	5.3	5.7	5.3	0	0
14:00	0	0	0	0	4.7	5.2	5.3	5.6	5.5	5.4	0	0
15:00	0	0	0	0	5.0	5.1	5.5	5.6	5.8	5.3	0	0
16:00	0.3	0	0	0	5.1	5.4	5.4	5.6	5.7	5.2	0	0
17:00	0.5	0.4	0	0	4.9	5.1	5.2	5.3	5.5	5.2	0	0.3
18:00	0.6	0.6	0.4	0.4	4.6	5.1	5.2	5.3	5.1	4.8	0	0.5
19:00	0.8	0.8	0.6	0.6	4.6	4.7	5.0	5.0	5.1	4.7	0.4	0.7
20:00	1	1	0.8	0.7	4.3	4.8	4.5	4.9	4.9	4.5	0.6	0.9
21:00	1.2	1.1	0.9	0.9	4.1	4.4	4.6	4.6	4.6	4.3	0.8	1.1
22:00	1.4	1.3	1.1	1	4.9	5.4	5.5	5.6	5.5	4.9	1	1.3
23:00	1.5	1.4	1.2	1.1	4.7	5.2	5.3	5.4	5.3	4.7	1.1	1.4

Table 7: Daily Cooling & Heating Loads for each month using HAP - Cairo University rock laboratory.

For Cooling Loads

For Heating Loads

Operating hours (7:00 – 22:00) [16 hours]

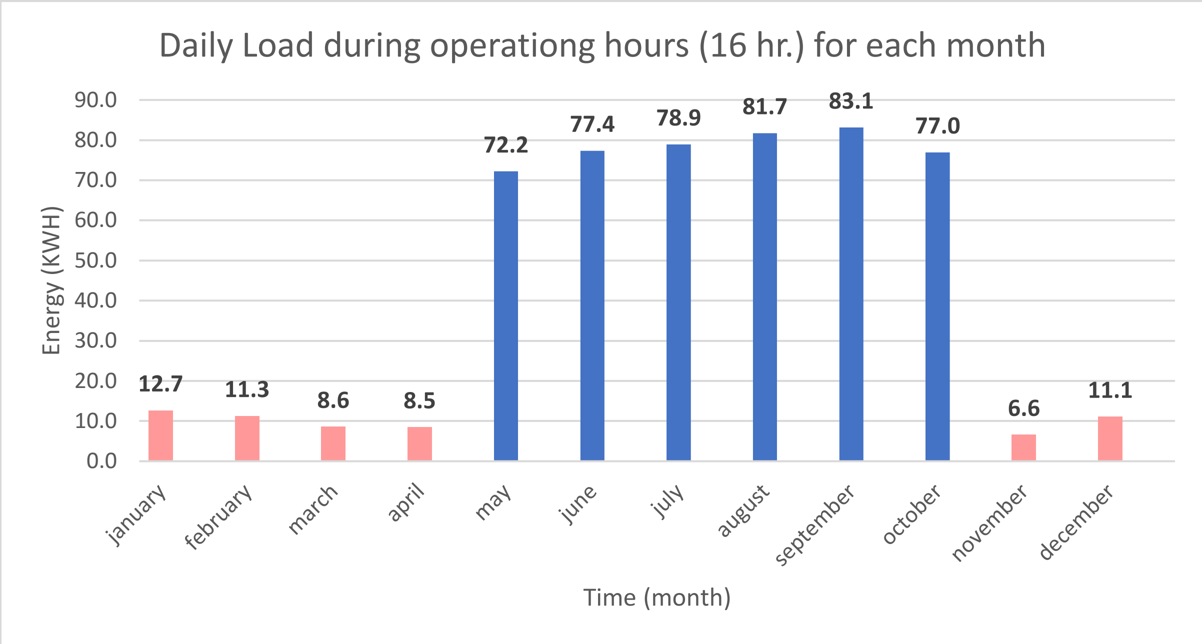


Figure 32: Daily Load during operating hours (16 hr.) for each month

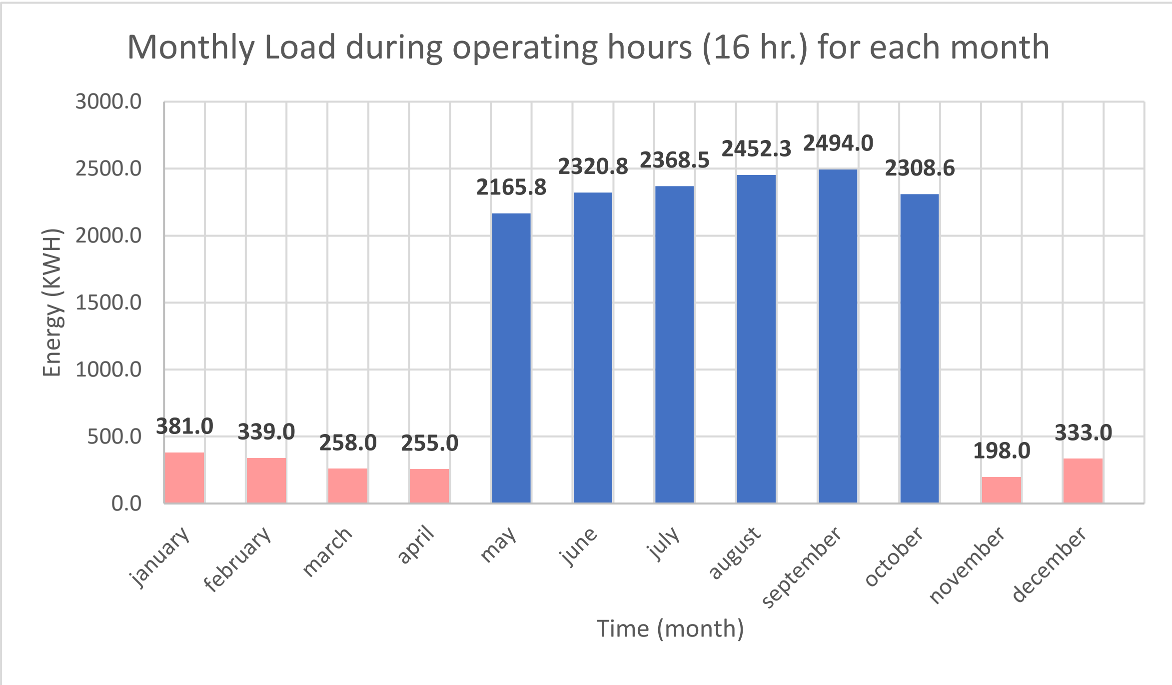


Figure 31: Monthly Load during operating hours (16 hr.) for each month

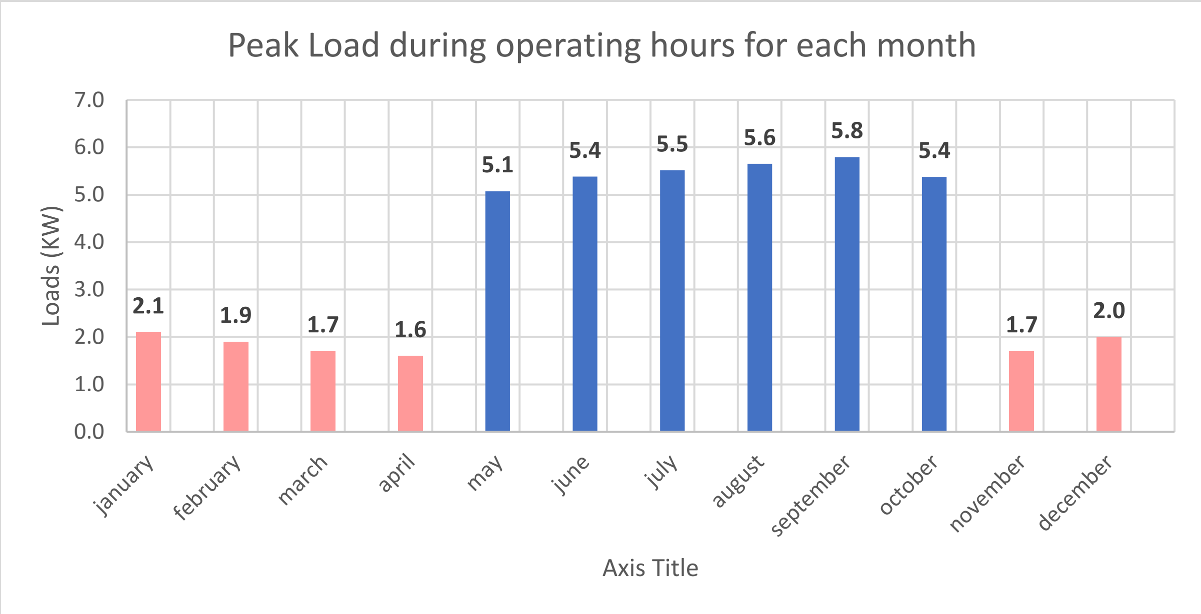


Figure 33: Peak Load during operating hours for each month

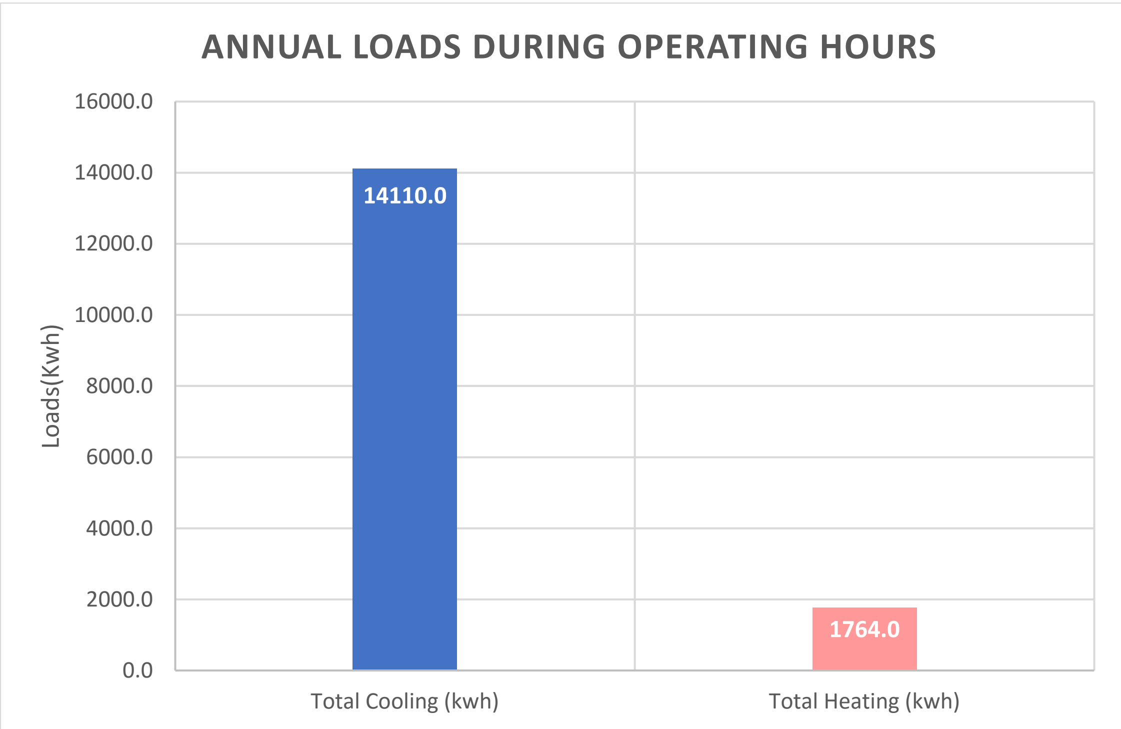


Figure 34: Annual Loads during operating hours - Cairo University rock laboratory

Therefore, from Figure (33):

$$\therefore\ Total\ Annual\ Energy\ Required = 14110 + 1764 = \mathbf{15874}\ \boldsymbol{KWH}$$

$$\therefore Total\ Energy\ Required\ in\ 30\ years\ (KWH) =\ \mathbf{15874 \times 30 = 476,220}\ \boldsymbol{KWH}$$

- ***Heat Pump Selection:***

ecoGEO⁺ B/C 1-6 PRO

R290

- **Modulating thermal power control** within a wide range (12,5-100%) and modulating flow rate control of both brine and production circuits (20-100%).
- **Natural refrigerant R290** : GWP 3.
- **Inverter technology.**
- **Compact design** including brine and production circulation pumps, brine and production expansion vessels (8l and 12l respectively), brine and production safety valves and DHW three-way valve.
- **Integrated management of up to 3 different emission temperatures, 2 buffer tanks** (heating and cooling), **1 DHW tank, 1 pool and hourly control of DHW recirculation.**
- **Integrated management of aerothermal collection modulating units,** in case of air source or hybrid configurations.
- **Integrated management of external On/Off or modulating auxiliary systems,** such as electrical heaters, On/Off boilers or modulating boilers.
- **Integrated free cooling in models 2 and 4.**
- **Integrated active cooling in models 3 and 4.**
- **Single-phase version available.**
- **Integrated photovoltaic hybridisation.**
- **Integrated energy meters** to measure the electrical consumption, the heating/cooling thermal power, the COP and the monthly and annual SPF.

SPECIFICATIONS ecoGEO⁺ B/C 1-6 PRO		UNITS	B1/C1	B2/C2	B3/C3	B4/C4
APPLICATION	Place of installation	-	Indoors			
	Type of brine system [1]	-	Ground source / Air source / Hybrid source			
	DHW, Heating and Pool	-	✓	✓	✓	✓
	High Temperature Recovery (HTR) system option	-	-	-	-	-
	Integrated Active cooling	-	-	-	✓	✓
	Integrated Passive cooling	-	-	✓	-	✓
PERFORMANCE	Modulation range of the compressor	%	12,5 to 100			
	Heating power output [2], B0W35	kW	1,0 to 6,0			
	COP [2], B0W35	-	4,3			
	Active cooling power output [2], B35W7	kW	-		1,0 to 6,0	
	EER [2], B35W7	-	-		4,4	
	Max. DHW temperature without / with support [5]	°C	75 / 80			
	Noise power emission level [6]	db	33 to 44			
	Energy label / ηs / SCOP W35 average climate control	-	A+++ / 182% / 4,64			
	Energy label / ηs / SCOP W55 average climate control	-	A++ / 140% / 3,60			
OPERATION LIMITS	Distribution / Set heating outlet temperature range	°C	10 to 75 / 20 to 75			
	Distribution / Set cooling outlet temperature range	°C	5 to 35 / 7 to 25			
	Brine inlet temperature range in heating applications	°C	-25 to 35			
	Brine inlet temperature range in cooling applications	°C	10 to 75			
	Minimum / Maximum refrigerant circuit pressure	bar	0,5 / 32			
	Production / Pre-load circuit pressure	bar	0,5 to 3,0 / 1,5			
	Brine / Pre-load circuit pressure	bar	0,5 to 3,0 / 0,7			
	Volume / Max. DHW storage tank pressure (ecoGEO⁺ C)	l / bar	165 / 8			
WORKING FLUIDS	R290 Refrigerant load	kg	0,15			
	Compressor oil type / load	kg	PZ46M / 0,3			
CONTROL ELECTRICAL DATA	1/N/PE 230 V / 50-60 Hz [8]	-	✓			
	Maximum recommended external protection [9]	-	-			
	Transformer primary circuit fuse	A	0,5			
	Transformer secondary circuit fuse	A	2,5			
ELECTRICAL DATA: SINGLE-PHASE	1/N/PE 230 V / 50-60 Hz [8]	-	✓			
	Maximum recommended external protection [9]	-	C16A			
	Maximum consumption [2], B0W35	kW / A	1,6 / 6,8			
	Maximum consumption [2], B0W55	kW / A	2,0 / 8,6			
	Minimum / Maximum starting current [7]	A	0,6 / 1,8			
	Correction of cosine Ø	-	0,96 / 1			
DIMENSIONS/WEIGHT	Height x width x depth	mm	ecoGEO⁺ B: 1058x550x602 · ecoGEO⁺ C: 1851x600x720			
	Empty weight (without assembly)	kg	B 125 · C 186	B 133 · C 194	B 125 · C 186	B 133 · C 194

Figure 35: Eco-Forest GSHP (1-6 Pro) Data specifications (for Cairo University Case study)

▪ ***<u>System type:</u>***

A closed loop Ground Source Heat pump is used which is a Water-to-Air heat pump. Operational Chart can be found in Appendix A.

Figure 36: Water-to-Air GSHP

3.1.2.2 Closed Loop GSHP Design:

Depending on the geographic area of Cairo University rock laboratory and where it is located, decisions were made to *build a vertical GSHP* since it will require only less space and will not by influenced by climate variations.

- ***<u>From Heating and cooling loads calculations:</u>***
  - Cooling block load (q_{lc}) $=$ 6.8 kW
  - Heating block load (q_{lh}) $=$ 2 kW
  - Design month (September) part-load factor (PLF_m) = 0.31 (From ASHRAE specifications)

- ***<u>From Pipe Selection for ground source heat pumps:</u>***
  - **Size:** Vertical U-tube = 32 mm, 152 mm (0.152 m) borehole diameter
  - **Material:** DR 11, HDPE

- ***<u>From Pipe Spacing:</u>***
  - 2 × 2 square grid (4 vertical bores) with 7 m bore-to-bore separation.

- ***<u>From Eco- Forest Heat pump input & output:</u>***
  - Heat pump ELT = 24°C
  - Assuming flow rate (3 $L/min.kW$):
    - **In Cooling:** $\Delta T_{Cooling} = 11.6\ °C\ , ELT < LLT$
    - **In Heating:** $\Delta T_{heating} = 6.3\ °C\ , ELT > LLT$
  - Heat pump LLT = 35.6°C (Cooling Mode)
  - Heat pump LLT = 17.7°C (Heating Mode)

- ***From Eco- Forest Graphs:***
  - Heat pump cooling efficiency $(EER) = 5.5\ Btu/Wh$
  - Heat pump heating efficiency $(COP_h) = 6.3$
  - Heat pulse analysis: Twenty year (7300 day), one month (30 day), & four hour (0.167 day) (assumed)
- ***For Equivalent Full-Load Cooling and Heating hours:***
  - $EFLH_c = 900\ h$ (assumed)
  - $EFLH_h = 450\ h$ (assumed)
- ***A thermal property test provided the following information for soil data:***
  - Ground temperature $(t_g) = 24.5°C$
  - Ground conductivity $(k_g) = 2.5\ W/m \cdot K$
  - Ground diffusivity $(\alpha_g) = 0.892\ mm^2/s = 0.077\ m^2/day$
  - Borehole fill conductivity $(k_b) = 2.5\ W/m \cdot K$

3.1.2.2.1 Governing Equations: (Vertical GSHP)

For Cooling mode:

Determine the ground heat transfer rates in cooling and heating and net annual heat to and from the ground:

$$q_{cond} = q_{lc} \times \frac{EER + 3.412}{EER} = 6.8 \times \frac{5.5 + 3.412}{5.5} = 11.018\ kW\ (-ve)$$

$$q_{evap} = q_{lc} \times \frac{COP_h - 1}{COP_h} = 2 \times \frac{6.3 - 1}{6.3} = 1.683\ kW\ (+ve)$$

$$q_a = \frac{q_{cond} \times EFLH_c + q_{evap} \times EFLH_h}{8760\ (hr.)} = \frac{-11.018 \times 900 + 1.683 \times 450}{8760} = -1045.53\ W$$

Determine the thermal resistances of the ground for the three prescribed heat pulses (Using Equations):

$$F_{o_f} = \frac{4\ \alpha_g\ \tau_f}{d^2} = \frac{4 \times (0.077) \times 7330.167}{(0.152)^2} = 97718.64$$

Then, from Fig. (14) $\rightarrow G_f = 0.98$

$$F_{o_1} = \frac{4\ \alpha_g\ (\tau_f - \tau_1)}{d^2} = \frac{4 \times (0.077) \times (7330.167 - 7300)}{(0.152)^2} = 402.16$$

Then, from Fig. (14) $\rightarrow G_1 = 0.54$

$$F_{o_2} = \frac{4\ \alpha_g\ (\tau_f - \tau_2)}{d^2} = \frac{4 \times (0.077) \times (7330.167 - 7330)}{(0.152)^2} = 2.226$$

Then, from Fig. (14) $\rightarrow G_2 = 0.18$

Then:

$$R_{ga} = \frac{G_f - G_1}{k_g} = \frac{0.98 - 0.54}{2.5} = 0.176\, m.k/W$$

$$R_{gm} = \frac{G_1 - G_2}{k_g} = \frac{0.54 - 0.18}{2.5} = 0.144\, m.k/W$$

$$R_{gst} = \frac{G_2}{k_g} = \frac{0.18}{2.5} = 0.072\, m.k/W$$

Determine the thermal resistances of the bore. Using the equation in Table (1) to find the estimated flow through each U-tube during cooling (loop transfers $q_{cond} = 11.018\ kW$)

To estimate loop water flow:

$$L/min \approx q\ (kW)\ \div\ [0.0692\ \times\ \Delta t\ (°C)\ \times\ No.of\ ParalleI\ U - Tubes]$$

$$Flow/U_{tube}\ (L/min.) = \frac{-11.018}{(0.0692 \times (24 - 35.6) \times 4)} = 3.43\ L/min.$$

At 20°C, the Reynolds number (Re) for a water fluid flowing at 10 L/min in a (D = 32mm). DR 11 tube is 7769.

Using interpolation, Re will be just over 2664.767 at 3.43 L/min, which is a transition flow.

$$\frac{3.43 - 10}{0 - 10} = \frac{Re - 7769}{0 - 7769}$$

$$\therefore Re = 2664.767$$

So, the bore resistance will be found based on the transition flow but the value for bore resistance will be interpolated between laminar and transition values. If the flow rate is adjusted during the final design phase, the results should be reconfirmed. Also note the 0.0692 multiplier for the equation above is based on water and the value for antifreeze solutions will be slightly lower, thus making the flow rate higher:

Tube Diameter and Dimension	Tube Location	Bore Diameter, mm	Thermal Resistance of Bore, m·°C/W — Fluid Reynolds No. = 2000, Grout Conductivity, W/m·°C			Fluid Reynolds No. = 4000, Grout Conductivity, W/m·°C			Fluid Reynolds No. = 10,000, Grout Conductivity, W/m·°C		
			0.70	1.40	2.10	0.70	1.40	2.10	0.70	1.40	2.10
25 mm DR 11 HDPE U-Tube	B	100	0.26	0.17	0.14	0.24	0.14	0.11	0.23	0.14	0.11
		125	0.29	0.18	0.15	0.26	0.16	0.12	0.26	0.11	0.12
	C	100	0.18	0.13	0.11	0.16	0.10	0.09	0.15	0.10	0.08
		125	0.19	0.13	0.11	0.17	0.11	0.09	0.16	0.10	0.08
	Double	125	0.16	0.10	0.08	0.14	0.08	0.06	0.14	0.08	0.06
32 mm DR 11 HDPE U-Tube	B	100	0.24	0.16	0.13	0.21	0.13	0.10	0.21	0.13	0.10
		125	0.26	0.17	0.14	0.23	0.14	0.11	0.23	0.14	0.11
		150	0.28	0.18	0.14	0.26	0.15	0.12	0.25	0.15	0.11
	C	100	0.17	0.12	0.11	0.15	0.10	0.08	0.14	0.09	0.08
		125	0.18	0.13	0.11	0.16	0.10	0.08	0.15	0.10	0.08
		150	0.19	0.13	0.11	0.17	0.11	0.09	0.16	0.10	0.08
	Double	125	0.15	0.09	0.07	0.13	0.08	0.06	0.13	0.08	0.06
		150	0.15	0.10	0.08	0.14	0.08	0.06	0.14	0.08	0.06
40 mm DR 11 HDPE U-Tube	B	125	0.24	0.16	0.13	0.22	0.13	0.11	0.21	0.13	0.10
		150	0.26	0.17	0.14	0.23	0.14	0.11	0.23	0.14	0.11
	C	125	0.17	0.12	0.11	0.15	0.10	0.09	0.14	0.09	0.08
		150	0.18	0.13	0.11	0.16	0.11	0.09	0.15	0.10	0.08
	Double	150	0.14	0.09	0.07	0.13	0.08	0.06	0.13	0.08	0.06

Table 8: Thermal Resistances of Bores with U-Tubes for Various Conditions

$$For\ k_{grout}\ =\ 2.5\ W/m \cdot K, laminar\ flow, 152\ mm, location\ B{:}\ R_b\ =\ 0.138\ m \cdot K/W$$

$$For\ k_{grout}\ =\ 2.5\ W/m \cdot K, transition\ flow, 152\ mm, location\ B{:}\ R_b\ =\ 0.118\ m \cdot K/W$$

$$For\ k_{grout}\ =\ 2.5\ W/m \cdot K, laminar\ flow, 152\ mm, location\ C{:}\ R_b\ =\ 0.108\ m \cdot K/W$$

$$For\ k_{grout}\ =\ 2.5\ W/m \cdot K, transition\ flow, 152\ mm, location\ C{:}\ R_b\ =\ 0.088\ m \cdot K/W$$

The average bore resistance is:

$$R_b\ = \frac{0.138 + 0.118 + 0.108 + 0.088}{2} = 0.113\ m \cdot K/W$$

For location BC, $k_{grout}\ =\ 2.5\ W/m \cdot K$, transition flow, 152 mm bore

The ground-loop differential temperature is $\Delta T_{Cooling} = 11.6\ °C$, thus the short-circuiting heat loss factor [$Fsc\ is$ 1.04] according to the following figure:

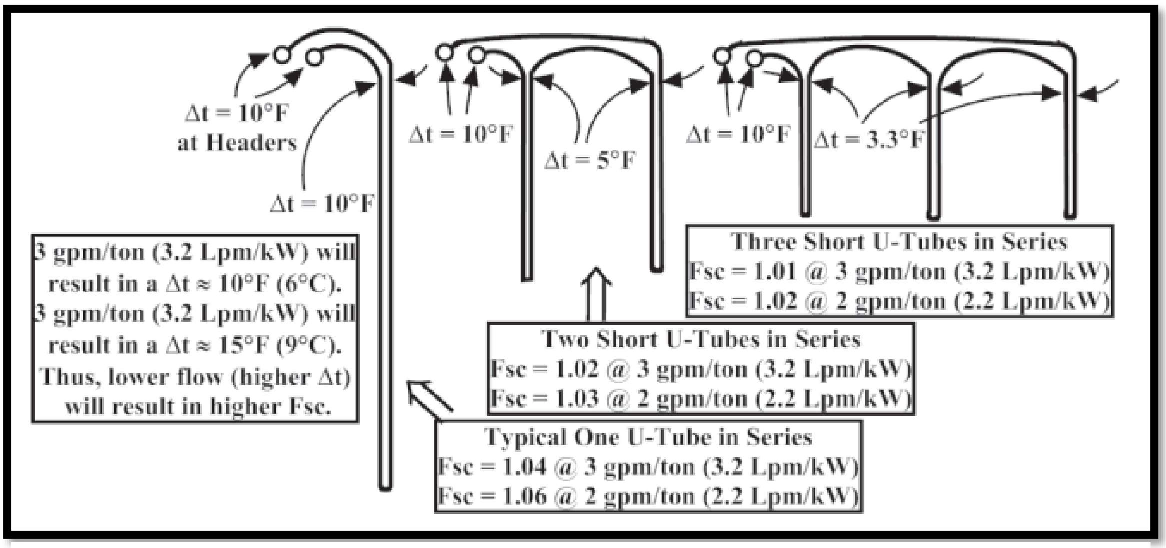


Figure 37: Short-Circuit Factor (Fsc) for Standard and Shallow Bore U-Tube Applications

The required total bore length for cooling is computed ***using Equation (2)***. The procedure for determining long-term ground temperature change (t_p) is assumed to be a value of –0.5°C.

$$L_{cooling} = \frac{q_a\, R_{ga} + q_{cond}\,\left(R_b + PLF_m\, R_{gm} + F_{sc}\, R_{gst}\right)}{t_g - \dfrac{ELT + LLT}{2} + t_p}$$

$$L_{cooling} = \frac{(-1045.53 \times 0.176) - 11.018 \times 10^3\ (0.113 + 0.31 \times\ 0.144 + 1.04 \times 0.072)}{24.5 - \dfrac{24 + 35.6}{2} - 0.5}$$

$$L_{cooling} = 473.43\ ft = 144\ m\ /\ 4\ bores\ = 36\ m/\ bore$$

3.1.2.2.2 GHX_Design_Toolbox Results:

From this software, results have shown that the required total bore length for cooling is $140\ m$ and by that validating the software works according to the previous governing equations. Details are found in Appendix A.

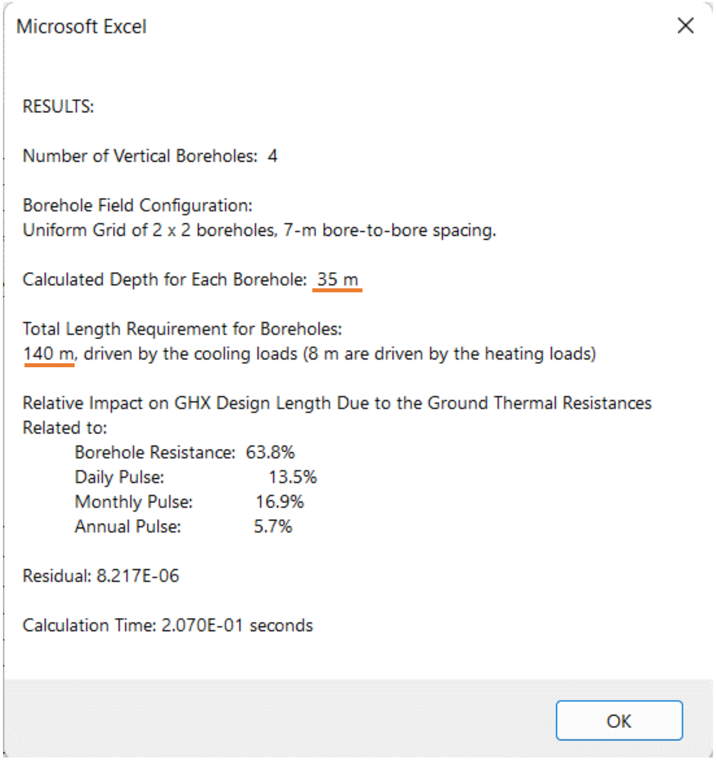
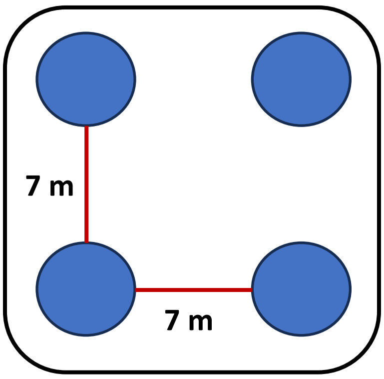


Figure 38: GHX_Design_Toolbox Results – Cairo University Case Study

Therefore, the error percentage of the results between the [GHX_Design_Toolbox](#) software and the governing equations can be evaluated, where:

$$\%\ Error = \frac{140 - 144}{140} \times 100 = -2.86\ \%$$

- ***Borehole field Configuration:***
  - Uniform Grid of 2 x 2 boreholes with 7 m bore-to-bore spacing.
  - Each borehole 35 m depth.
  - Land Area Required $= 7 \times 7 = 49\ m^2$



3.1.2.3 Cost Assessment

- Geothermal heat Pump cost evaluation:

i. Drilling Cost:

Drilling cost		
Cost / m (LE/m)	Total drilling length (m)	Total drilling cost (LE)
1300	140	182,000

ii. Unit Cost:

Unit cost			
Cost (Euro)	No. Of Units	Cost (LE)	Total cost (LE)
8126.71	1	276,308.14	276,308.14

Then, the initial cost is:

$$Initial\ Cost = 182{,}000 + 276{,}308.14 = \mathbf{458{,}308.14}\ \text{LE}$$

iii. Operation & Maintenance Cost:

Operation & Maintenance Cost			
Electricity tariff (LE)	Elec. Cons. (KWH/Yr.)	Total Op Cost (LE)	Maintenance Cost per year (LE)
1.45	3334.7	4835.3	2000

For more details about the electrical consumption will be found in Appendix A.

Then, the Operation & Maintenance cost is:

$$\text{Operation \& Maintenance } Cost\ per\ Year = 4835.3 + 2000 = \mathbf{6835.3}\ \text{LE}$$

Figure 39: GSHP Specifications (for Cairo University Case study)

- Air-Conditioning cost evaluation:

Selecting SHARP AC units:

SHARP	*Cost*
1.5 hp	20,450 EGP
3 hp	27,500 EGP
4 hp	42,610 EGP

- **(Sharp AY-XP24UHE Cool & Heat Digital split AC with plasma cluster – 3HP)** Is chosen according to Area and thermal loads of the site.

- ***From its Data sheet:***

Brand	*Sharp*
Model No.	***AY-XP24UHE***
Cooling Capacity	***24000 BTU = 7.03 kw***
Heating Capacity	***26000 BTU = 7.65 kw***
Wattage	***2.1 Kw***

i. **Initial Cost:**

Initial cost	
Capital cost	27,500 LE
Installation	1,000 LE
No. of AC Units	1
Total AC units' cost (LE)	28,500 LE

Then, the initial cost is:

$$Initial\ Cost = \mathbf{28,500}\ \text{LE}$$

ii. **Operation & Maintenance Cost:**

Operation & Maintenance Cost			
Electricity tariff (LE)	Elec. Cons. (KWH/Yr)	Total Op Cost (LE)	Maintenance Cost per year (LE)
1.45	7400	10,730	4,500

Then, the Operation & Maintenance cost is:

$$\mathbf{Operation\ \&\ Maintenance}\ \boldsymbol{Cost\ per\ Year} = \mathbf{10,730 + 4,500 = 15,230\ LE}$$

➢ Payback period:

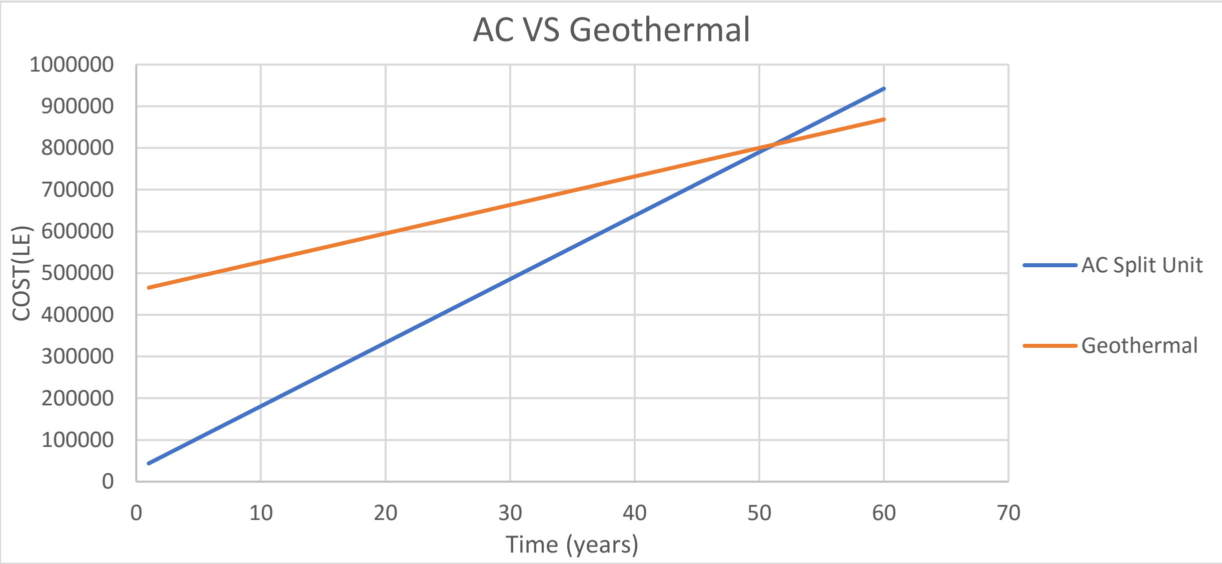


Figure 40: Payback period (Ac vs GSHP) – Cairo University Case study

- ***The payback period is 51 years*** if installing Geothermal heat pumps instead of conventional air conditioning. (Not feasible)

3.1.2.4 CFD Simulation

- ***Case Study: Flow through a U-tube pipe (For Cooling Mode Only)***

According to the no. of boreholes built at Cairo university, it was shown that the length of the bore hole was 40 m.

- This modelling shows a pipe with the following specifications:

No.	Pipe Parameters	Value
1	*Outside Diameter*	$42.16\ mm$
2	*Inner Diameter*	$36.16\ mm$
3	*Thickness*	$3\ mm$
4	*Length*	$40\ m$

- From analytical calculations:

No.	Fluid Parameters	Value
1	*Material*	*Water (Built-in)*
2	*Volume Flow Rate*	$4.18\ L/min$
3	*Laminar Flow and Steady State*	

__Geometry: (Using Autodesk: Fusion 360)__

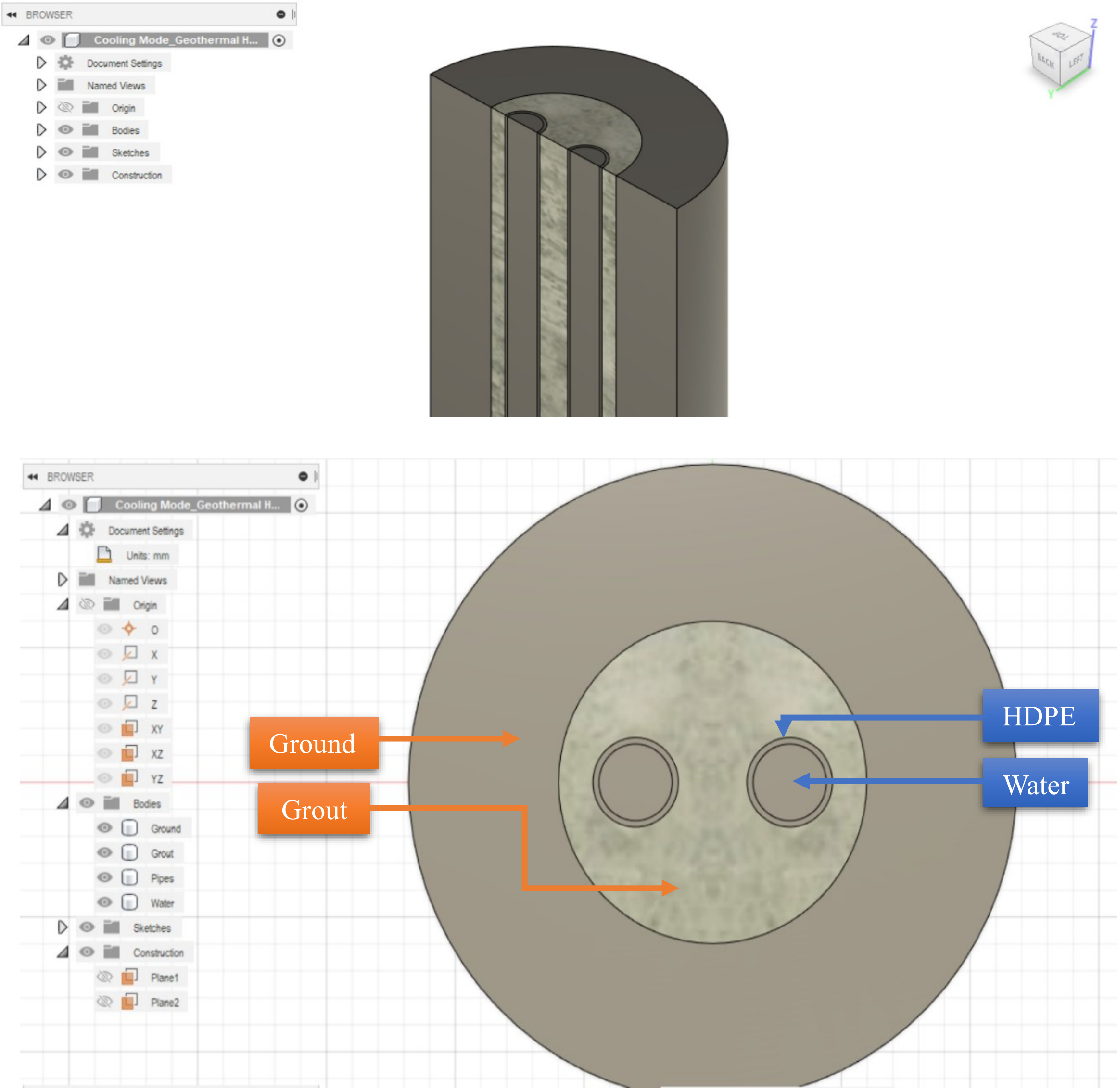


Figure 41: CAD Geometry - Cairo University Case Study

Boundaries: (From Figure 42)

Name	*Type*	*No. Assigned*
Inlet 1	***Boundary***	***1***
Outlet 1	***Boundary***	***2***
Wall 1	***Boundary***	***3***

Input Values:

Geometry in 3D space	
- ***Material: Fluid***	***Water (Built-In)***
- ***Velocity at inlet***	$\boldsymbol{u = 0.0683\ m/s}$
- ***Temperature at inlet***	$\boldsymbol{T_{in} = 29.6\,°C}$
- ***Average Pressure at outlet***	$\boldsymbol{P_{av} = 0}$, $(\boldsymbol{Fully\ developed})$

Geometry: (In COMSOL)

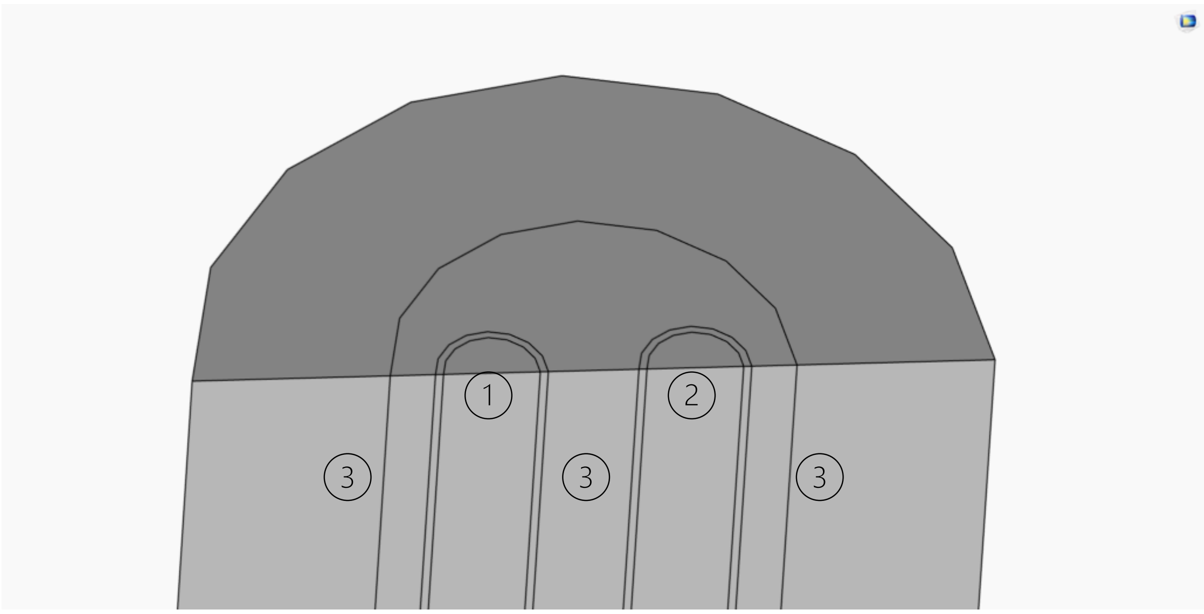


Figure 42: Single U-tube Pipe Geometry in COMSOL - Cairo University Case Study

Boundary Conditions:

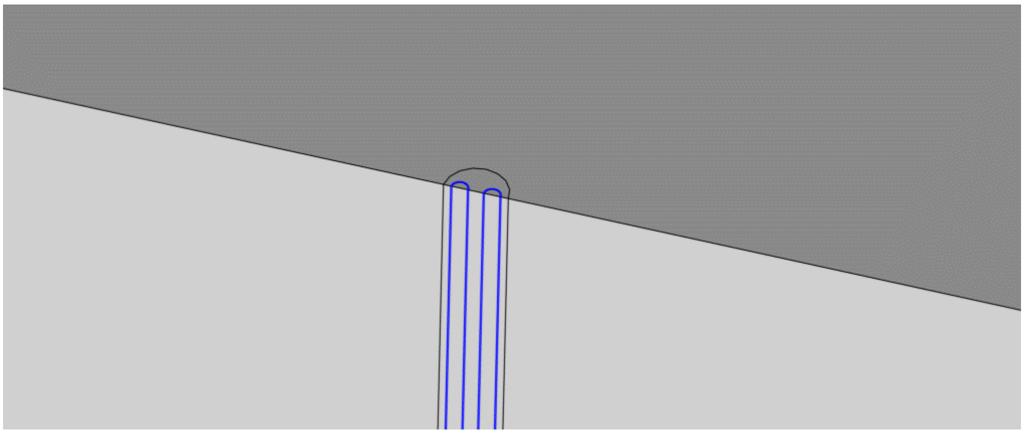


Figure 43: Pipe Boundary Conditions – Cairo University Case Study

- Wall Conditions: No slip conditions

Meshing:

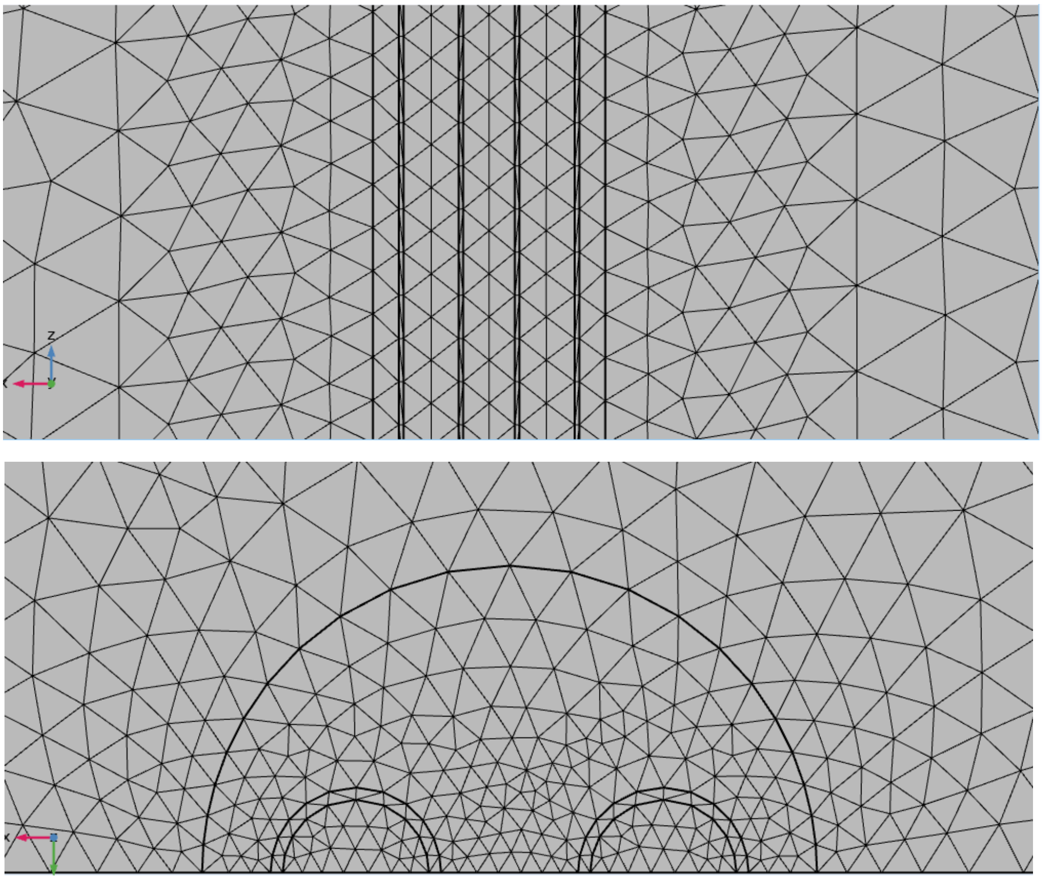


Figure 44: Pipe Meshing – Cairo University Case Study

- Element Size: Finer

Results:

- Temperature: (3D Plot Group)

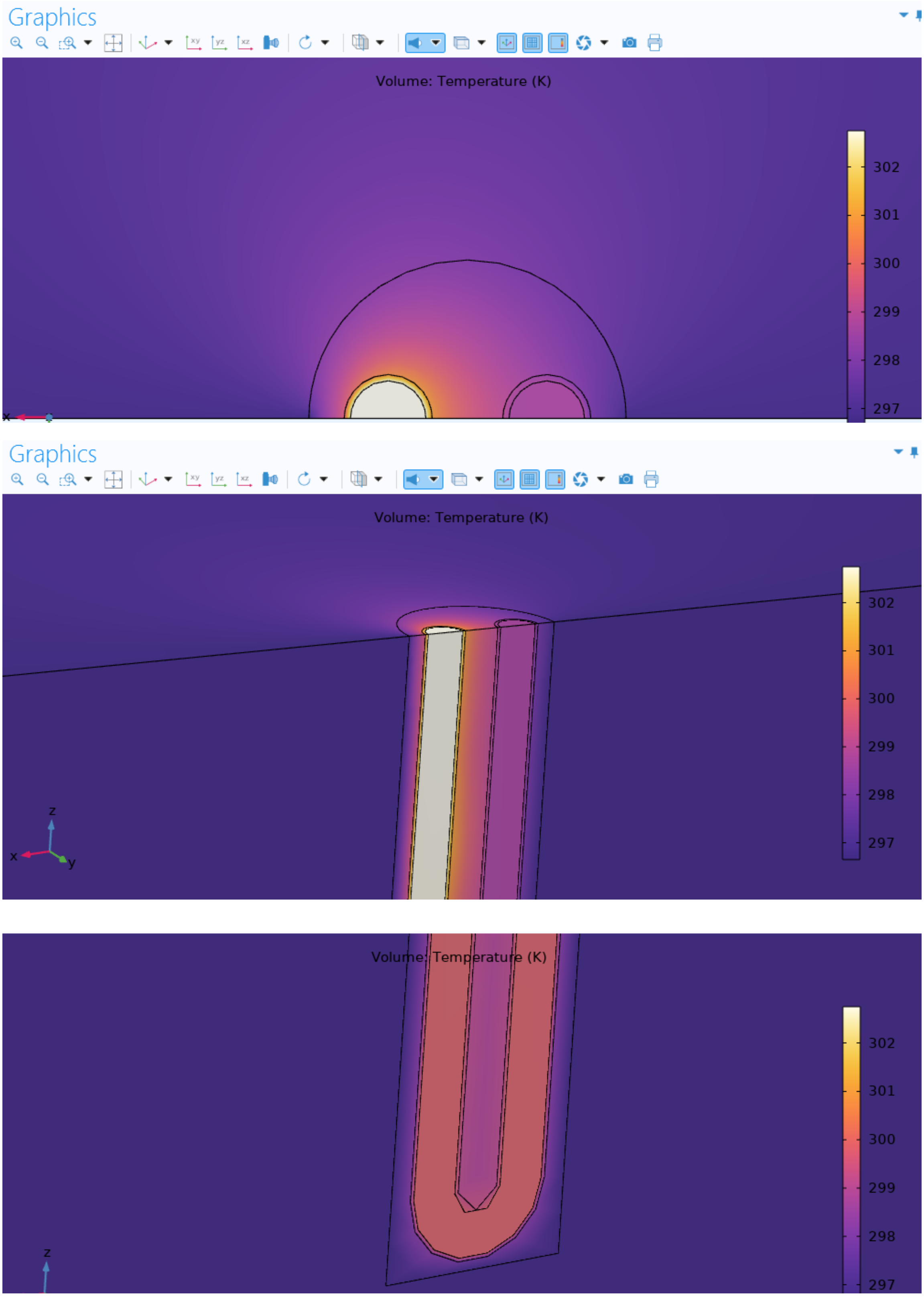


Figure 45: 3-D surface plot for Temperature distribution – Cairo University Case Study

- Velocity: (3D Plot Group)

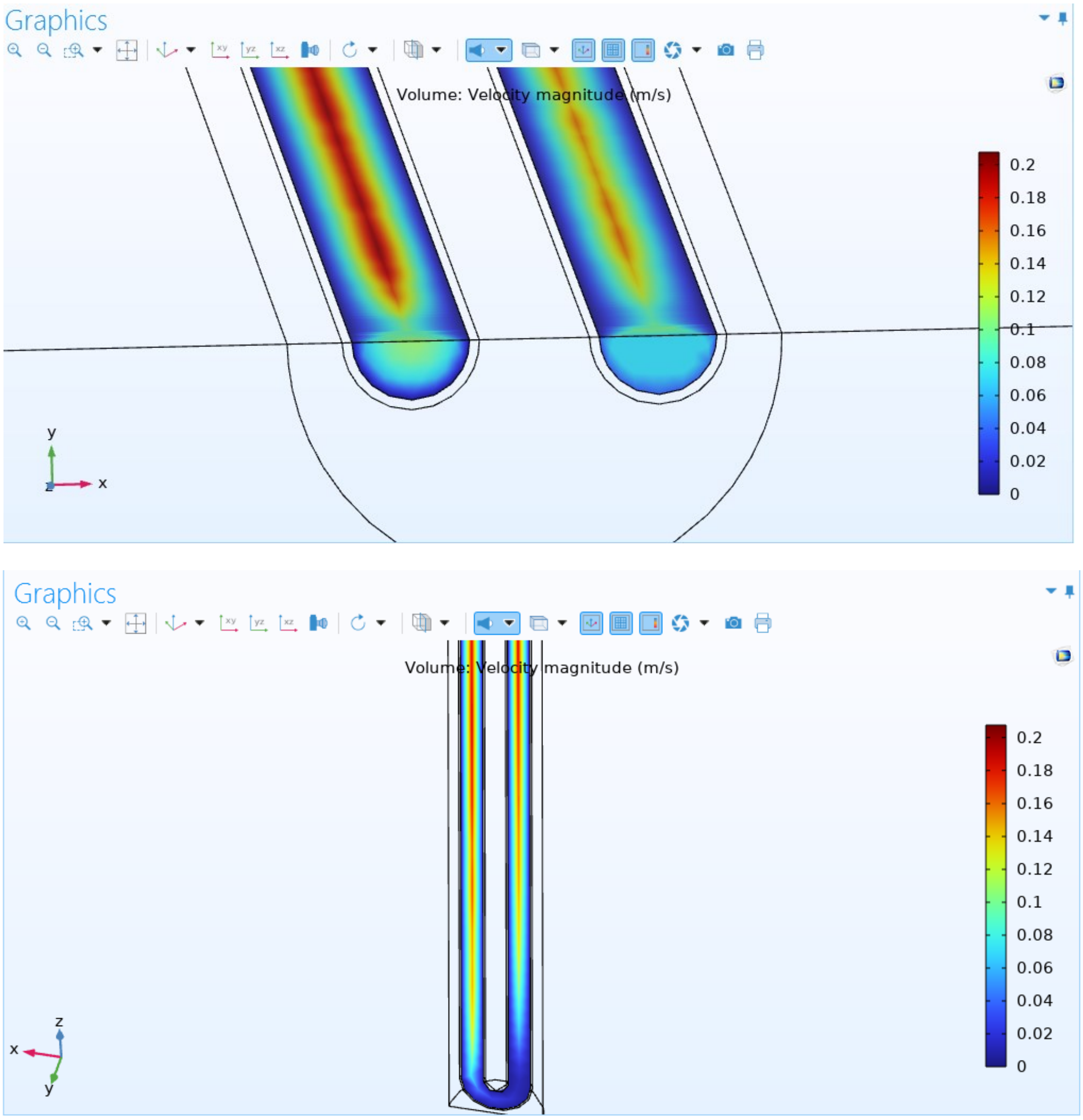


Figure 46: 3-D plot group for velocities distribution – Cairo University Case Study

- ***Simulation time:*** [6 Minutes 33 seconds]

The velocity profile is to observe the flow along the pipe which affect the heat transfer by convection along the pipe.

Model Validation:

To validate our model, check the flow rate of fluid:

$$Q_{in} = Q_{out}\ (For\ a\ half\ symmetric\ Geometry)$$

$$\therefore Q_{in} = 1.8294\ L/min$$

$$\therefore Q_{out} = 1.8314\ L/min$$

$$\therefore Q_{in} \approx Q_{out} = 2 \times 1.8314 = 3.6628\ L/min.$$

From a steady state analysis:

$$\therefore T_{out} = 25.53\ °C\ \text{(From simulation)}$$

$$\therefore\ T_{ground} = 23.5°C$$

$$\therefore \boldsymbol{Effectiveness\ (\theta)} = \frac{T_{in} - T_{out}}{T_{in} - T_{ground}} = \frac{29.6 - 25.53}{29.6 - 23.5} \times 100 = \boldsymbol{66.72}\ \%$$

Comment:

To improve the model simulation results (Factors affecting T_{out}), we can:

1. Adjust the meshing to get the exact volume flow rate of the fluid.
2. Make sure of the software database, for example:
   Knowing the exact value of the thermal conductivity of solids for accurate heat transfer results.

3.1.2.5 Life Cycle Assessment

3.1.2.5.1 Input Data

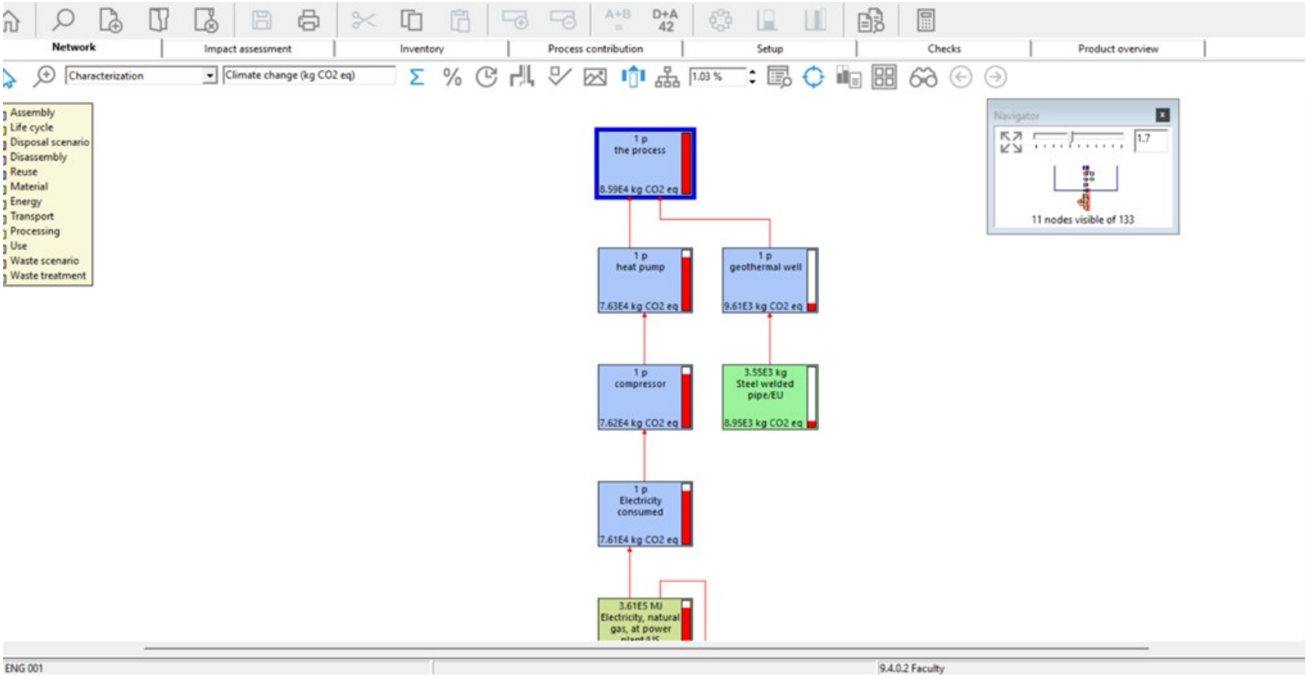


Figure 48: Flowchart of Geothermal Heat pump LCA

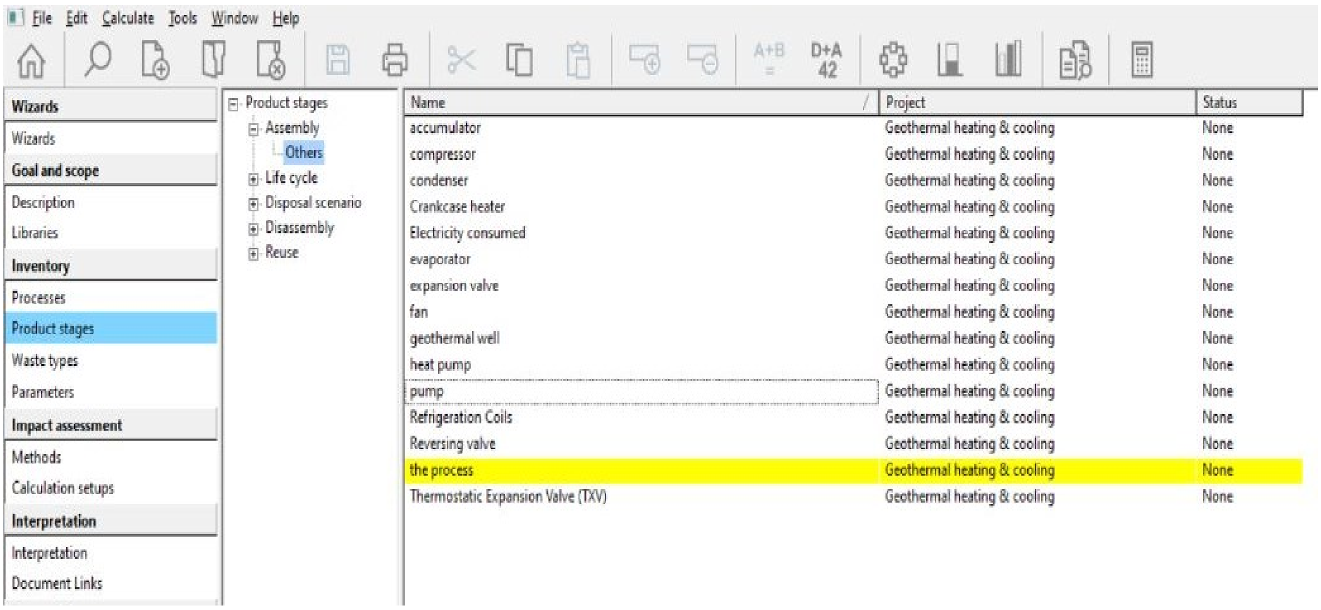


Figure 47: LCA Product Stages of GSHP

Table 9: Life Cycle Assessment Input Data

		Material/ Assemblies	Amount	Unit
Heat Pump	***Accumulator***	Aluminum, primary, cast alloy slab from continuous casting	0.0458	kg
	Compressor	Cold rolled sheet, steel, at plant/RNA	5.31	kg
		Cast iron {GLO}| market for | APOS, S	1.83	kg
		Iron sinter market for iron sinter | APOS, S	0.78	kg
		Aluminum, secondary, shape casted/RNA	0.27	kg
		Steel, low-alloyed {GLO}| market for | APOS, S	5.43	kg
		Copper, anode {GLO}| market for copper, anode | APOS, S	1.745	kg
		Electricity consumed	1	P
	Condenser	Brass {CH}| market for brass | APOS, S	0.855	kg
	Crankcase heater	Aluminum. Primary, cast alloy slab from continuous casting {GLO}| market for | APOS, S	0.03	kg
	Electricity consumed	Electricity, natural gas, at power plant/US	100041.08	kWh
	Evaporator	Copper, anode {GLO}| market for copper, anode | APOS, S	0.713	kg
	Fan	Aluminum, primary, cast alloy slab from continuous casting {GLO}| market for copper, anode | APOS, S	0.304	kg
	Water Pump (0.5kw)	Water pump, 22kW {GLO}| market for water pump, 22kW | APOS, S	0.02272727	p
	Refrigeration Coils	Copper, anode {GLO}| market for copper, anode | APOS, S	7.15	kg
		Refrigerant R134a {GLO}| market for | APOS, S	0.15	kg
	Reversing valve	Copper, anode {GLO}| market for copper, anode | APOS, S	0.413	kg
	Thermostatic Expansion Valve (TXV)	Brass {CH}| market for brass | APOS, S	0.19592	kg
Geothermal well	***Steel pipe***	Steel welded pipe/EU	3553.47	kg
	water	Tap water {BR} market for tap water | APOS, S	193.53	kg
	HDPE pipe	HDPE pipes E	89.95	kg
	Grout	Concrete block {BR} market for Concrete block| APOS, S	8512.79	kg

Note: Screenshots are found in Appendix A

3.1.2.5.2 Results

From the following figures, the two processes (heat pump – geothermal well) are being put in comparison to see which have the most contribution in each category. In this scope, Natural gas extraction, processing and power plant consumption operations are contributed for most of the emmsions in the LCA study.

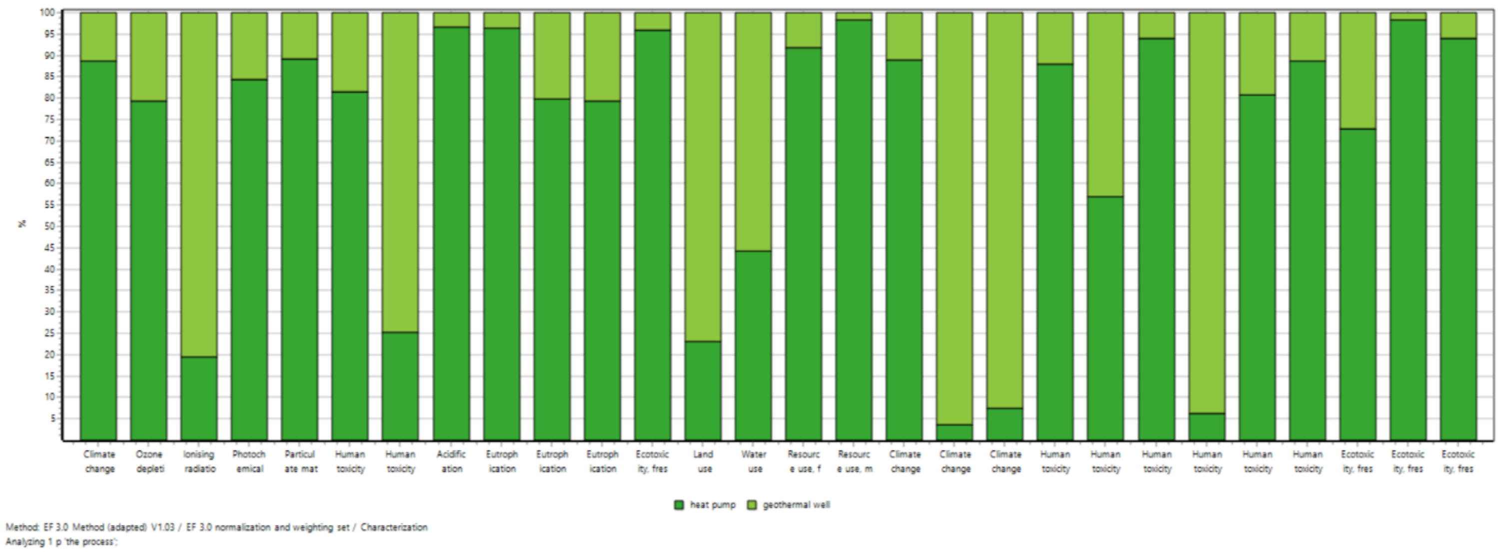


Figure 49: Comparative analysis of impact indicators according to the Environmental Footprint v3.0 E/A/Characterization

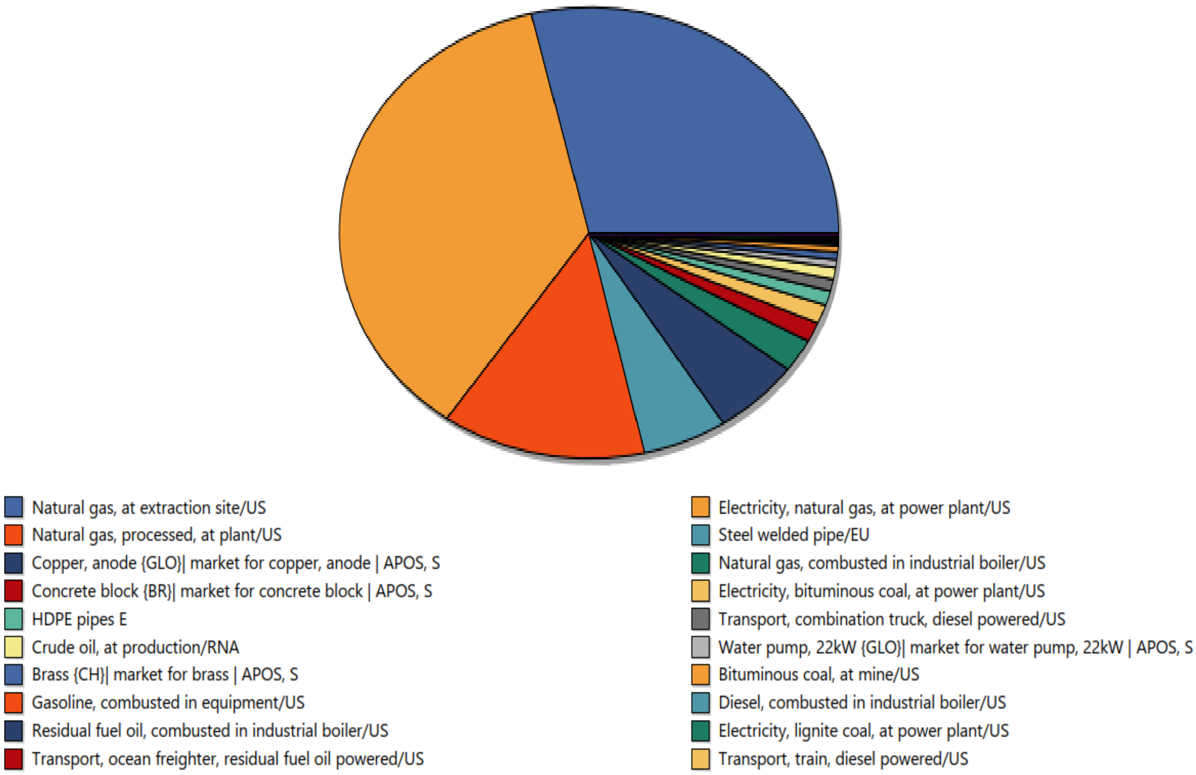


Figure 50: Process Contribution - Pie Chart

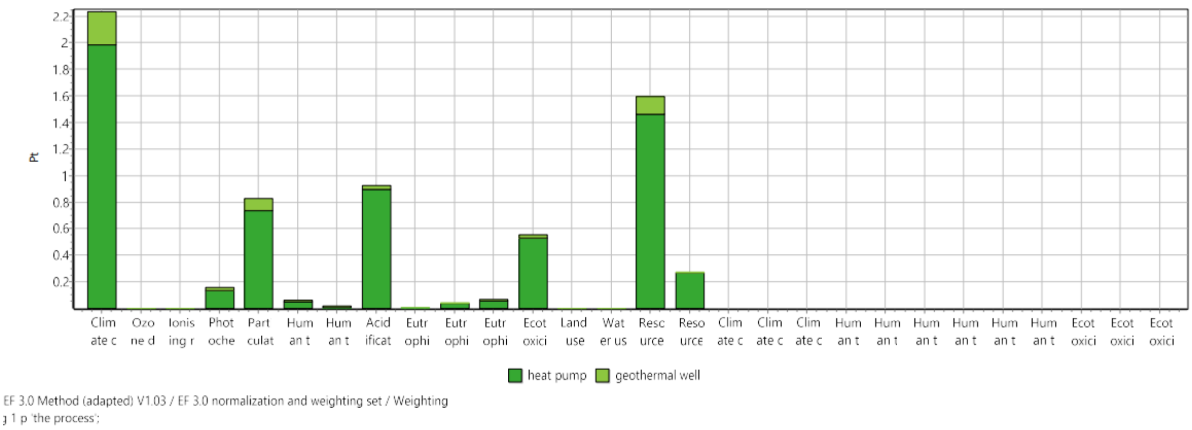


Figure 52: Comparative between two processes to EF 3.0 Method (adapted) V1.03/EF 3.0 normalization & weighting set /Weighting

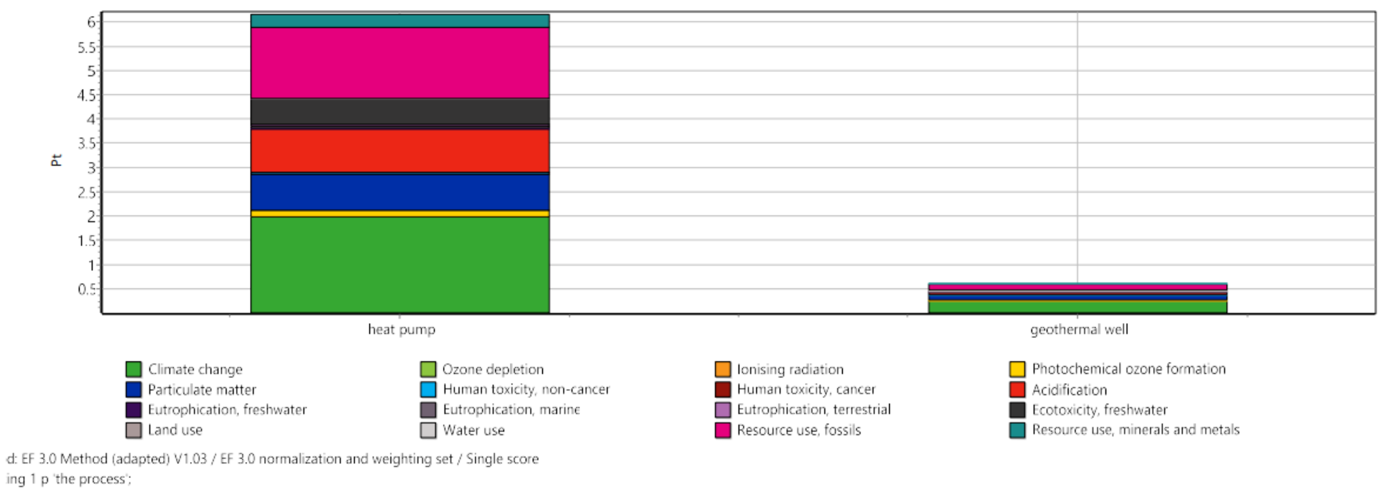


Figure 51: Comparative between two processes to EF 3.0 Method (adapted) V1.03/EF 3.0 normalization & weighting set /Single Score

- ***<u>Effect on Climate Change:</u>***

  - Total CO_2 produced = 71,900 kg CO_2 ,While the rest of emissions in (kg CO_2 $eq.$) = 14,000 kg CO_2 $equivalent.$
  - Total heating load injected / produced from the ground= 565,920 kWhth

$$\text{GHG produced} = \frac{71{,}900 + 14{,}000}{565920} = \mathbf{0.1517}\ \boldsymbol{kgCO2eq./kWhth}$$

- ***The most 5 processes contributing to each category in the environmental impact assessment:***

1. *Climate change:*

No	Process	Project	Unit	Total	heat pump	geothermal well
	Total of all processes		kg CO2 eq	8.59E4	7.63E4	9.61E3
1	Electricity, natural gas, at power plant/US	USLCI	kg CO2 eq	5.91E4	5.91E4	x
2	Natural gas, at extraction site/US	USLCI	kg CO2 eq	1.05E4	1.05E4	x
3	Steel welded pipe/EU	Industry data	kg CO2 eq	8.95E3	x	8.95E3
4	Natural gas, combusted in industrial boiler/US	USLCI	kg CO2 eq	3.13E3	3.13E3	x
5	Natural gas, processed, at plant/US	USLCI	kg CO2 eq	1.68E3	1.68E3	x

2. *Ozone depletion:*

No	Process	Project	Unit	Total	heat pump	geothermal well
	Total of all processes		kg CFC11 eq	0.000157	0.000125	3.24E-5
1	Refrigerant R134a {GLO}| market for | APOS, S	Ecoinvent 3 -	kg CFC11 eq	0.000118	0.000118	x
2	Concrete block {BR}| market for concrete block | APOS, S	Ecoinvent 3 -	kg CFC11 eq	3.24E-5	x	3.24E-5
3	Copper, anode {GLO}| market for copper, anode | APOS, S	Ecoinvent 3 -	kg CFC11 eq	3.29E-6	3.29E-6	x
4	Water pump, 22kW {GLO}| market for water pump, 22kW | APOS, S	Ecoinvent 3 -	kg CFC11 eq	2E-6	2E-6	x
5	Brass {CH}| market for brass | APOS, S	Ecoinvent 3 -	kg CFC11 eq	1.04E-6	1.04E-6	x

3. *Ionizing radiation:*

No	Process	Project	Unit	Total	heat pump	geothermal well
	Total of all processes		kBq U-235 eq	57.7	11.2	46.5
1	Steel welded pipe/EU	Industry data	kBq U-235 eq	31.9	x	31.9
2	Concrete block {BR}| market for concrete block | APOS, S	Ecoinvent 3 -	kBq U-235 eq	14.6	x	14.6
3	Copper, anode {GLO}| market for copper, anode | APOS, S	Ecoinvent 3 -	kBq U-235 eq	6.01	6.01	x
4	Water pump, 22kW {GLO}| market for water pump, 22kW | APOS, S	Ecoinvent 3 -	kBq U-235 eq	2.67	2.67	x
5	Brass {CH}| market for brass | APOS, S	Ecoinvent 3 -	kBq U-235 eq	1.68	1.68	x

4. *Photochemical ozone formation:*

No	Process	Project	Unit	Total	heat pump	geothermal well
	Total of all processes		kg NMVOC eq	136	115	21.5
1	Natural gas, processed, at plant/US	USLCI	kg NMVOC eq	50.4	50.4	x
2	Electricity, natural gas, at power plant/US	USLCI	kg NMVOC eq	50.4	50.4	x
3	Steel welded pipe/EU	Industry data	kg NMVOC eq	18.7	x	18.7
4	Transport, combination truck, diesel powered/US	USLCI	kg NMVOC eq	3.41	3.41	x
5	Natural gas, combusted in industrial boiler/US	USLCI	kg NMVOC eq	2.75	2.75	x

5. *Particulate matter:*

No	Process	Project	Unit	Total	heat pump	geothermal well
	Total of all processes		disease inc.	0.00549	0.00489	0.000593
1	Natural gas, processed, at plant/US	USLCI	disease inc.	0.00452	0.00452	x
2	Steel welded pipe/EU	Industry data	disease inc.	0.000554	x	0.000554
3	Electricity, natural gas, at power plant/US	USLCI	disease inc.	0.000275	0.000275	x
4	Electricity, bituminous coal, at power plant/US	USLCI	disease inc.	4.93E-5	4.93E-5	x
5	Concrete block {BR}| market for concrete block | APOS, S	Ecoinvent 3 -	disease inc.	2.97E-5	x	2.97E-5

6. *Human toxicity, non-cancer:*

No	Process	Project	Unit	Total	heat pump	geothermal well
	Total of all processes		CTUh	0.00073	0.000594	0.000136
1	Natural gas, at extraction site/US	USLCI	CTUh	0.000376	0.000376	x
2	Steel welded pipe/EU	Industry data	CTUh	0.000132	x	0.000132
3	Copper, anode {GLO}| market for copper, anode | APOS, S	Ecoinvent 3 -	CTUh	0.000117	0.000117	x
4	Electricity, natural gas, at power plant/US	USLCI	CTUh	6.4E-5	6.4E-5	x
5	Crude oil, at production/RNA	USLCI	CTUh	1.03E-5	1.03E-5	x

7. *Human toxicity, cancer:*

No	Process	Project	Unit	Total	heat pump	geothermal well
	Total of all processes		CTUh	1.45E-5	3.66E-6	1.09E-5
1	HDPE pipes E	Industry data	CTUh	9.66E-6	x	9.66E-6
2	Copper, anode {GLO}| market for copper, anode | APOS, S	Ecoinvent 3 -	CTUh	1.49E-6	1.49E-6	x
3	Steel welded pipe/EU	Industry data	CTUh	1.1E-6	x	1.1E-6
4	Electricity, natural gas, at power plant/US	USLCI	CTUh	7.04E-7	7.04E-7	x
5	Natural gas, at extraction site/US	USLCI	CTUh	5.58E-7	5.58E-7	x

8. *Acidification:*

No	Process	Project	Unit	Total	heat pump	geothermal well
	Total of all processes		mol H+ eq	827	799	27.9
1	Natural gas, processed, at plant/US	USLCI	mol H+ eq	740	740	x
2	Electricity, natural gas, at power plant/US	USLCI	mol H+ eq	35.4	35.4	x
3	Steel welded pipe/EU	Industry data	mol H+ eq	24.3	x	24.3
4	Electricity, bituminous coal, at power plant/US	USLCI	mol H+ eq	8.83	8.83	x
5	Copper, anode {GLO}| market for copper, anode | APOS, S	Ecoinvent 3 -	mol H+ eq	7.56	7.56	x

9. *Ecotoxicity, freshwater:*

No	Process	Project	Unit	Total	heat pump	geothermal well
	Total of all processes		CTUe	1.23E6	1.18E6	4.99E4
1	Natural gas, at extraction site/US	USLCI	CTUe	1.11E6	1.11E6	x
2	Copper, anode {GLO}| market for copper, anode | APOS, S	Ecoinvent 3 -	CTUe	4.46E4	4.46E4	x
3	Concrete block {BR}| market for concrete block | APOS, S	Ecoinvent 3 -	CTUe	3.92E4	x	3.92E4
4	Crude oil, at production/RNA	USLCI	CTUe	2.35E4	2.35E4	x
5	Steel welded pipe/EU	Industry data	CTUe	1.07E4	x	1.07E4

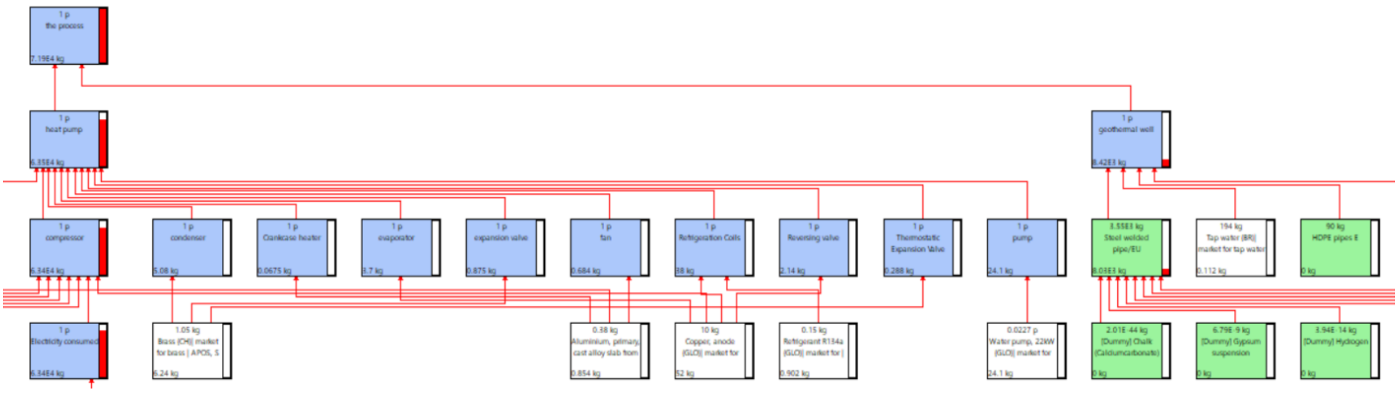
10. Land use:

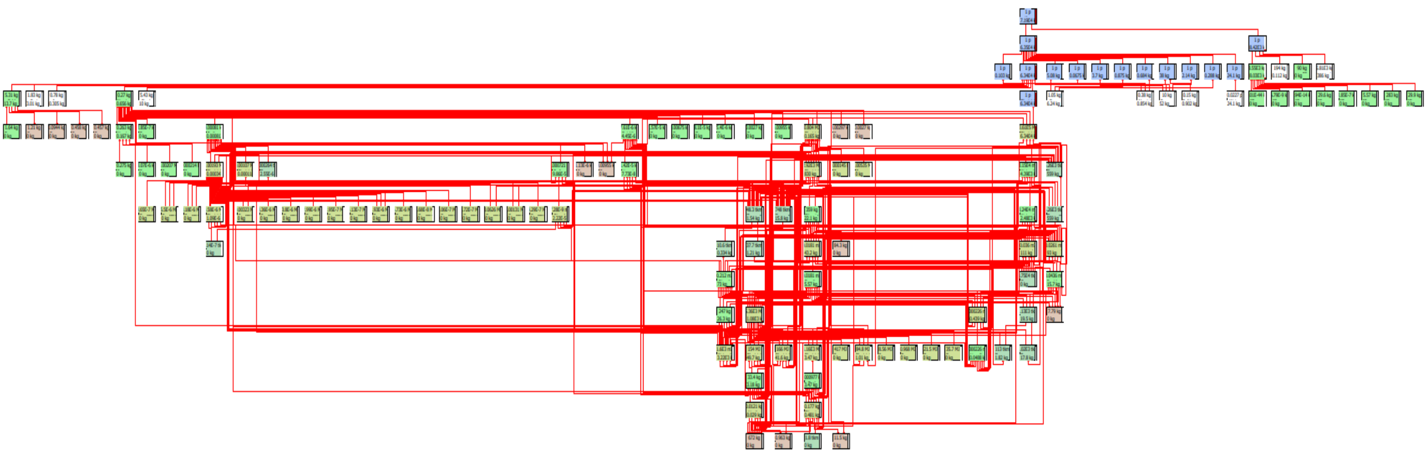
No	Process	Project	Unit	Total	heat pump	geothermal well
	Total of all processes		Pt	1.25E4	2.89E3	9.64E3
1	Concrete block {BR}| market for concrete block | APOS, S	Ecoinvent 3 -	Pt	6.88E3	x	6.88E3
2	Steel welded pipe/EU	Industry data	Pt	2.77E3	x	2.77E3
3	Copper, anode {GLO}| market for copper, anode | APOS, S	Ecoinvent 3 -	Pt	2.42E3	2.42E3	x
4	Water pump, 22kW {GLO}| market for water pump, 22kW | APOS, S	Ecoinvent 3 -	Pt	231	231	x
5	Brass {CH}| market for brass | APOS, S	Ecoinvent 3 -	Pt	173	173	x

11. Water use:

No	Process	Project	Unit	Total	heat pump	geothermal well
	Total of all processes		m3 depriv.	378	167	211
1	HDPE pipes E	Industry data	m3 depriv.	180	x	180
2	Copper, anode {GLO}| market for copper, anode | APOS, S	Ecoinvent 3 -	m3 depriv.	120	120	x
3	Concrete block {BR}| market for concrete block | APOS, S	Ecoinvent 3 -	m3 depriv.	24.4	x	24.4
4	Aluminum, secondary, ingot, at plant/RNA	USLCI	m3 depriv.	20.3	20.3	x
5	Water pump, 22kW {GLO}| market for water pump, 22kW | APOS, S	Ecoinvent 3 -	m3 depriv.	13.1	13.1	x

- ***Detailed Flowchart:***





3.2. El-Alamein Case Study:

- ***Introduction:***

The second phase of the project is designing a geothermal cooling/heating plant for:

1. One of the Gate towers in El-Alamein city at Egypt's north coast.
2. A villa in Beverly Hills compound at Egypt's north coast.

The design will be validated using *GHX_Design_Toolbox* to check the ground response on the underground system. Moreover, an economic feasibility will be done for the project for implementation assessment.

3.2.1 Gate towers in El-Alamein city:

3.2.1.1 Input data: -

Gate Towers Site Specifications	
Location	El-Alamein city, North coast, Egypt **DMS: 30° 51′ 26″ N, 28° 51′ 19″ E**
Area, m^2	594.8 m^2 / Floor
Current use	Hotel

Figure 53: Gate Towers in El-Alamein city

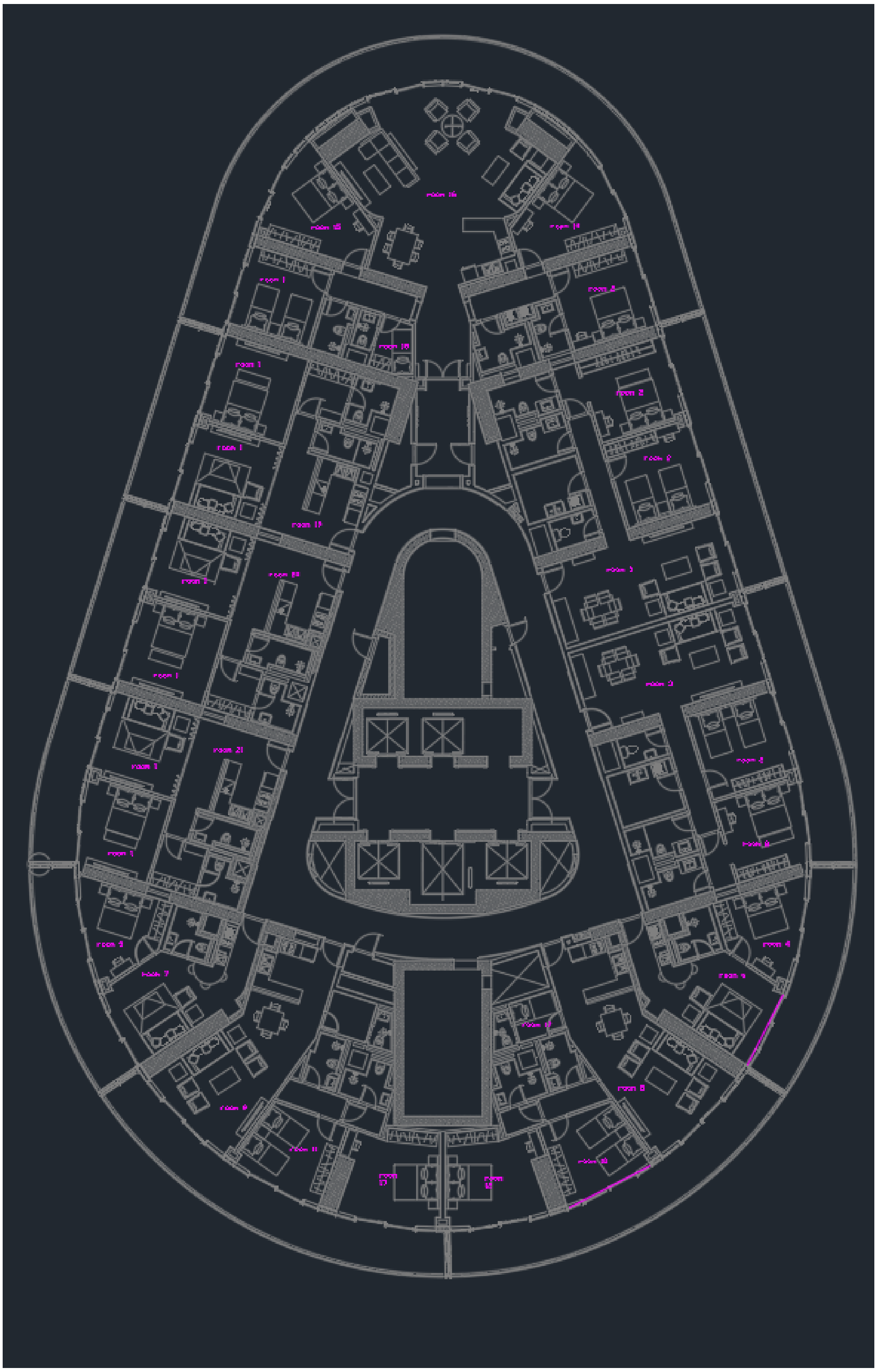


Figure 54: Plan view of one floor

3.2.1.2 Results:

3.2.1.2.1 Cooling and heating loads for floor annually:

The cooling & heating loads calculated by ***HAP*** for the floor at **$24°C$** demand temperature according to ***ASHRE standard design.***

Room	*Surface Area (m^2)*	*Wall Area (m^2)*	*Partition Area (m^2)*
1	13.4	10.5	34
2	13.4	10.5	34
3	30	10.5	61.2
4	12.7	16.2	30
5	12.7	16.2	30
6	21.6	10.5	59.1
7	21.6	10.5	59.1
8	37.1	14.4	71.1
9	37.1	14.4	71.1
10	14	12.3	30.9
11	14	12.3	30.9
12	17.4	13.5	40.5
13	17.4	13.5	40.5
14	16	14.1	34.8
15	16	14.1	34.8
16	62	25.2	86.7
17	3.9	0	23.7
18	3.9	0	23.7
19	17.3	0	57.6
20	17.3	0	57.6
21	17.3	0	57.6

Table 10: Data for one floor - Gate Towers

Time	January	February	March	April	May	June	July	August	September	October	November	December
0:00	23.9	18.7	15.9	14.4	74.4	78.4	79.4	80.1	73.2	64	17.3	23.5
1:00	26.3	19.9	17.3	16	70.8	76.8	75.7	74.8	68.5	60.4	18.3	26.9
2:00	29.5	24.1	18.6	17.4	66	72.9	72.1	74.1	70.1	58.7	20.5	28.9
3:00	31.4	25.5	19.5	18.4	64.9	67.6	70.7	68.5	64.1	53.1	23.6	31.9
4:00	34.1	29.1	20	19	60.4	64.9	66.3	65.8	61	54	25.1	33.2
5:00	35.1	29.4	20	19	60.6	64.8	68.8	64.2	61.6	50.6	27.3	35.4
6:00	36.4	31.7	20.5	18.2	57.1	65.8	63.2	65.6	58	54.2	27.3	35.3
7:00	35.6	30.2	21.4	16.7	57	62.8	68.2	63	58	51.2	27.9	35.9
8:00	34.9	30.6	19.7	14.3	58.3	64.3	67	66.5	59.5	55.8	25.8	33.7
9:00	31.9	26.6	18.3	12.6	65.3	71.7	70.3	69.2	64.2	53.6	24.2	32.1
10:00	29.4	25.4	11.2	0	73.9	80.2	78.1	81.4	71.9	62.7	20.3	27.9
11:00	20.5	11.9	0	0	85.5	90.1	91.3	87.8	86.5	71.8	10	20.5
12:00	10.5	0	0	0	95.4	99.7	99.8	98.2	89.7	84	0	9.3
13:00	0	0	0	0	98.6	102.7	106.5	102.2	95	84.3	0	0
14:00	0	0	0	0	101	104.3	103.6	104.7	97.9	94	0	0
15:00	0	0	0	0	99.7	101.5	105.5	101.6	99.9	94.9	0	0
16:00	0	0	0	0	98.3	104.5	102.5	104.4	100.6	96.5	0	0
17:00	0	0	0	0	100.6	106.6	104.1	106	103.5	95.4	0	0
18:00	0	0	0	0	100.1	107.8	104.8	107.5	100.9	93.8	0	0
19:00	0	0	0	0	102.2	104.6	104.1	107.6	99.8	94.2	0	0
20:00	0	0	0	0	99	100.2	101.9	103.3	96.3	87.5	9.3	9.8
21:00	14.7	12.9	8.2	0	95.1	100.9	98.3	97.9	86.9	77.7	14	14.9
22:00	17.3	16.4	15	14.3	81.3	87.9	90.2	87.1	78.4	71.1	16.3	17.3
23:00	20.1	18.4	17.1	16.4	73.9	81.7	84.9	83.9	76.1	66.9	18.1	20.6

Table 11: Daily Cooling & Heating Loads for each month using HAP - Gate Towers

For Cooling Loads

For Heating Loads

Operating hours

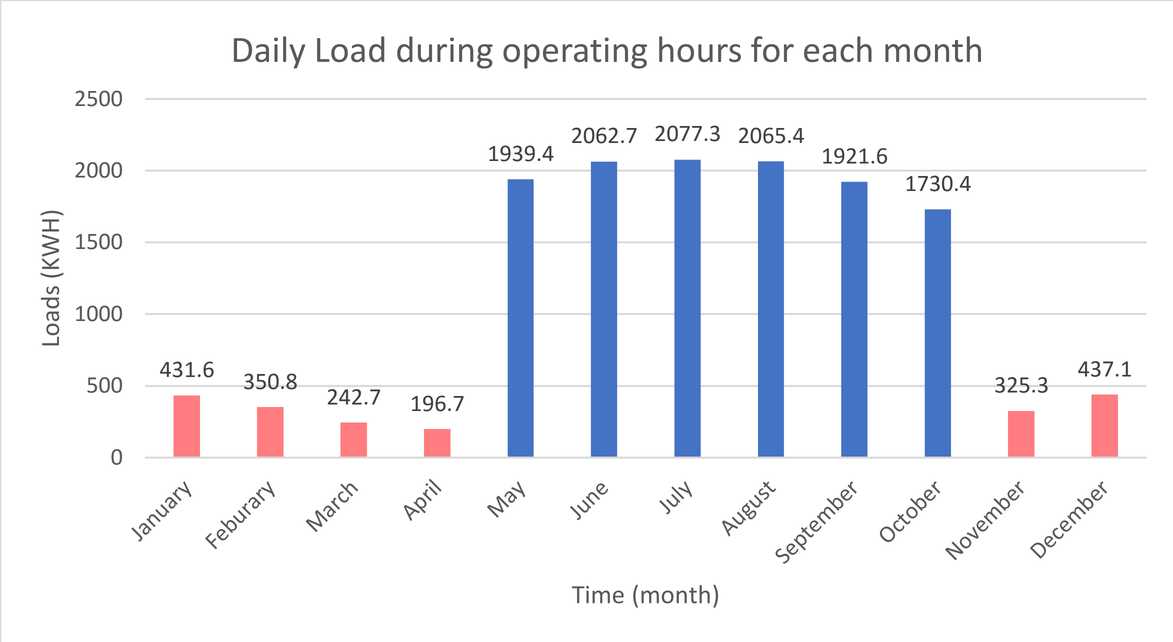


Figure 55: Daily Load during operating hours for each month for one floor - Gate Towers

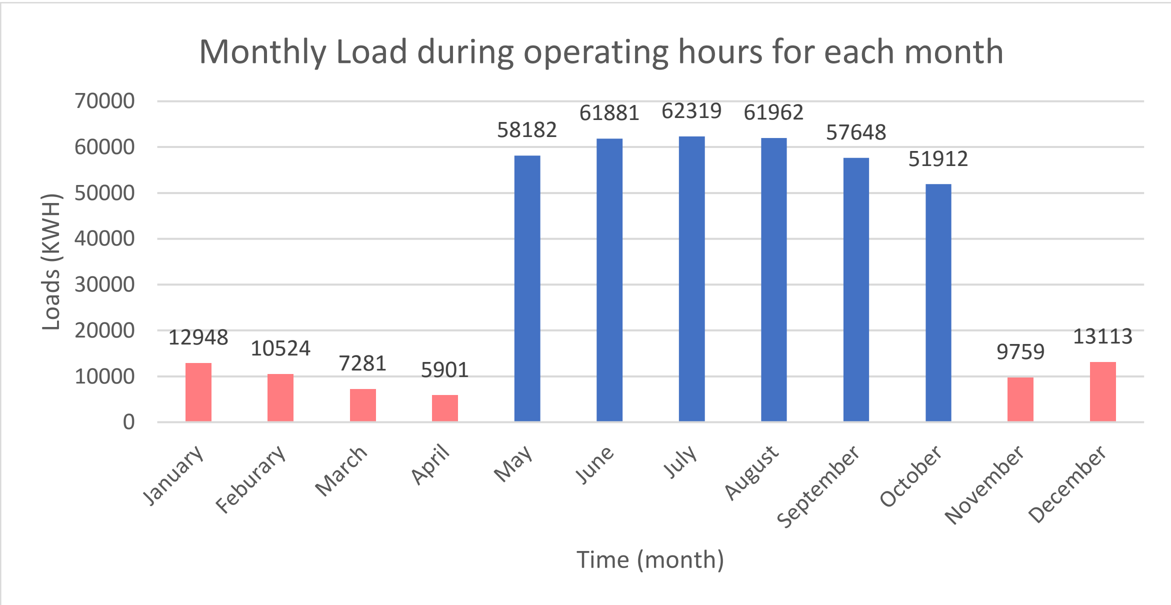


Figure 56: Monthly Load during operating hours for each month for one floor - Gate Towers

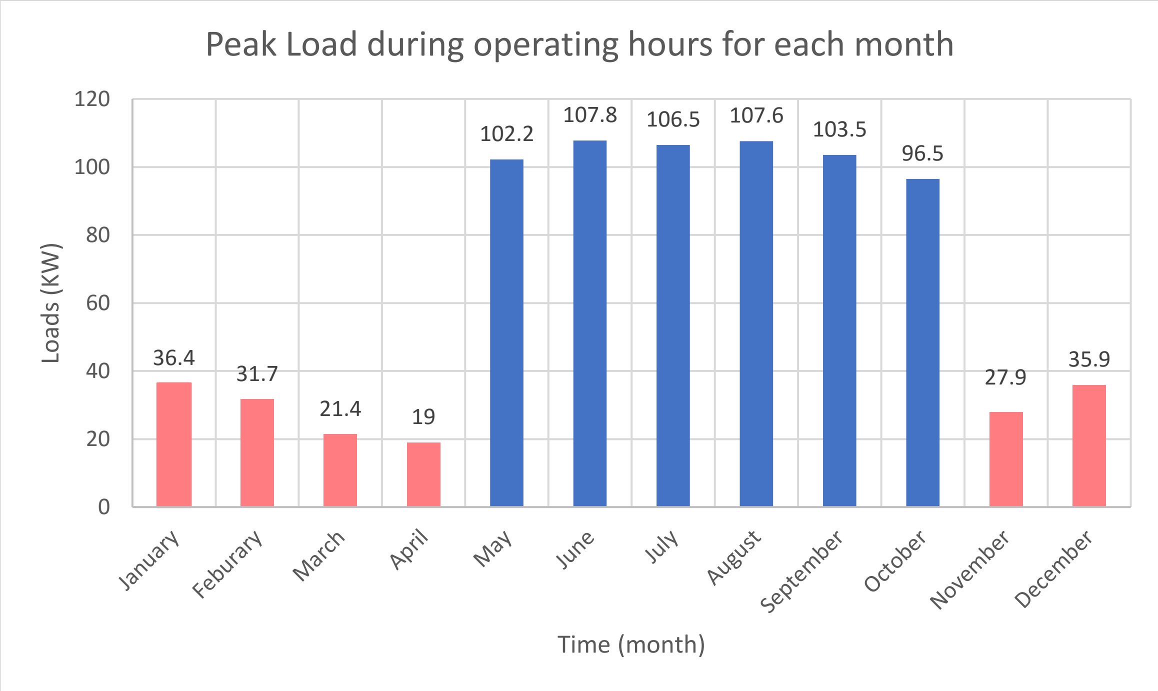


Figure 57: Peak Load for each month for one floor - Gate Towers

3.2.1.2.2 Cooling Cycle Design Conditions:

No.	Parameters	Value
1	*Cooling Load* $\equiv Q_L$	$108\ KW$
2	*Low Pressure* $\equiv P_L$	$3.2\ bar$
3	*High Pressure* $\equiv P_H$	$12\ bar$
4	*Compression ratio* $\equiv r_c$	3.75
5	*Ambient Temperature* $\equiv T_{amb.}$	30 ℃
6	*Condenser Temperature* $\equiv T_{cond.}$	24 ℃
7	*Entering Fluid Temperature* $\equiv T_{w,in}$	18.6 ℃
8	*Volume flow rate* $\equiv \dot{V}$	$216\ L/min.$

- ***<u>Therefore, the refrigerant Type is (R410A)</u>***

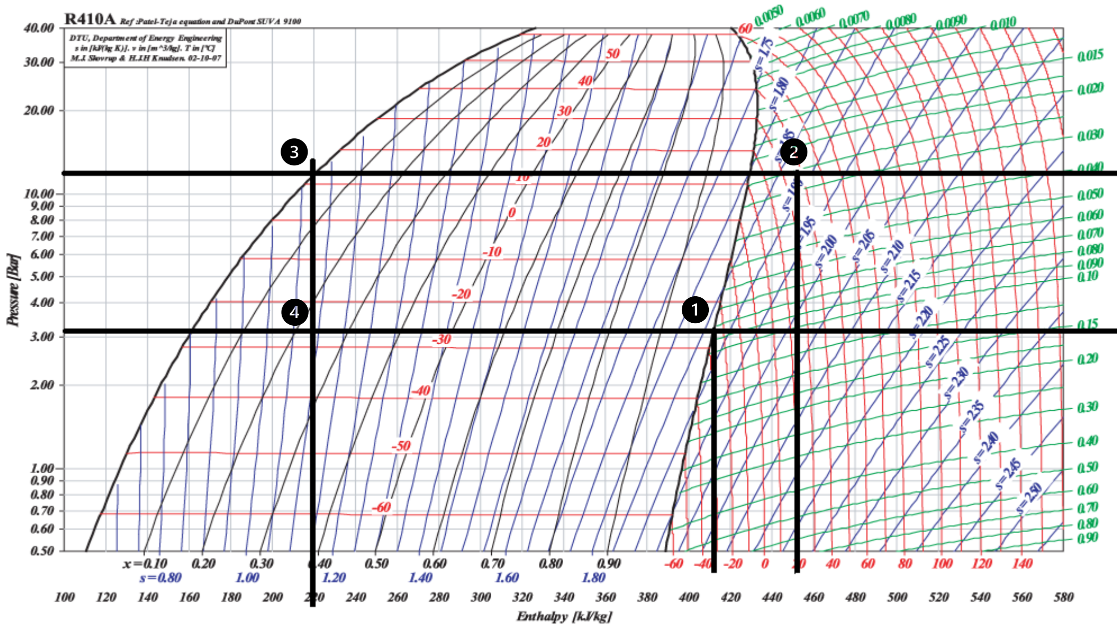


Figure 58: P-h Diagram (R-410A)

- ***<u>From P-h Chart:</u>***

Point	*P (Bar)*	*T (℃)*	*h (kJ/kg)*	*S*
1	3.2	-27	413	1.87
2	12	35	455	1.87
3	12	13	220	-
4	3.2	-27	220	-

- ***<u>Assumptions:</u>***

1- Compression process is isentropic.
2- Expansion process is isobaric.
3- No losses as the Condenser Heat exchange with water ($Q_h = Q_w$)

3.2.1.2.3 Calculations:

$$Q_L = 108\ Kw = \dot{m}_{ref}(h_1 - h_4)\ \rightarrow\ \dot{m}_{ref} = 0.5595\ kg/s$$

From design:

$$\dot{V}_w = 216\ L/min\ \&\ \rho_w = 1000\ kg/m^3$$

$$\boxed{\dot{m}_w = \frac{216 * 1000}{1000 * 60} = 3.6\ kg/s}$$

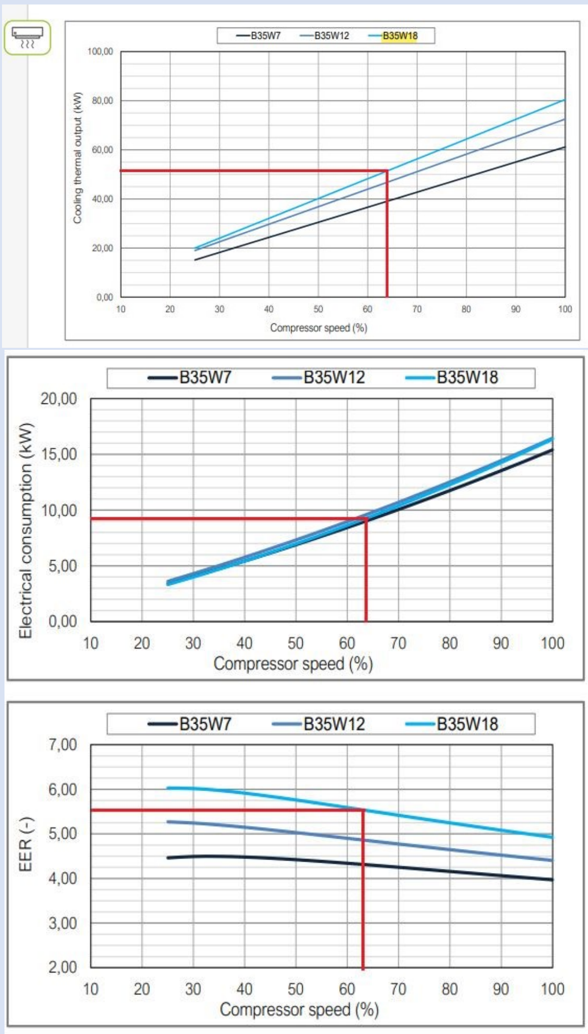
Applying Energy Balance on Condenser:

$$\dot{m}_{ref}(h_2 - h_3) = \dot{m}_w * C_w * (T_{wo} - T_{wi})$$

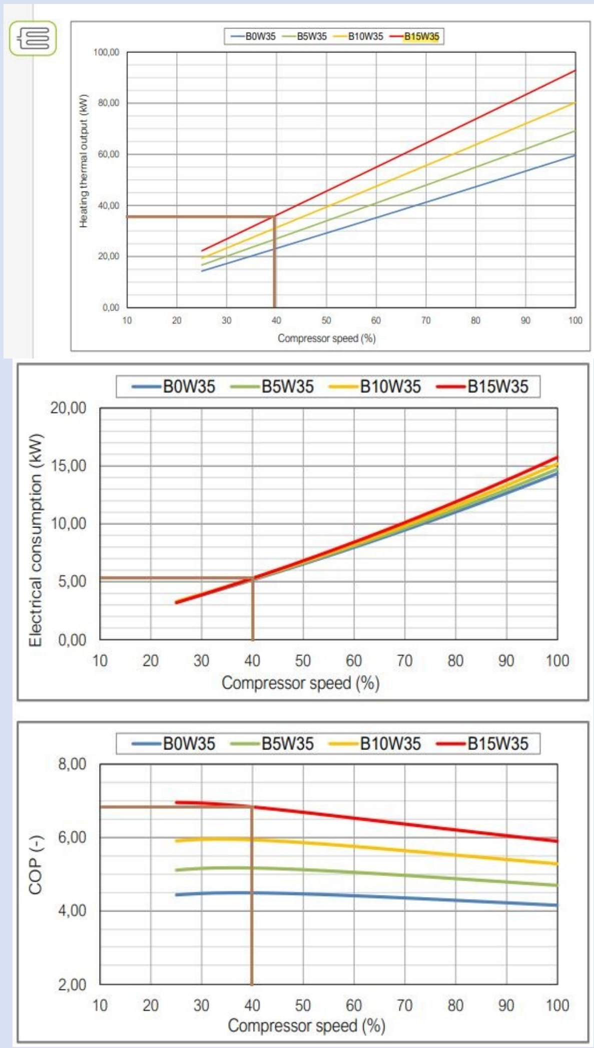
$$0.5595 * (455 - 220) = 3.6 * 4.18 * (T_{wo} - 18.6)$$

$$\boxed{T_{wo} = 27.33\ ℃}$$

Eco-forest Cooling Graphs



Eco-forest Heating Graphs



3.2.1.2.4 Heat Pump Selection:

ecoGEO HP 15-70

- **Modulating thermal power control** within a wide range (25-100%) and modulating flow rate control of both brine and production circuits (20-100%).
- **Inverter technology** and scroll compressor.
- **Integrated management of up to 5 different emission temperatures, 2 buffer tanks** (heating and cooling), **1 DHW tank, 1 pool and hourly control of DHW recirculation.**
- **Management of aerothermal collection modulating units**, in case of air source or hybrid configurations by means of the **ecoSMART e-source.**
- **Integrated management of external On/Off or modulating auxiliary systems**, such as electrical heaters, On/Off boilers or modulating boilers.
- **Management of cascade systems up to 6 units by means of the ecoSMART Supervisor.**
- **Integrated management of simultaneous cooling/heating systems** according to scheme.
- **Free cooling (Passive cooling) management.**
- **Integrated active cooling in models 3.**
- **Three-phase version available.**
- **Compatible with ecoSMART e-manager and ecoSMART e-system.**
- **Integrated energy meters** to measure the electrical consumption, the heating/cooling thermal power, the COP and the monthly and annual SPF.

SPECIFICATIONS ecoGEO HP 15-70		UNITS	HP1	HP3
APPLICATION	Place of installation	-	Indoors	
	Type of brine system [1]	-	Ground source / Air source / Hybrid source	
	DHW with external tank	-	✓	✓
	Heating and Pool	-	✓	✓
	External Passive cooling management	-	✓	✓
	Integrated Active cooling	-	-	✓
PERFORMANCE	Modulation range of the compressor	%	25 to 100	
	Heating power output [1], B0W35	kW	17,1 to 59,6	
	COP [1], B0W35	-	4,5	
	Active cooling power output [1], B35W7	kW	-	15,1 to 61,5
	EER [1], B35W7	-	-	4,5
	Max. DHW temperature without / with support	°C	60 / 70	
	Noise power emission level [3]	db	53 to 71	
	Energy label / ηs / SCOP W35 average climate control	-	A+++ / 180% / 4,71	
	Energy label / ηs / SCOP W55 average climate control	-	A++ / 139% / 3,67	
OPERATION LIMITS	Distribution / Set heating outlet temperature range [2]	°C	10 to 60 / 20 to 60	
	Distribution / Set cooling outlet temperature range [2]	°C	5 to 35 / 7 to 25	
	Brine inlet temperature range in heating applications [2]	°C	-20 to 35	
	Brine inlet temperature range in cooling applications [2]	°C	10 to 60	
	Minimum / Maximum refrigerant circuit pressure	bar	2 / 45	
	Production / Pre-load circuit pressure	bar	0,5 to 5,0	
	Brine / Pre-load circuit pressure	bar	0,5 to 5,0	
WORKING FLUIDS	R410A Refrigerant load	kg	4,7	5,5
	Compressor oil type / load	kg	POE / 3,6	
	Nominal primary flow rate, B0W35 [1] (ΔT = 3 °C)	l/h	3230 to 13195	
	Nominal secondary flow rate, B0W35 [1] (ΔT = 5 °C)	l/h	2465 to 10265	
CONTROL ELECTRICAL DATA	1/N/PE 230 V / 50-60 Hz [5]	-	✓	
	Maximum recommended external protection [7]	-	C1A	
	Transformer primary circuit fuse	A	0,63	
	Transformer secondary circuit fuse	A	4,0	
ELECTRICAL DATA: THREE-PHASE	3/N/PE 400 V / 50-60Hz [5]	-	✓	
	Maximum recommended external protection [7]	-	C50A	
	Maximum consumption [2], B0W35	kW / A	14,3 / 23,2	
	Maximum consumption [2], B0W55	kW / A	20,4 / 32,3	
	Minimum / Maximum starting current [4]	A	7,5 / 11,8	
	Correction of cosine Ø	-	0,96-1	
DIMENSIONS/WEIGHT	Height x width x depth	mm	1063x870x785	
	Empty weight (without assembly)	kg	320	325

1. In compliance with EN 14511, this includes the consumption of the circulation pumps and the compressor driver.
2. With variable speed circulating pumps, managed by the ecoGEO HP heat pump.
3. According to EN 12102.
4. Starting current depends on working condition of the hydraulic circuits.
5. The admissible voltage range for proper operation of the heat pump is ±10%.
6. Maximum consumption can vary significantly according to working conditions, or if the compressor's range of operation is restricted.
7. External protection exclusively regarding the ecoGEO heat pump controller electrical consumption. This protection should be updated in case of using the controller single-phase electrical supply to wire other equipments depending on the features of such equipments.
8. In case of air source or hybrid source configuration, it is required to combine the ecoGEO HP heat pump with the ecoSMART e-source.

Note: primary circuit and secondary circuit circulation pumps not included.

Figure 59: Eco-Forest GSHP (HP 15-70) Data specifications - El-Alamein Gate Towers

3.2.1.2.5 Summary

According to the previous figure of Eco- Forest GSHP datasheets, the selection can be shown in the following table:

El-Alamein Gate Towers Results	
No. of heat pumps selected	2
Maximum capacity of one heat pump	$70\ kW$

I. Vertical

From *GHX_Design_Toolbox* software, results have shown that the required total bore length for cooling is 1584 m. Details are found in Appendix A.

RESULTS:

Number of Vertical Boreholes: 16

Borehole Field Configuration:
Uniform Grid of 4 x 4 boreholes, 7-m bore-to-bore spacing.

Calculated Depth for Each Borehole: 99 m

Total Length Requirement for Boreholes:
1584 m, driven by the cooling loads (240 m are driven by the heating loads)

Relative Impact on GHX Design Length Due to the Ground Thermal Resistances
Related to:
Borehole Resistance: 63.6%
Daily Pulse: 12.7%
Monthly Pulse: 16.6%
Annual Pulse: 7.1%

Residual: 5.384E-06

Calculation Time: 2.578E-01 seconds

Figure 60: GHX_Design_Toolbox Results – Vertical GSHP – El-Alamein Gate Towers

II. Horizontal

From *GHX_Design_Toolbox* software, results have shown that the required total trench length for cooling is 1579 m. Details are found in Appendix A.

Microsoft Excel

RESULTS:

Trench Configuration: 20 @ 3.0 m spacing
Calculated Length for Each Trench: 78.9 m

Total Length Requirement for Trenches:
1579 m, driven by the cooling loads (-80 m are driven by the heating loads)

Relative Impact on GHX Design Length Due to the Ground Thermal Resistances
Related to:
Trench Resistance: 67.3%
Daily Pulse: .5%
Monthly Pulse: 3.4%
Annual Pulse: 28.9%

Residual: 8.429E-06

Calculation Time: 6.797E-01 seconds

Figure 61: GHX_Design_Toolbox Results – Horizontal GSHP – El-Alamein Gate Towers

3.2.2 Villa unit in El-Alamein city:

3.2.2.1 Input data:

The Villa is a ***two-floor building*** that is allocated overlooks the north coast and by evaluating the building's orientation and defining the building characteristics such as wall insulation, windows installation, lighting system & etc…, a proper geothermal cooling system can be built.

Figure 62: Ground Floor Plan View

Figure 63: First Floor Plan View

3.2.2.2 Results:

3.2.2.2.1 Cooling and heating loads for floor annually:

The cooling & heating loads calculated by ***HAP*** for the floor at ***24ºC*** demand temperature according to ***ASHRE standard design.***

time	January		February		March		April		November		December	
H	QL (KW)	E (KW)	QL (KW)	E (KW)	QL (KW)	E (KW)	QL (KW)	E (KW)	QL (KW)	E (KW)	QL (KW)	E (KW)
0:00	9.8	2.1304	8.9	1.9837	8	1.837	7.7	1.7881	7.8	1.8044	9.7	2.1141
1:00	11.8	2.4564	9.4	2.0652	8.5	1.9185	8.1	1.8533	8.3	1.8859	11.4	2.3912
2:00	12.7	2.6031	9.9	2.1467	8.9	1.9837	8.5	1.9185	8.8	1.9674	12.6	2.5868
3:00	14.6	2.9128	12.4	2.5542	9.2	2.0326	8.8	1.9674	9.1	2.0163	14.1	2.8313
4:00	15.1	2.9943	12.4	2.5542	9.4	2.0652	8.9	1.9837	11.1	2.3423	14.9	2.9617
5:00	16.6	3.2388	14.6	2.9128	9.3	2.0489	8.8	1.9674	11.1	2.3423	16	3.141
6:00	16.4	3.2062	13.8	2.7824	9.9	2.1467	8.4	1.9022	12.6	2.5868	16.2	3.1736
7:00	17.1	3.3203	15.3	3.0269	10.1	2.1793	7.7	1.7881	11.7	2.4401	16.5	3.2225
8:00	16	3.141	13.4	2.7172	9.8	2.1304	7.8	1.8044	12.3	2.5379	15.7	3.0921
9:00	15.7	3.0921	12.4	2.5542	8.9	1.9837	6.4	1.5762	10.2	2.1956	15	2.978
10:00	13.6	2.7498	12.8	2.6194	5.9	1.4947	0	0.533	10	2.163	13.3	2.7009
11:00	10.2	2.1956	5.3	1.3969	0	0.533	0	0.533	3	1.022	9.4	2.0652
12:00	3.1	1.0383	0	0.533	0	0.533	0	0.533	0	0.533	0	0.533
13:00	0	0.533	0	0.533	0	0.533	0	0.533	0	0.533	0	0.533
14:00	0	0.533	0	0.533	0	0.533	0	0.533	0	0.533	0	0.533
15:00	0	0.533	0	0.533	0	0.533	0	0.533	0	0.533	0	0.533
16:00	0	0.533	0	0.533	0	0.533	0	0.533	0	0.533	0	0.533
17:00	0	0.533	0	0.533	0	0.533	0	0.533	0	0.533	0	0.533
18:00	0	0.533	0	0.533	0	0.533	0	0.533	0	0.533	0	0.533
19:00	0	0.533	0	0.533	0	0.533	0	0.533	0	0.533	0	0.533
20:00	4.4	1.2502	3.9	1.1687	3.7	1.1361	3.5	1.1035	5.3	1.3969	5.3	1.3969
21:00	7.4	1.7392	7.3	1.7229	6.6	1.6088	5.7	1.4621	7.2	1.7066	7.3	1.7229
22:00	8.5	1.9185	8.8	1.9674	8.6	1.9348	8.4	1.9022	8	1.837	8.2	1.8696
23:00	9.2	2.0326	9.4	2.0652	9.3	2.0489	9.3	2.0489	8.7	1.9511	9	2
Tolal E /day (KWH)		45.7506		40.502		33.3463		30.396		36.4596		44.5118
Total E (working hour)/month (KWH)		1372.518		1215.06		1000.389		911.88		1093.788		1335.354

Figure 65: Heating Loads using HAP - Villa

time	May		June		July		August		September		October	
H	QL (KW)	E (KW)	QL (KW)	E (KW)	QL (KW)	E (KW)	QL (KW)	E (KW)	QL (KW)	E (KW)	QL (KW)	E (KW)
0:00	29.1	4.2	32.3	4.7	33.4	4.8	32.4	4.7	31.6	4.6	29.4	4.3
1:00	28.4	4.1	30.2	4.4	32.0	4.6	30.6	4.4	30.2	4.4	26.4	3.9
2:00	27.1	4.0	29.2	4.2	29.3	4.3	29.7	4.3	27.9	4.1	24.7	3.6
3:00	25.2	3.7	28.0	4.1	28.2	4.1	27.4	4.0	26.5	3.9	23.0	3.4
4:00	24.7	3.6	27.0	3.9	27.8	4.1	27.7	4.0	25.8	3.8	22.6	3.3
5:00	23.1	3.4	25.4	3.7	26.1	3.8	26.0	3.8	23.9	3.5	22.2	3.3
6:00	23.4	3.5	24.9	3.7	25.6	3.8	25.6	3.8	24.9	3.7	22.5	3.3
7:00	22.6	3.3	24.9	3.7	25.6	3.8	25.5	3.7	24.8	3.6	22.1	3.3
8:00	23.9	3.5	25.4	3.7	26.1	3.8	26.1	3.8	25.3	3.7	22.4	3.3
9:00	26.3	3.9	28.5	4.2	28.6	4.2	28.0	4.1	26.5	3.9	23.6	3.5
10:00	32.4	4.7	36.3	5.2	35.9	5.2	34.5	5.0	31.7	4.6	25.8	3.8
11:00	39.8	5.7	42.3	6.0	43.4	6.2	40.2	5.7	38.2	5.5	33.1	4.8
12:00	44.2	6.3	47.5	6.7	46.4	6.6	45.7	6.5	44.1	6.3	39.2	5.6
13:00	45.7	6.5	49.0	6.9	49.9	7.1	48.5	6.9	46.7	6.6	44.1	6.3
14:00	47.9	6.8	51.1	7.2	49.6	7.0	51.1	7.2	47.8	6.8	44.2	6.3
15:00	46.9	6.7	50.1	7.1	51.4	7.3	52.1	7.4	48.5	6.9	44.9	6.4
16:00	46.8	6.6	50.5	7.1	51.3	7.3	50.7	7.2	49.4	7.0	45.3	6.4
17:00	46.5	6.6	50.0	7.1	50.4	7.1	49.6	7.0	50.6	7.2	45.3	6.4
18:00	45.2	6.4	49.2	7.0	49.6	7.0	49.7	7.0	47.9	6.8	44.0	6.3
19:00	43.4	6.2	47.7	6.8	47.1	6.7	45.9	6.5	46.0	6.5	41.4	5.9
20:00	41.2	5.9	45.8	6.5	45.7	6.5	43.5	6.2	43.4	6.2	38.7	5.5
21:00	39.6	5.7	42.3	6.0	42.4	6.0	41.0	5.9	37.7	5.4	35.0	5.0
22:00	33.1	4.8	37.0	5.3	37.2	5.3	37.0	5.3	35.2	5.1	31.0	4.5
23:00	30.4	4.4	33.0	4.8	35.6	5.1	34.9	5.0	33.3	4.8	29.8	4.3
Tolal E /day (KWH)		120.4		130.1		131.6		129.5		124.7		112.8
Total E (working hour)/month (KWH)		3613.272		3901.728		3946.608		3884.592		3739.752		3383.976

Figure 64: Cooling Loads using HAP - Villa

3.2.2.2.2 Heat Pump Selection:

ecoGEO HP 15-70

- **Modulating thermal power control** within a wide range (25-100%) and **modulating flow rate control** of both brine and production circuits (20-100%).
- **Inverter technology** and scroll compressor.
- **Integrated management of up to 5 different emission temperatures, 2 buffer tanks** (heating and cooling), **1 DHW tank, 1 pool and hourly control of DHW recirculation.**
- **Management of aerothermal collection modulating units,** in case of air source or hybrid configurations by means of the **ecoSMART e-source.**
- **Integrated management of external On/Off or modulating auxiliary systems,** such as electrical heaters, On/Off boilers or modulating boilers.
- **Management of cascade systems** up to 6 units by means of the **ecoSMART Supervisor.**
- **Integrated management of simultaneous cooling/heating systems** according to scheme.
- **Free cooling (Passive cooling) management.**
- **Integrated active cooling in models 3.**
- **Three-phase version available.**
- **Compatible with ecoSMART e-manager and ecoSMART e-system.**
- **Integrated energy meters** to measure the electrical consumption, the heating/ cooling thermal power, the COP and the monthly and annual SPF.

SPECIFICATIONS ecoGEO HP 15-70		UNITS	HP1	HP3
APPLICATION	Place of installation	-	Indoors	
	Type of brine system [1]	-	Ground source / Air source / Hybrid source	
	DHW with external tank	-	✓	✓
	Heating and Pool	-	✓	✓
	External Passive cooling management	-	✓	✓
	Integrated Active cooling	-	-	✓
PERFORMANCE	Modulation range of the compressor	%	25 to 100	
	Heating power output [1], B0W35	kW	17,1 to 59,6	
	COP [1], B0W35	-	4,5	
	Active cooling power output [1], B35W7	kW	-	15,1 to 61,5
	EER [1], B35W7	-	-	4,5
	Max. DHW temperature without / with support	°C	60 / 70	
	Noise power emission level [3]	db	53 to 71	
	Energy label / ηs / SCOP W35 average climate control	-	A+++ / 180% / 4,71	
	Energy label / ηs / SCOP W55 average climate control	-	A++ / 139% / 3,67	
OPERATION LIMITS	Distribution / Set heating outlet temperature range [2]	°C	10 to 60 / 20 to 60	
	Distribution / Set cooling outlet temperature range [2]	°C	5 to 35 / 7 to 25	
	Brine inlet temperature range in heating applications [2]	°C	-20 to 35	
	Brine inlet temperature range in cooling applications [2]	°C	10 to 60	
	Minimum / Maximum refrigerant circuit pressure	bar	2 / 45	
	Production / Pre-load circuit pressure	bar	0,5 to 5,0	
	Brine / Pre-load circuit pressure	bar	0,5 to 5,0	
WORKING FLUIDS	R410A Refrigerant load	kg	4,7	5,5
	Compressor oil type / load	kg	POE / 3,6	
	Nominal primary flow rate, B0W35 [1] (ΔT = 3 °C)	l/h	3230 to 13195	
	Nominal secondary flow rate, B0W35 [1] (ΔT = 5 °C)	l/h	2465 to 10265	
CONTROL ELECTRICAL DATA	1/N/PE 230 V / 50-60 Hz [5]	-	✓	
	Maximum recommended external protection [7]	-	C1A	
	Transformer primary circuit fuse	A	0,63	
	Transformer secondary circuit fuse	A	4,0	
ELECTRICAL DATA: THREE-PHASE	3/N/PE 400 V / 50-60Hz [5]	-	✓	
	Maximum recommended external protection [7]	-	C50A	
	Maximum consumption [2], B0W35	kW / A	14,3 / 23,2	
	Maximum consumption [2], B0W55	kW / A	20,4 / 32,3	
	Minimum / Maximum starting current [4]	A	7,5 / 11,8	
	Correction of cosine Ø	-	0,96-1	
DIMENSIONS/WEIGHT	Height x width x depth	mm	1063x870x785	
	Empty weight (without assembly)	kg	320	325

1. In compliance with EN 14511, this includes the consumption of the circulation pumps and the compressor driver.
2. With variable speed circulating pumps, managed by the ecoGEO HP heat pump.
3. According to EN 12102.
4. Starting current depends on working condition of the hydraulic circuits.
5. The admissible voltage range for proper operation of the heat pump is ±10%.
6. Maximum consumption can vary significantly according to working conditions, or if the compressor's range of operation is restricted.
7. External protection exclusively regarding the ecoGEO heat pump controller electrical consumption. This protection should be updated in case of using the controller single-phase electrical supply to wire other equipments depending on the features of such equipments.
8. In case of air source or hybrid source configuration, it is required to combine the ecoGEO HP heat pump with the ecoSMART e-source.

Note: primary circuit and secondary circuit circulation pumps not included.

Figure 66: Eco-Forest GSHP (HP 15-70) Data specifications - El-Alamein Villa

3.2.2.2.3 Summary

From this software, results have shown that the required total bore length for cooling is $450\ m$. Details are found in Appendix A.

Microsoft Excel

RESULTS:

Number of Vertical Boreholes: 6

Borehole Field Configuration:
Uniform Grid of 2 x 3 boreholes, 7-m bore-to-bore spacing.

Calculated Depth for Each Borehole: 75 m

Total Length Requirement for Boreholes:
450 m, driven by the cooling loads (114 m are driven by the heating loads)

Relative Impact on GHX Design Length Due to the Ground Thermal Resistances
Related to:

Borehole Resistance: 48.2%
Daily Pulse: 33.6%
Monthly Pulse: -26.1%
Annual Pulse: 44.3%

Residual: 4.698E-06

Calculation Time: 2.578E-01 seconds

Figure 67: GHX_Design_Toolbox Results – Vertical GSHP – El-Alamein Villa

- $No. of\ boreholes\ =\ 6\ boreholes\ (2 \times 3)$
- Each borehole $75\ m$ depth
- $Spacing\ =\ 7\ m$
- $Land\ area\ required\ =\ 7 \times 14\ =\ 98\ m^2$

The land area required for geothermal heating/cooling system represents about ***46% of total land area***.

3.2.2.2.4 Cost Assessment

- Geothermal heat Pump cost evaluation:

  i. **Drilling Cost:**

Drilling cost		
Cost / m (LE/m)	Total drilling length (m)	Total drilling cost (LE)
1300	450	585,000

  ii. **Unit Cost:**

Unit cost			
Cost (Euro)	No. Of Units	Cost (LE)	Total cost (LE)
22752.93	1	773599.62	773599.62

Then, the initial cost is:

$$Initial\ Cost = 585{,}000 + 773599.62 = \mathbf{1{,}358{,}599.62}\ \mathrm{LE}$$

  iii. **Operation & Maintenance Cost:**

Operation & Maintenance Cost			
Electricity tariff (LE)	Elec. Cons. (KWH/Yr.)	Total Op Cost (LE)	Maintenance Cost per year (LE)
1.45	29,398.917	42,628.43	20,000

Then, the Operation & Maintenance cost is:

$$\mathrm{Operation\ \&\ Maintenance}\ Cost\ per\ Year = 42628.43 + 20000 = \mathbf{62{,}628.43}\ \mathrm{LE}$$

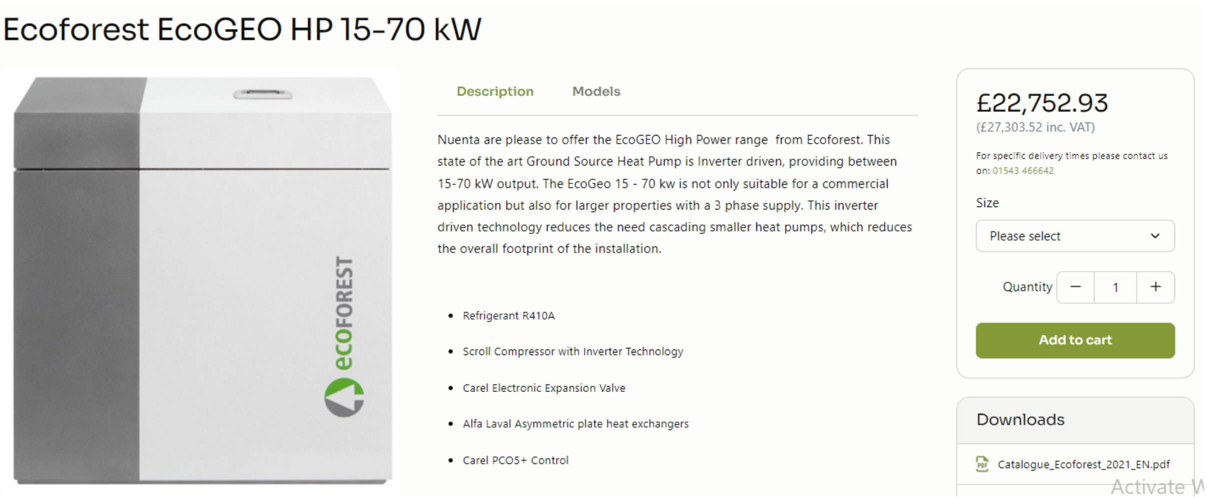


Figure 68: GSHP Specifications (for Villa in El-Alamein)

- Geothermal heat Pump + PV system cost evaluation:

i. PV Cost:

PV cost			
Cost (USD)	No. Of Units	Cost (LE)	Total cost (LE)
130	180	3695.64	665,215.2

ii. Unit Cost:

Inverter cost			
Cost (USD)	No. Of Units	Cost (LE)	Total cost (LE)
1300	5	36956.4	184,782

Then, the initial cost is:

$$Initial\ Cost = 1{,}358{,}599.62 + 665{,}215.2 + 184{,}782 = \mathbf{2,208,596}\ \mathrm{LE}$$

iii. Operation & Maintenance Cost:

Operation & Maintenance Cost annually		
Operation cost (LE)	Maintenance cost (LE)	Total cost per year (LE)
19189	6396	25585

Then, the Operation & Maintenance cost annually is:

$$\mathrm{Operation\ \&\ Maintenance}\ Cost\ per\ Year = 25585 + 20000 = \mathbf{45,585}\ \mathrm{LE}$$

- Air-Conditioning cost evaluation:

Floor	No.	Space	Floor area (m^2)	TR	Hp (electric)	Split needed	Cost (EGP)	KWE installed
Ground	1	Reception	93.309	5	9.5	2*4hp + 1*1.5hp	105,670	6.98
	2	Dining room	27.946	1.5	3	1*3hp	30,830	2.2
	3	Kitchen	12.139	0.65	1.5	1*1.5hp	20,450	1.1
	4	Main entrance	6.02	0.322	1.5	1*1.5hp	20,450	1.1
	5	Nany room	5.841	0.31	1.5	1*1.5hp	20,450	1.1
	6	Bedroom 1	16.272	0.87	1.5	1*1.5hp	20,450	1.1
First	7	Bedroom 2	32.2792	1.73	3	1*3hp	30,830	2.2
	8	Master bedroom & dressing	46.2397	2.5	3	2*1.5hp	40,900	2.2
	9	living and kitchen	40	2.15	4	1*4hp	42,610	2.94
	10	bedroom1	18.1	0.97	1.5	1*1.5hp	20,450	1.1
				Σ	30	SUM	353,090	22.02

Selecting SHARP AC units:

SHARP	*Cost*
1.5 hp	20,450 EGP
3 hp	30,830 EGP
4 hp	42,610 EGP

i. **Initial Cost:**

Initial cost	
No. of AC Unit	Total AC units' cost (LE)
10	353,090

Then, the initial cost is:

$$\textit{Initial Cost} = \mathbf{353,090}\ \text{LE}$$

ii. **Operation & Maintenance Cost:**

Operation & Maintenance Cost			
Electricity tariff (LE)	Elec. Cons. (KWH/Yr)	Total Op Cost (LE)	Maintenance Cost per year (LE)
1.45	163,960.92	237,743.334	15,000

Then, the Operation & Maintenance cost is:

$$\text{Operation \& Maintenance } \textit{Cost per Year} = 237743.334 + 15000 = \mathbf{252,743.334}\ \text{LE}$$

➢ Payback period:

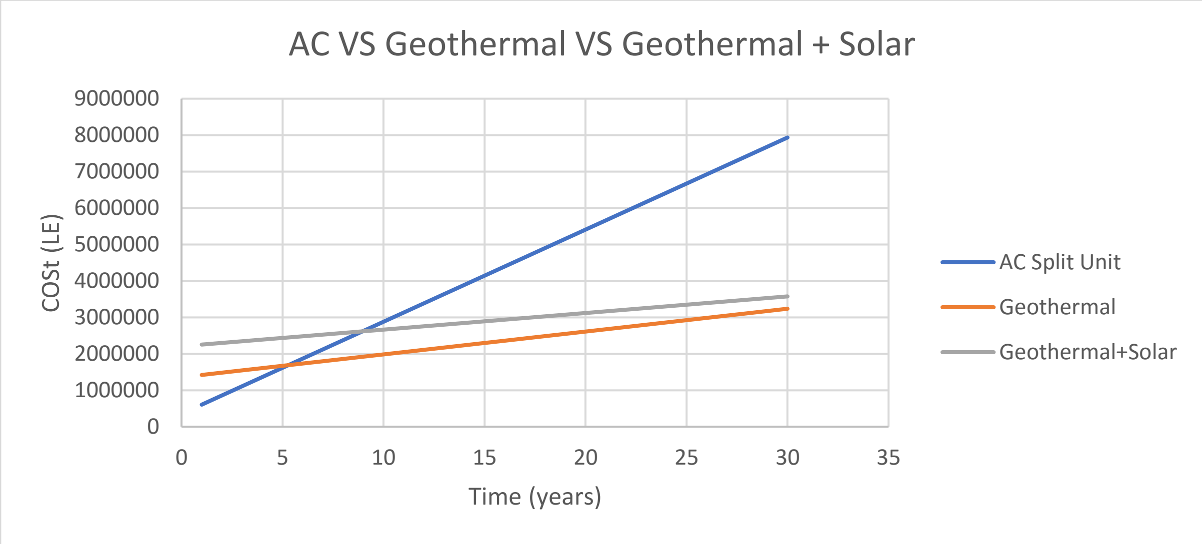


Figure 69: Payback period (Ac vs GSHP vs GSHP+PV) – El-Alamein Villa

- ***The payback period is 5.6 years*** if installing Geothermal heat pumps instead of conventional air conditioning.

Years	AC	GEO
1	605833	1421228
2	858577	1483856
3	1111320	1546485
4	1364063	1609113
5	1616807	1671742
6	1869550	1734370

- ***The payback period is 9.17 years*** for installing Geothermal heat pumps and PV system instead of conventional air conditioning.

Years	AC	Geo + PV
1	605833	2254182
2	858577	2299767
3	1111320	2345352
4	1364063	2390937
5	1616807	2436522
6	1869550	2482107
7	2122293	2527692
8	2375037	2573277
9	2627780	2618862
10	2880523	2664447

- From the graph above, it is clearly that the most economic case is installing Geothermal heat pumps as it has shortest payback period & has the minimum cost at the lifetime of the system (30 years).

3.2.2.2.5 CO2 Emission Savings: (GSHP vs AC)

AC	***GEO***	***CO2 emission factor (gCO2/kWh)***
Elec. Cons. (KWH/Yr)	Elec. Cons. (KWH/Yr)	400
163960.92	29398.917	
CO2 emission (gCO2/yr)	CO2 emission (gCO2/yr)	
65,584,368	11,759,566.8	

Then, CO_2 Emission Savings per year (g CO_2/yr):

$$\boxed{\mathbf{CO_2\ Emission\ Savings} = 65{,}584{,}368 - 11{,}759{,}566.8 = \mathbf{53,824,801.2\ gCO2eq.}}$$

$$\%\ Savings = \frac{\mathbf{53,824,801.2}}{\mathbf{65,584,368}} \times \mathbf{100} = \mathbf{82\%}$$

3.3. Acoustics building- Ain Shams University Case Study:

Our goal is to calculate the cost of replacing one chiller (38.5 TR) with a Geothermal heat pump to cover the acoustics building's cooling/heating loads to reduce electric consumption & carbon footprint.

The Acoustics building is located at Faculty of engineering – Ain Shams University.

3.3.1 Input Data

Ground Properties	
Thermal conductivity	$2.1\ W/m.k$
Volumetric heat capacity	$2.5\ MJ/m^3.k$
Ground surface temperature	$24°C$
Geothermal heat flux	$0.08\ W/m^2$
Borehole Specifications	
Type	*Single U-tube*
Depth of one borehole	$100\ m$
Spacing between boreholes	$7\ m$
Borehole Diameter	$152\ mm$
Thermal resistance for pipe/grout	$0.074\ m.k/W$
Grout thermal conductivity	$0.6\ W/m.k$
Flow rate for one borehole	$16.2\ L/min$
Pipe Specifications	
Outer diameter	$32\ mm$
Thickness	$3\ mm$
Thermal conductivity	$0.42\ W/m.k$
Shank spacing	$120\ mm$

Table 12: Acoustics Building Input Data

Note:

- Borehole depth is variable with available excavation area.
- Minimum spacing between boreholes $[7-8]\ m$.
- Recommended flow rate per borehole: $Q = 13.333 - 16.667\ L/min$

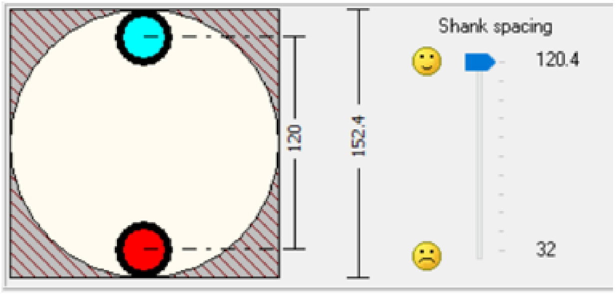


Figure 70: Borehole Shank spacing.

Heat carrier Fluid (Water)	
Thermal conductivity	$0.608\ W/m.k$
Specific heat capacity	$4180\ J/kg.k$
Density	$997.2\ kg/m^3$
Viscosity	$0.000891\ Kg/m.s$
Freezing point	$0°C$

- ***Software used:***

Earth Energy Designer 4.2	
Simulation period	$25\ Years$
First month of operation	*September*

EED
Earth Energy Designer

- ***Base Load:***

Month	Heating (MWh)	Cooling (MWh)
January	0.248	0
February	0.182	0
March	0.098	0
April	0.078	0
May	0	1.588
June	0	1.706
July	0	1.701
August	0	1.707
September	0	1.64
October	0	1.452
November	0.11	0
December	0.208	0
C.O.P	At heating = 4.3	At Cooling = 4.5

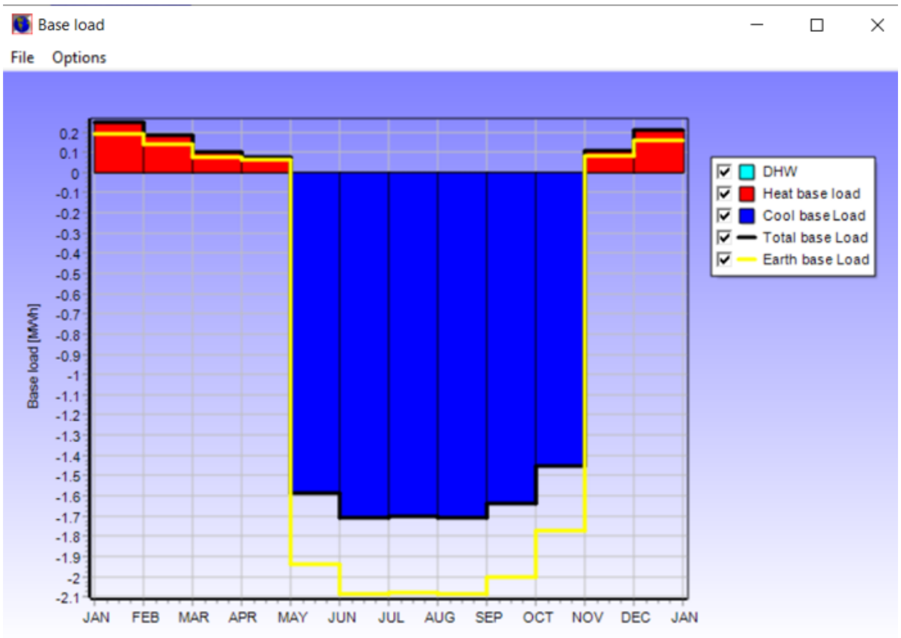


Figure 71: Acoustics Building Base load annually.

- ***Peak loads:***

Month	Heating (KW)	Cooling (KW)	Duration (h)
January	51	0	2
February	49.4	0	1
March	46.2	0	1
April	44.6	0	1
May	0	112.4	8
June	0	117.9	9
July	0	117.4	10
August	0	119.4	8
September	0	115.8	7
October	0	105.4	5
November	47.4	0	1
December	50.2	0	1

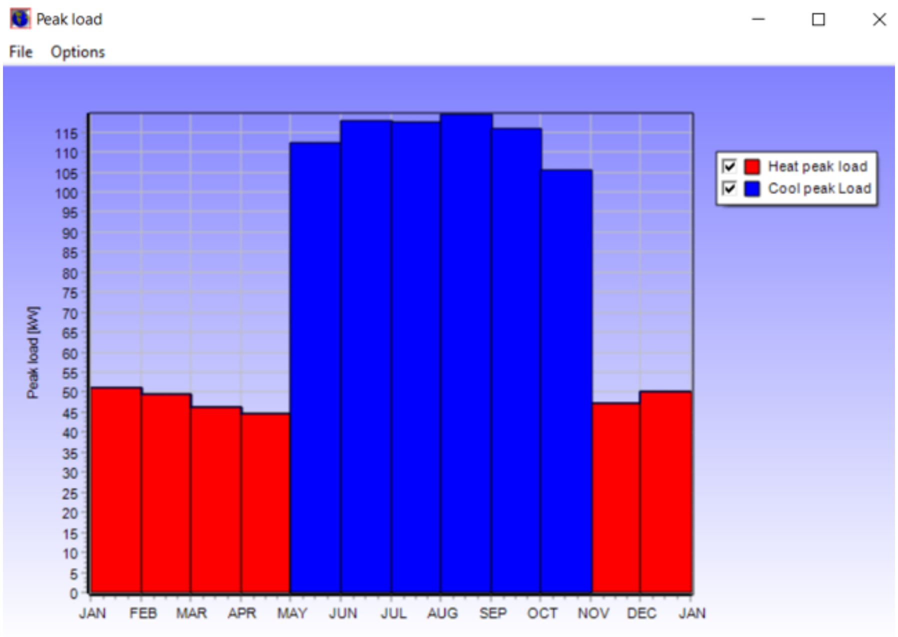


Figure 72: Acoustics Building Peak loads annually.

3.3.2 Results

3.3.2.1 Design Geometry Dimensions

Best Case Results	
No. of boreholes	*18 boreholes*
Boreholes configuration	*3 * 8 configuration*
Spacing between boreholes	$7\ m$
Depth of one borehole	$98\ m$
Total length	$1759\ m$
Land area	$686\ m^2$
Surface area dimensions: L × W	$49\ m \times 14\ m$

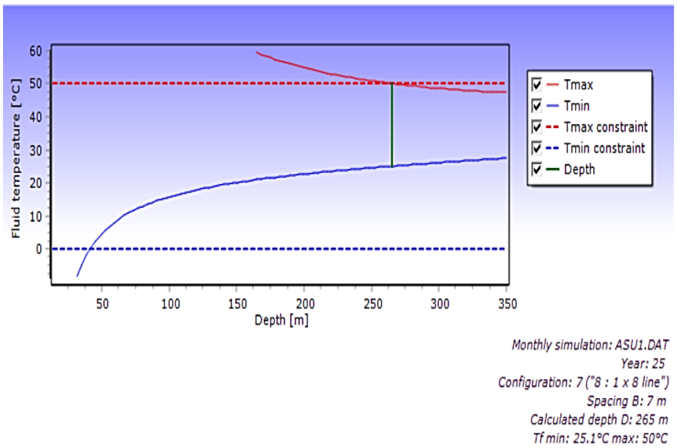


Figure 74: Fluid Temperature vs depth [For Acoustics Building]

Figure 73: Borehole field configuration [For Acoustics Building]

3.3.2.2 Acoustic Building Cost Assessment

Cost	
Excavation & instruments	*1300 LE for 1 m*
Total cost per 1759 m = 2,286,700 LE	
Two heat pump units used	*Ecoforest (100 kw- 40 kw)*
Total cost of two heat pump respectively** (957,372 + 17946) **= 975,318 LE	
Total system cost = 3,262,018 LE	

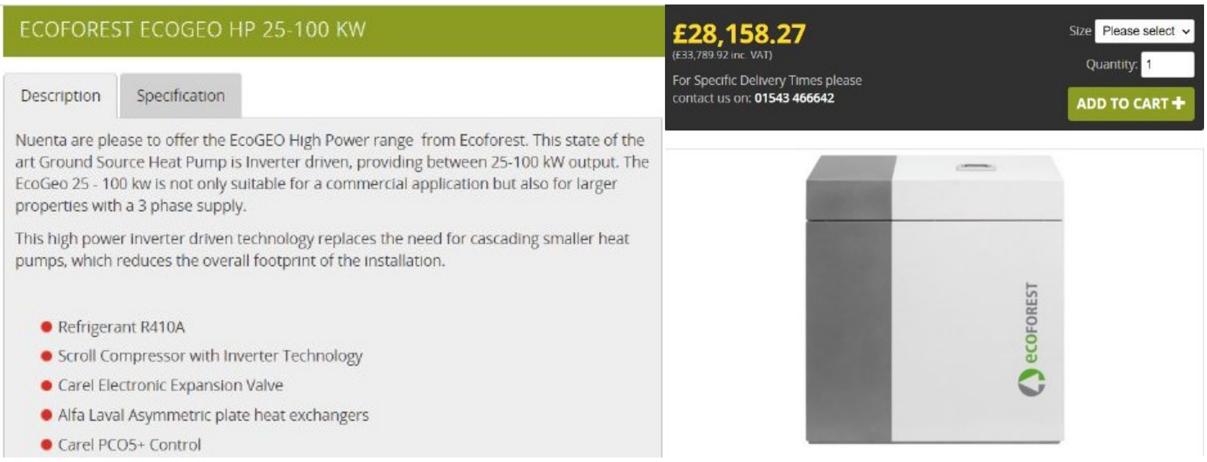


Figure 75: Acoustic Building Heat pump specifications

3.4. Prototype- Ain Shams University Case Study:

3.4.1 Prototype Experimental Results

The prototype was made to operate for 24 hours, and data was recorded for each hour.
Pin locations and prototype sensor configuration is found in Figure.28 in Section 2.5.4.5

From readings, records have shown that:

- Pin. (4) was put at depth 45 cm from the soil's surface as the soil 's height is 75 cm and Pin (2), Pin (3) and Pin (4) has a vertical separation distance of 10 cm between each other.

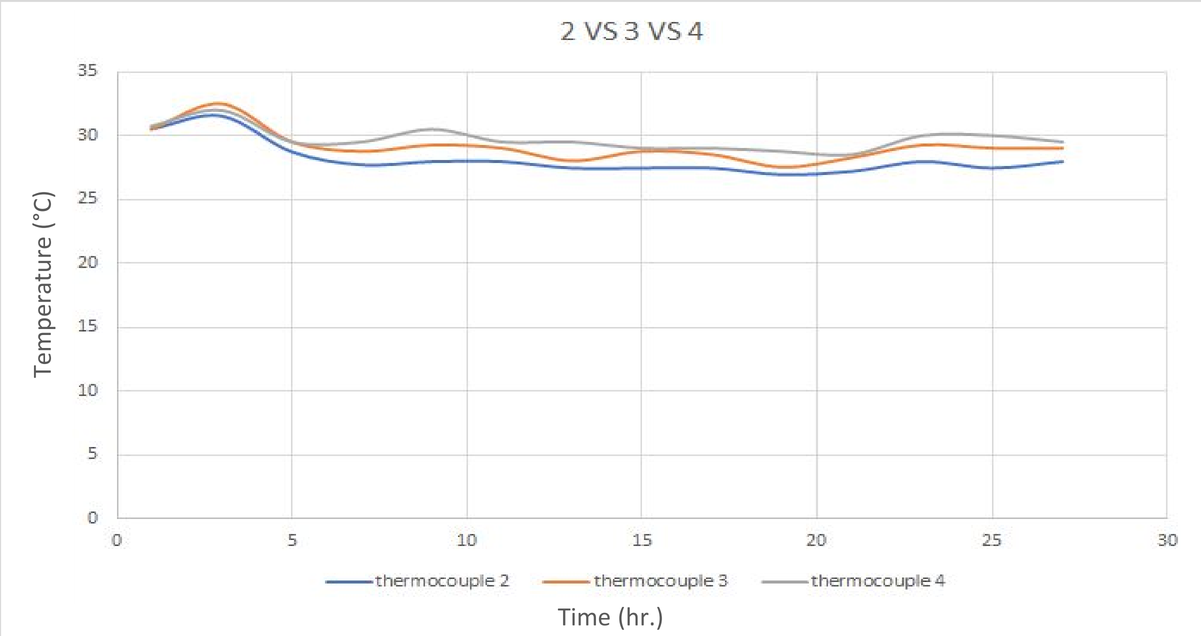


Figure 76: Pin (2), (3) & (4) – (Temperature vs Time) readings

For every *10 cm* in the vertical direction, there is a temperature difference of about $[0.5 - 1.5]\ °C$

- Pin (1) and (5) was put at depth 35 cm from soil's surface & are located at the box corners.

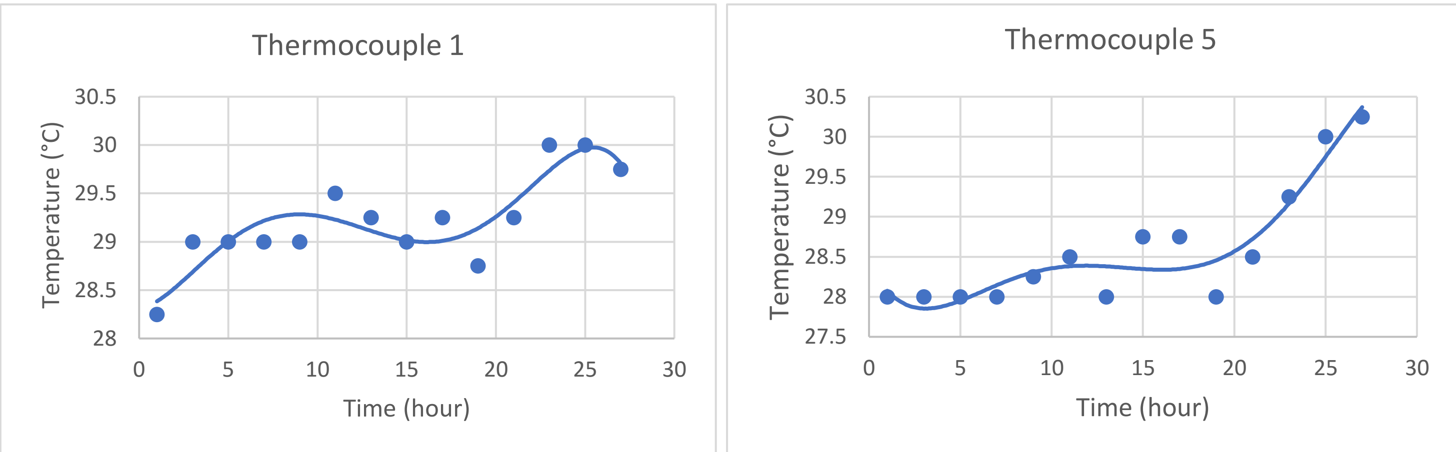


Figure 77: Pin (1) & Pin (5) – (Temperature (°C) vs Time (hr.)) readings

During the 27-hour experiment: pin 1 temperature increased by around 1.5$°C$, while pin 5 temperature increased by 2.2$°C$

- Pin. (6) was put at the soil's surface, where results have shown that:

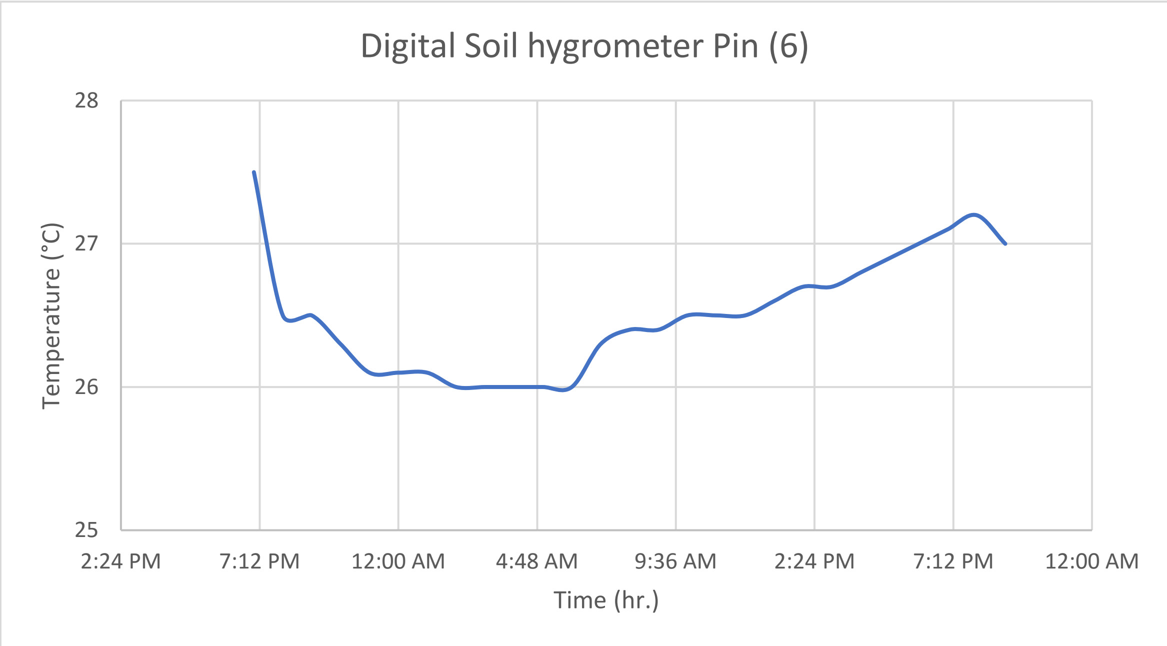


Figure 78: Pin (6) – (Temperature vs Time) readings

From Pin (6), the soil temperature range between $[26 - 27]\ °C$ at the soil's surface throughout the experiment.

Figure 79: Pin (6) positioning

To understand the diffusion time made by experiment, from figure 78:

Diffusion time started at 6:00 am and changed temperature from $26°C$ to $27°C$ after 12 hours, which shows that:

$$\boxed{Experimental\ diffusion\ time = 0.5\ day}$$

For knowing thermal diffusivity = $\mathbf{0.058\ m^2/day}$:

$$\boxed{Theoretical\ Diffusion\ time = \frac{(Characterictic\ Length)^2}{Thermal\ Diffusivity} = \frac{(0.18)^2}{0.058} = 0.558\ day}$$

$$\%\ Error = \frac{0.5\ - 0.558}{0.5} \times 100 = \ -11.6\%$$

- Pin (7) and Pin (8) represents the inlet and outlet water temperature entering and exiting from the heat pump, where results have shown that:

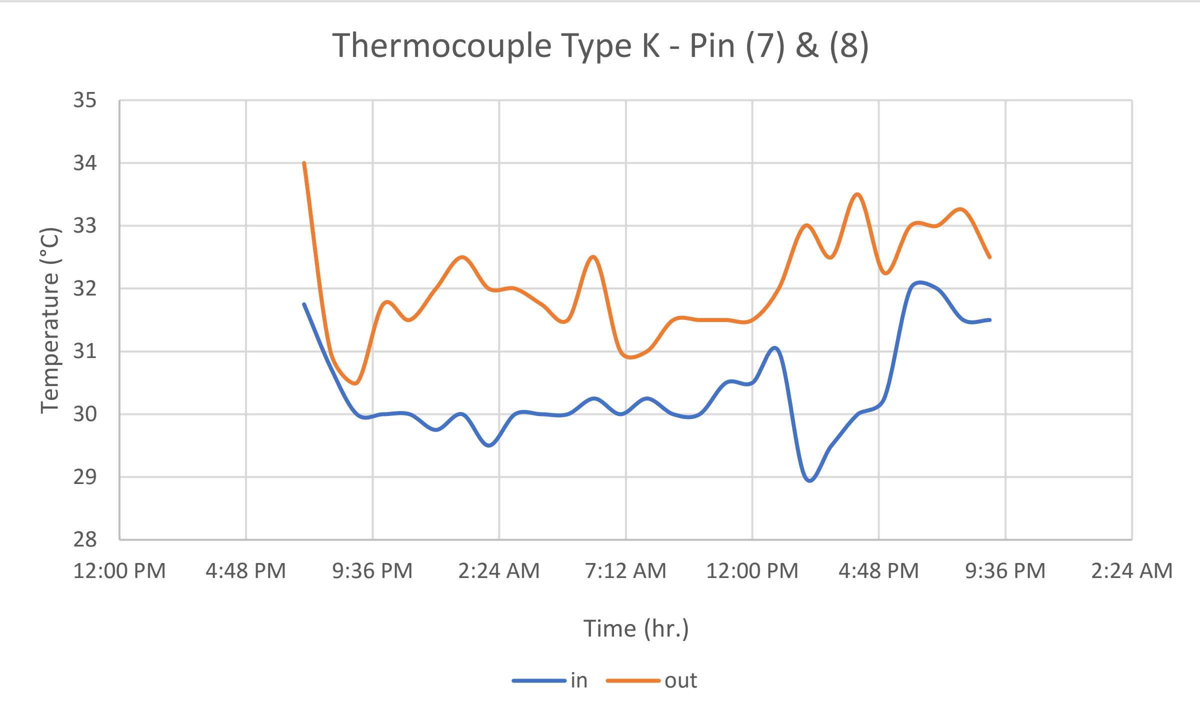


Figure 80: Pin (7) & Pin (8) – (Temperature vs Time) readings

As shown in the graph, T_{in} varies from 29°C to 32°C, while T_{out} varies from 30°C to 34°C

To explain the value of the temperature readings, the effectiveness of the ground heat exchanger should be measured, where:

$$\theta\ (\%) = \frac{T_{in} - T_{out}}{T_{in} - T_{ground}} \times 100$$

By considering T_{ground} is a function of pin (6), then from the following graph, valuable data can be analysed to understand the performance of the prototype system:

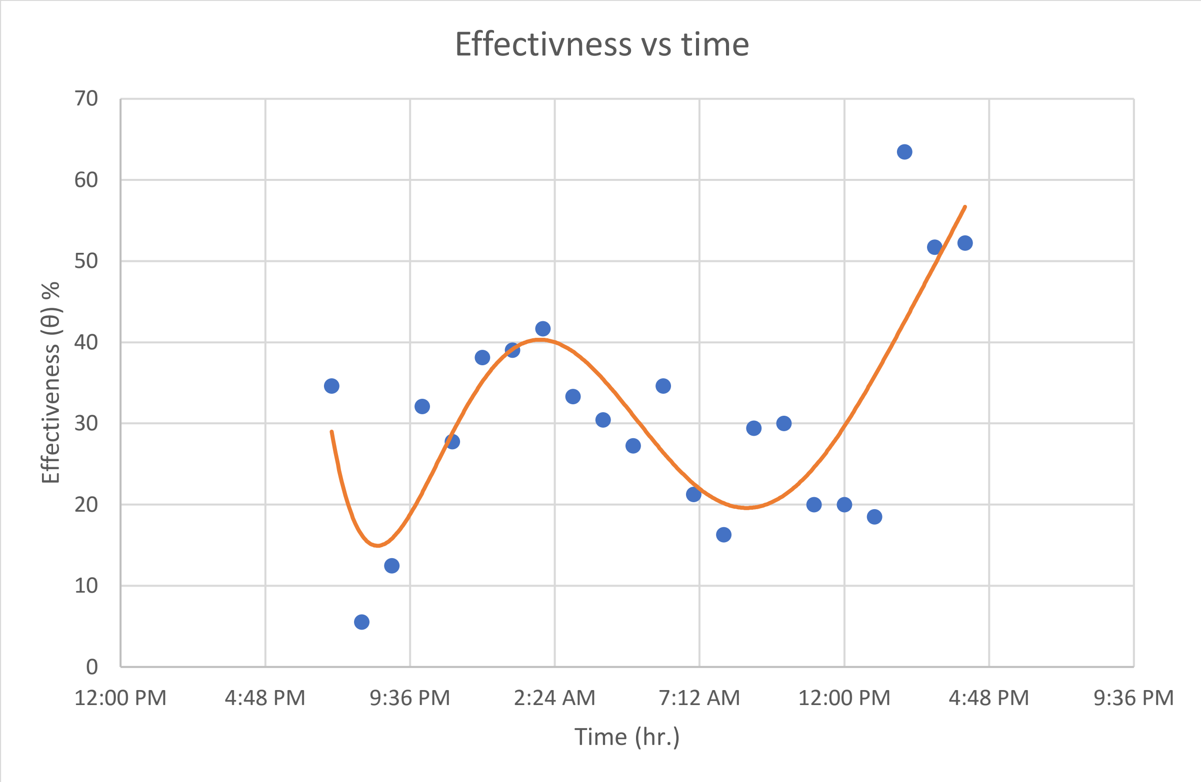


Figure 81: Effectiveness vs time

Effectiveness increased during the test from 35% to 62%

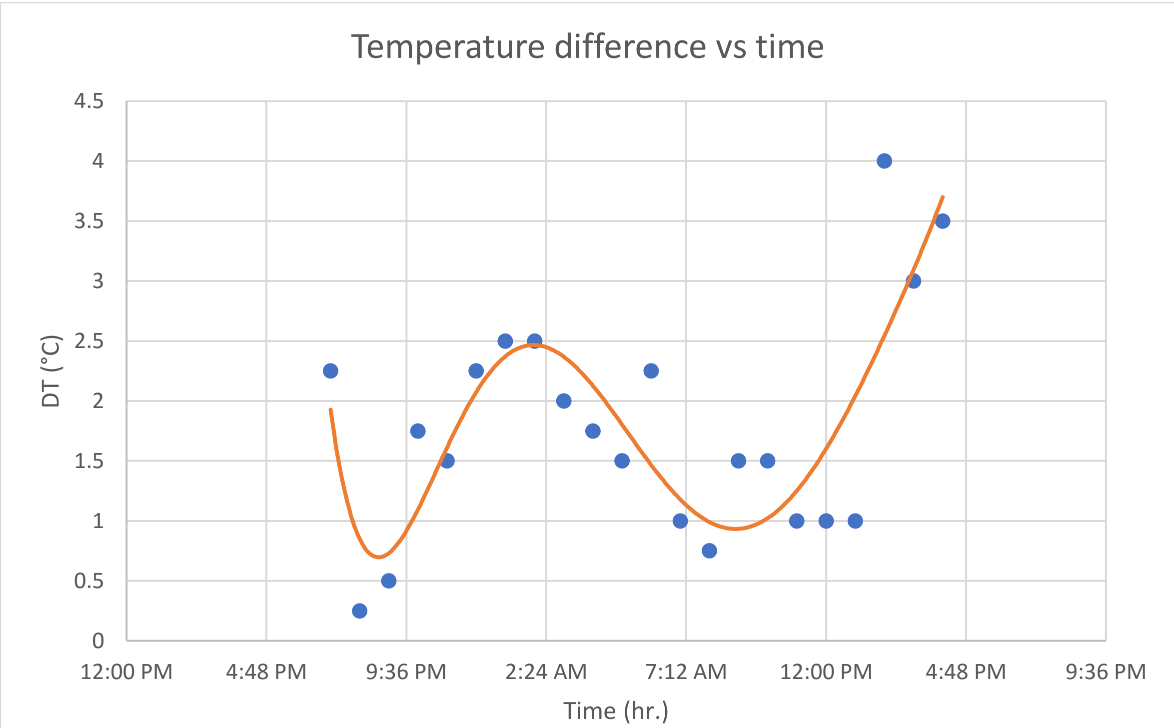


Figure 82: Temperature difference vs time

As shown in the figure, the temperature difference ranges from $0.25°C\ to\ 4°C$.

3.4.2 Prototype Performance Results

To study the prototype performance, observing the refrigerant ability of cooling water by setting a requirement of calculating the power consumption of cooling 1.5 L of water and to understand the success of the prototype's (refrigerant – water) cooled system, a comparison is made for (refrigerant – air) cooled system.

- ***Water Cooled System:***

Water-cooled water dispenser Inputs		
Circulating diaphragm DC pump	*1.2 L/min.*	*0.24 Ampere*
Plate Heat Exchanger	*10 plates*	
Refrigerant	*R-134a*	

- ***Test Results:***

The 1.5 L of water cooled *from* $\boldsymbol{31°C}$ *to* $\boldsymbol{19°C}$ in ***0.77 Hour (46.2 min)***

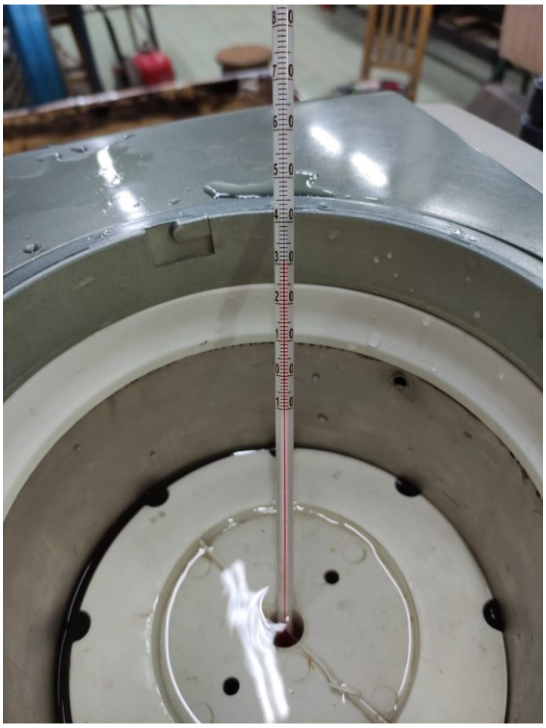


Figure 84: Thermometer reading before cooling (31°C)

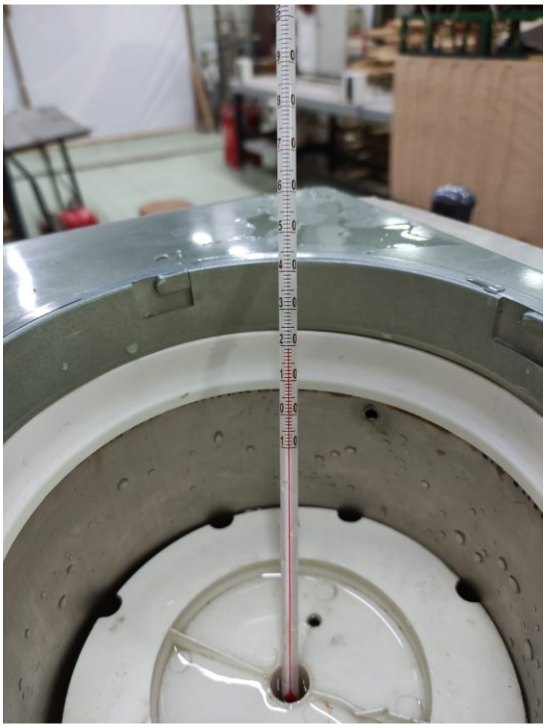


Figure 83: Thermometer reading after cooling (19°C)

- ***Calculations for getting power consumption:***

1. *Work of the compressor (W_C) = AC Voltage * Current * Power Factor*

$$W_c = 220v * 0.57A * 0.8 = 100.32\ watt$$

2. *Cooling power (Q_L) =* $m_w * c_w * (T_{initial} - T_{final})$

$$Q_L = 1.5\ kg * 4180\ J/(kg.k) * (31^oC - 19^oC) = 75{,}240\ Joule$$

$$Q_L = \frac{75240\ Joule}{0.77\ Hour * 3600\ Sec} = \boldsymbol{27.14\ watt}$$

3. *Rejected power to the ground* $(Q_H) = Q_L + W_c$

$$Q_H = 27.14watt + 100.32watt = 127.56\ watt$$

4. *Compressor Power consumption = Compressor work * Operating hours*

$$Compressor\ power\ consumption = 100.32watt * 0.77hour = 77.25\ Wh$$

$$Circulating\ power\ consumption = 0.24\ Ampere * 24DC\ volt * 0.77Hour = 4.43\ Wh$$

$$Total\ power\ consumption = 77.25\ Wh + 4.43\ Wh = \mathbf{81.6816\ Wh}$$

- ***Air Cooled System:***

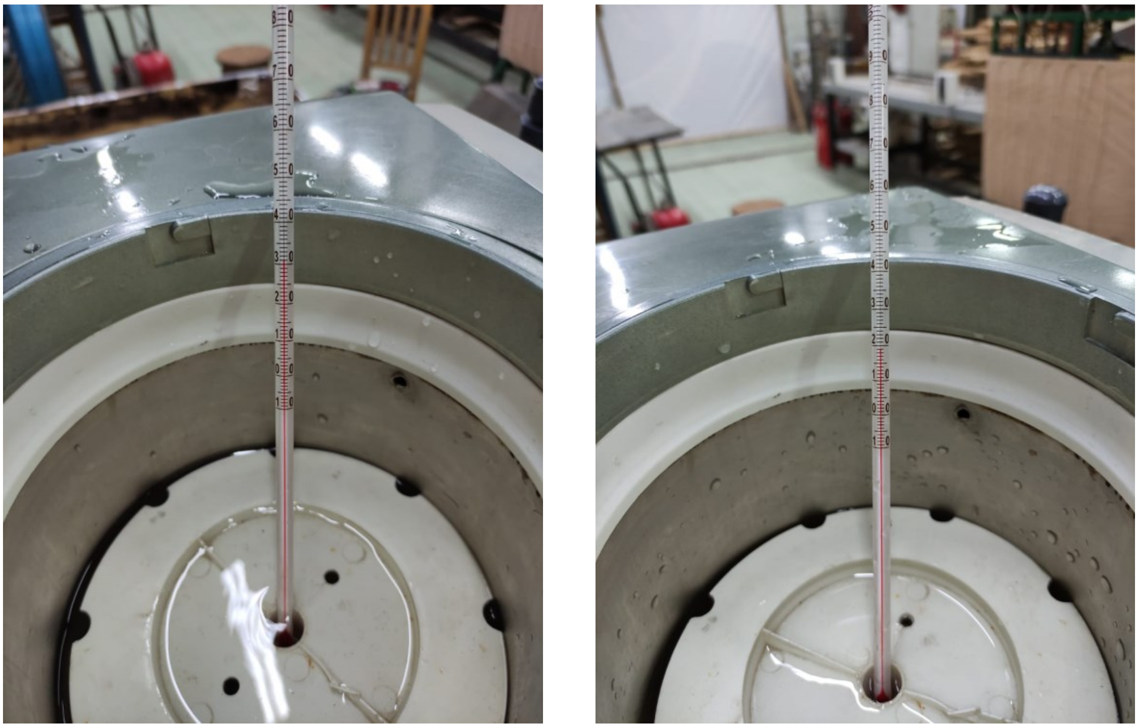
Air-cooled water dispenser Inputs	
Cooling Water Ability	2 Liter/hr.
Rated Cooling Current	0.6 Ampere
Cooling Power (Q_L)	90 Watt
Refrigerant	R-134a

- ***Assumptions:***
  - Power Factor of the dispenser = 0.8

Figure 85: Air-Cooled Specifications

- ***Test Results:***

  The 1.5 L of water cooled *from* $\mathbf{31°C}$ *to* $\mathbf{19°C}$ in ***<u>0.9 Hour (56 min)</u>***



(a) (b)

Figure 86: Air-Cooled Thermometer reading (a)before cooling = 31°C - (b) after cooling = 19°C

Ammeter Reading = ***<u>0.57 A</u>***

Figure 87: Air-Cooled Avometer reading.

- ***<u>Calculations for getting power consumption:</u>***

1. *Work of the compressor (W_C) = AC Voltage * Current * Power Factor*

$$W_c = 220v * 0.57A * 0.8 = 100.32\ watt$$

2. *Cooling power (Q_L) =* $m_w * c_w * (T_{initial} - T_{final})$

$$Q_L = 1.5\ kg * 4180\ J/(kg.k) * (31^oC - 19^oC) = 75{,}240\ Joule$$

$$Q_L = \frac{75240\ Joule}{0.9\ Hour * 3600\ Sec} = \mathbf{23.22\ watt}$$

3. *Rejected power to the ground (Q_H) =* $Q_L + W_c$

$$Q_H = 23.22watt + 100.32watt = 123.54\ watt$$

4. *Compressor Power consumption = Compressor work * Operating hours*

$$Compressor\ power\ consumption = 100.32\ watt * 0.9hour = \mathbf{90.288\ Wh}$$

P.O.C	*Water-Cooled*	*Air-Cooled*
Cooling Energy (Q_L)	20.897 Wh	20.897 Wh
Total energy consumption	81.682 Wh	90.288 Wh
COP	0.27	0.23

3.4.3 Prototype Experimental Validation

3.4.3.1 Temperature difference Validation

Experimental Data Inputs		
Ground Temperature	$T_{g(average\ of\ all\ pins)} = 28.5\ °C$	
Pipe Selection (1-inch UPVC Pipe)	$D_i = 28.4\ mm$	$D_o = 33.5\ mm$
Pipe Thermal Conductivity	$k_p = 0.19\ W/(m.k)$	
Trench Dimensions ($L \times W \times H$)	$(2 \times 1 \times 1.5)\ m$	
Total pipe length (L_t)	$26\ m$ *[Buried 4 loops and a riser]*	
Maximum pump flow rate (from market)	$\dot{V} = 1.2\ L/min$ or $\dot{m}_w = 0.0204\ kg/s$	
Heat Capacity of water (C_w)	$4180\ J/(kg.k)$	
$T_{w,in(average)}$	$32\ °C$	

- ***From governing equations:***

$$L_t = (\dot{m}_w * C_w * R_{tot}) * \ln\left[\frac{T_{w,in} - T_g}{T_{w,out} - T_g}\right]$$

As, $T_{w,in} = water\ temp\ entering\ ground\ \ (Hotter)$

$$T_{w,out} = water\ temp\ leaving\ ground\ \ (colder)$$

- ***To find R_{total}:***

$$\because R_{tot} = \mathrm{R_{conv}} + \mathrm{R_{pipe}} + \mathrm{R_{soil}}$$

$$\mathrm{R_{conv}} = \frac{1}{\pi D_i h_w}, \qquad \mathrm{R_{pipe}} = \frac{\ln\left(\frac{D_o}{D_i}\right)}{2\pi K_{pipe}}$$

- **Water Properties from water table** ($at\ T_{w,avg.}$):

$$\rho_w = 999\left(\frac{\mathrm{kg}}{\mathrm{m^3}}\right), \mu = 0.769 * 10^{-3}\left(\frac{\mathrm{kg}}{\mathrm{ms}}\right), \mathrm{Pr} = 5.2, \mathrm{k_w} = 0.62\left(\frac{\mathrm{W}}{\mathrm{mK}}\right)$$

$$Water\ Velocity\ in\ pipes = V = \frac{\dot{V}}{A_p}$$

$$V = \frac{1.2 * 10^{-3}}{60 * \frac{\pi}{4}(28.4 * 10^{-3})^2} = 0.0316\left(\frac{m}{s}\right)$$

$$N_u = \frac{h_w * D_i}{K_w} = 0.023 * \left(\frac{\rho_w * V * D_i}{\mu}\right)^{0.8} * (Pr)^{0.3}$$

$$\frac{h_w * 28.4 * 10^{-3}}{0.62} = 0.023\left(\frac{999 * 0.0316 * 28.4 * 10^{-3}}{0.769 * 10^{-3}}\right)^{0.8} * (5.2)^{0.3}$$

$$h_w = 233.84\left(\frac{W}{m^2.K}\right)$$

$$\therefore \mathrm{R_{conv}} = \frac{1}{\pi * 28.4 * 10^{-3} * 233.84} = 0.0479\left(\frac{m\,.\,k}{W}\right)$$

$$\therefore\ \mathrm{R_{pipe}} = \frac{\ln\left(\frac{33.5}{28.4}\right)}{2\pi * 0.19} = 0.1383\left(\frac{m\,.\,k}{W}\right)$$

- ***According to GHX tool:***

$$\therefore \mathrm{R_{soil}} = 0.0742\left(\frac{m\,.\,k}{W}\right)$$

$$\therefore \mathrm{R_{total}}\ = \ 0.0479\ +\ 0.1383\ +\ 0.0742 = \mathbf{0.2604}\left(\frac{\boldsymbol{m\,.\,k}}{\boldsymbol{W}}\right)$$

- ***Substituting in the governing equation to find*** $T_{w,out}$***:***

$$L_t = (\dot{m}_w * C_w * R_{tot}) * \ln\left[\frac{T_{w,in} - T_g}{T_{w,out} - T_g}\right] [27]$$

As, $\quad T_{w,in} = water\ temp\ entering\ ground\ \ (Hot)$

$$T_{w,out} = water\ temp\ leaving\ ground\ \ (cold)$$

$$\therefore T_{w,out} = 29.586\ °C$$

$$\boxed{\Delta \mathrm{T_{w,(average)}} = T_{w,in} - T_{w,out} = 2.414\ °\mathrm{C}}$$

3.4.3.2 Plate Heat Exchanger Validation

- ***BHEx Capacity:***

$$\boxed{\mathrm{Q = A * U * \Delta T_{LM}}}$$

- ***BHEx Size:***

BHEx configuration
A = 30 cm
B = 12 cm
No. of plates = 10
Total Area = 0.36 m^2

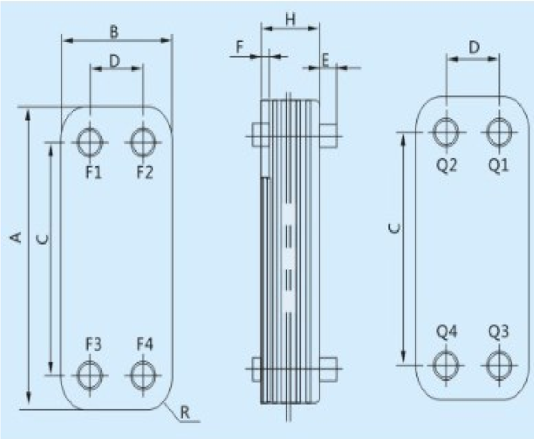


Figure 88: BHEx Configuration

Overall heat transfer coefficient:

$$\boxed{\mathrm{U} = \frac{1}{\frac{1}{h_w} + \frac{S}{k} + \frac{1}{h_f}}}$$

s = plate thickness = 0.2 mm

k = *plate* thermal conductivity = 15 $W/m.k$

h_w = The heat transfer convection coefficient of water $(W/m^2.k)$

h_f = The heat transfer convection coefficient of freon $(W/m^2.k)$

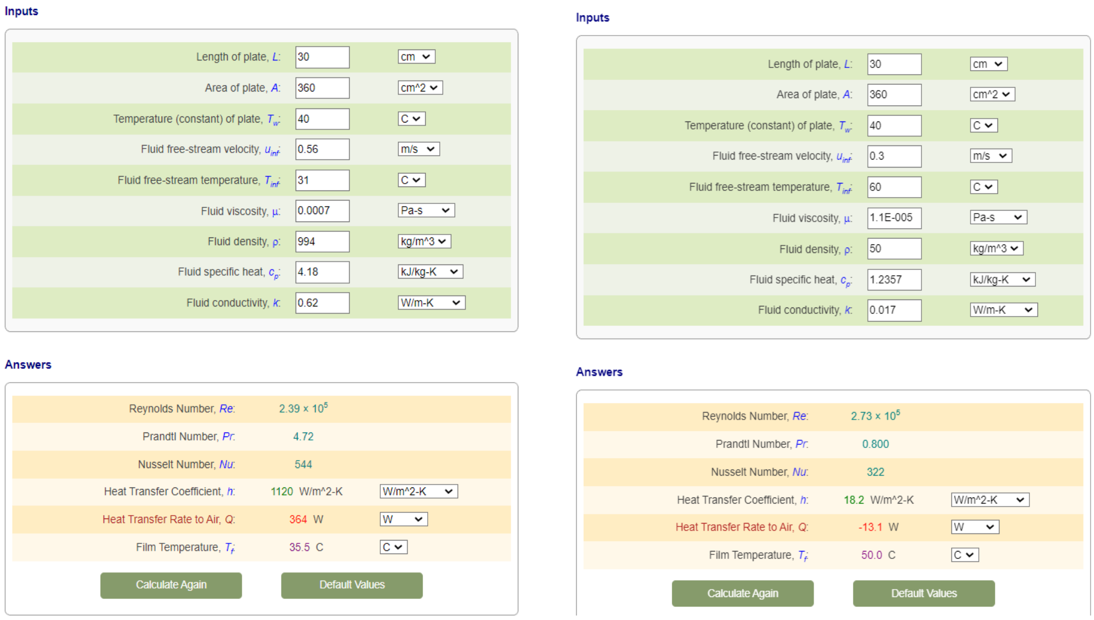


Figure 89: BHEx Inputs and Results

$$U = \frac{1}{\frac{1}{1120} + \frac{0.3 * 10^{-3}}{15} + \frac{1}{18.2}} = 17.9\, W/m^2.k$$

- **ΔT_{LM}:**

$$T_f = 60\,°C,\quad T_{wi} = 31\,°C,\quad T_{wo} = 33.5\,°C$$

$$\boxed{\therefore \text{ BHEx capacity} = \text{Q} = 0.36 * 17.9 * \frac{(60-31)-(60-33.5)}{\ln\frac{(60-31)}{(60-33.5)}} = \mathbf{178.75\ Watt}}$$

3.4.4 Prototype CFD Setup

3.4.4.1 Initial and Boundary Conditions

The initial ground and GHE temperatures and the far-field boundary temperature, which was equal to the undisturbed ground temperature, was modelled here as a uniform temperature, where:

$$T_{ground} = 25\ °C \qquad \boldsymbol{eq.(I)}$$

The inlet fluid temperature to the soil was assumed to be constant of value as $T_{in} = 33.5\ °C$, while the output fluid temperature acts as a time-dependent carrier fluid temperature which was obtained from the numerical model and the prescribed time-dependent heat transfer equations. This effectively acts as the transfer function of a heat pump that receives the fluid at a certain temperature and rejects heat, thus changing the temperature of the fluid, which is reinjected into the ground.

$$T_{out(t)} = T_{in} - \frac{Q_{GHE}}{\rho_W v A\ C_{p,w}} \qquad \boldsymbol{eq.(II)}$$

A boundary condition of the fluid flow rate at the inlet pipe (s) of about 1 L/min. or 0.03 m/s and from the pipe geometry, the flow is laminar flow.

$$v_{in} = 0.03\ m/s \qquad \boldsymbol{eq.(III)}$$

A reference atmospheric pressure in the outlet pipe (s) for the purpose of forced convection was assumed:

$$p_o = 101{,}325\ Pa \qquad \boldsymbol{eq.(IV)}$$

There were thermal insulation conditions on the inner sides of the box as well as the pipe walls are no slip condition.

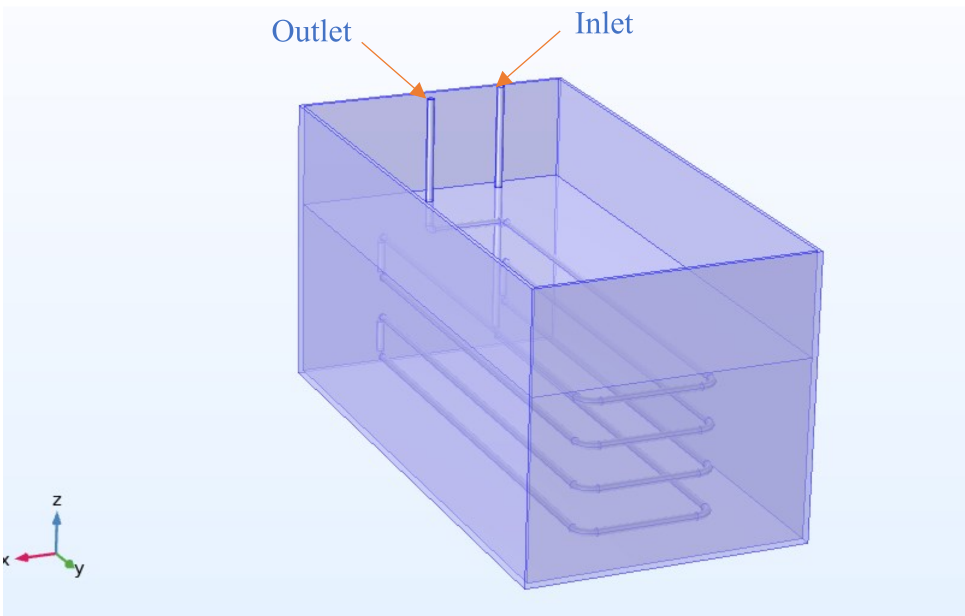


Figure 90: Prototype CAD on COMSOL software

3.4.4.2 Meshing

The following Figure.76 shows the meshing of the prototype:

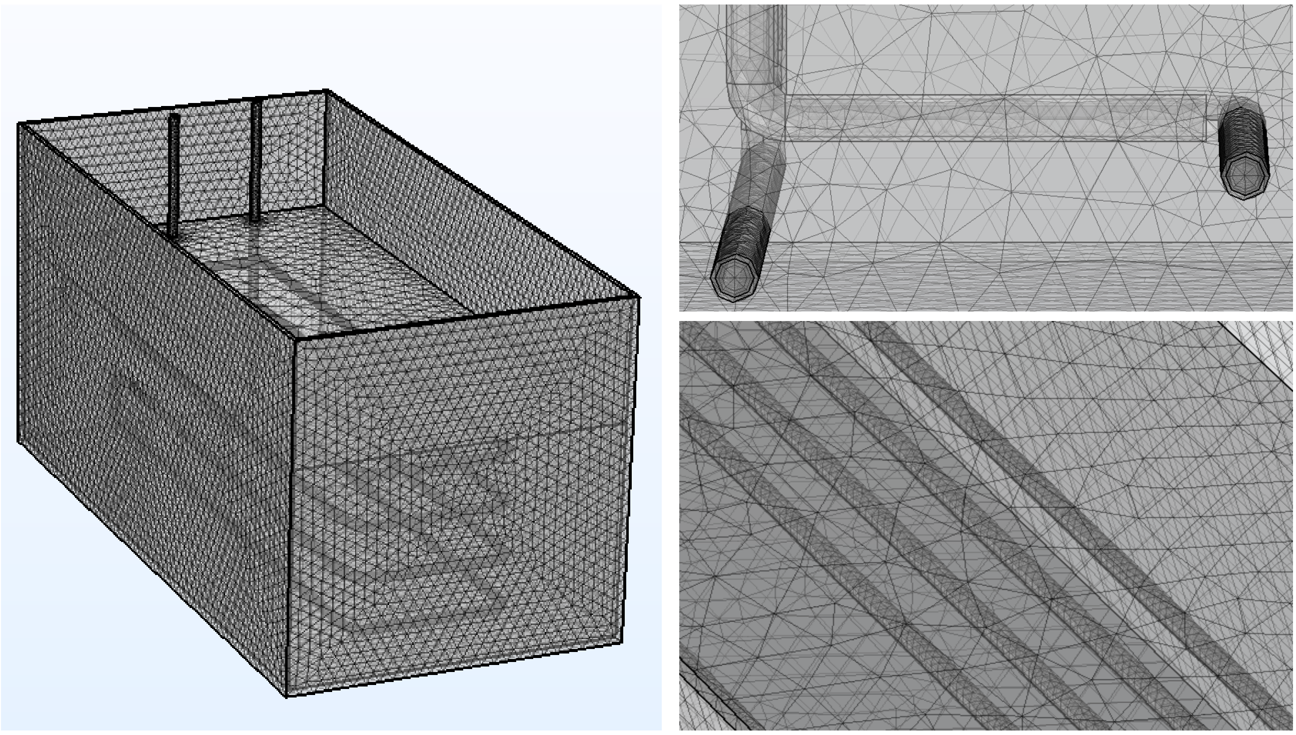


Figure 91: ASU Prototype Meshing

3.4.5 Prototype CFD Results

3.4.5.1 Steady State Solution

To be able to make a study of stationary solution, adding another boundary condition should be considered as in this case, the soil domain will be given a constant soil temperature along the simulation. So, a steady state is created between the inlet water temperature to the soil and the soil temperature. Thus, considering the soil in the box as an infinite heat sink similar to the ground behaviour as the soil temperature theoretically shouldn't change.

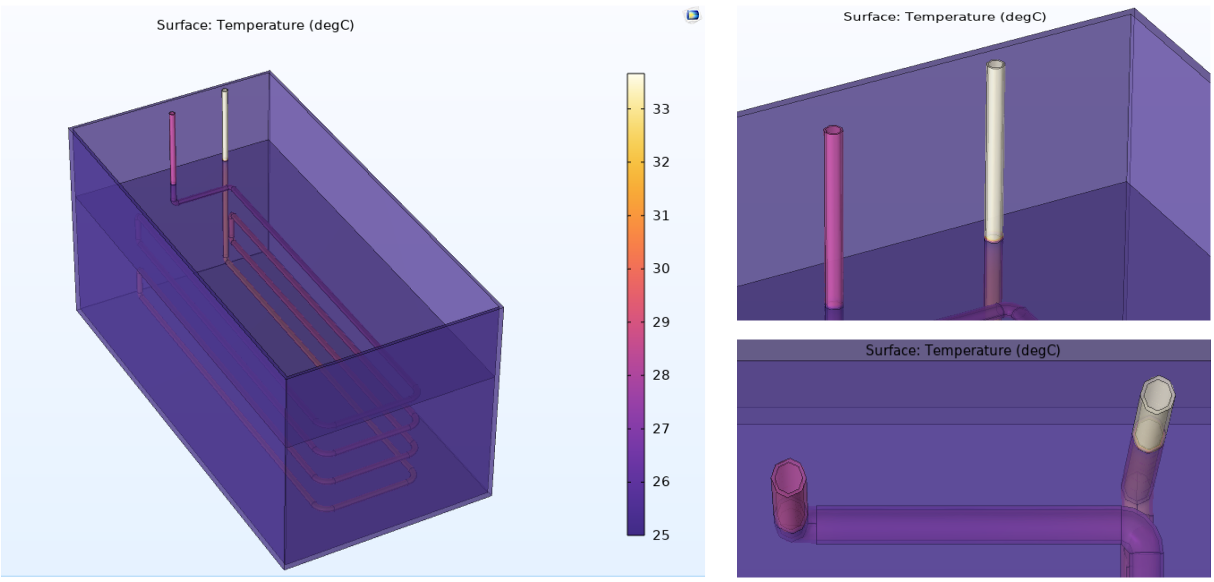


Figure 92: Steady state solution - Temperature distribution (°C)

Results have shown that the outlet water temperature from the soil:

$$T_{out} = 28.896\ °C$$

And since the inlet fluid temperature is $33.5\ °C$ and the undistributed ground temperature is $25\ °C$, then the simulated effectiveness will be:

$$\theta\ (\%) = \frac{33.5 - 28.896}{33.5 - 25} \times 100 = 54.16\%$$

And the temperature difference is:

$$\Delta T = 33.5 - 28.896 = 4.604\ °C$$

3.4.5.2 Transient Solution

On the contrast of the previous study, the study of the soil thermal behaviour is being simulated by assuming certain operation conditions which are:

- Supplying constant inlet fluid temperature which is equal to $33.5\ °C$
- Soil as a heat sink will have its wall insulated which means the soil will store heat with respect to time.
- The study will be done twice but with different time intervals.

I. 24-hour Time Interval Results:

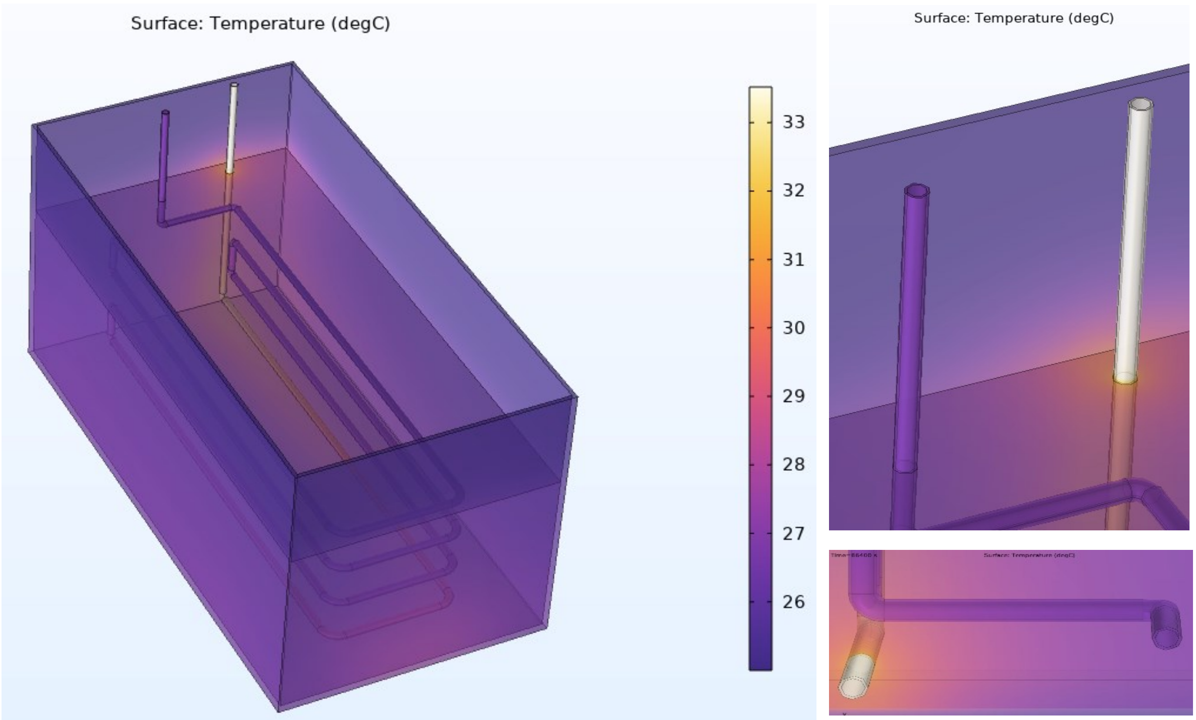


Figure 93: Transient solution after passing 24 hours- Temperature distribution (°C)

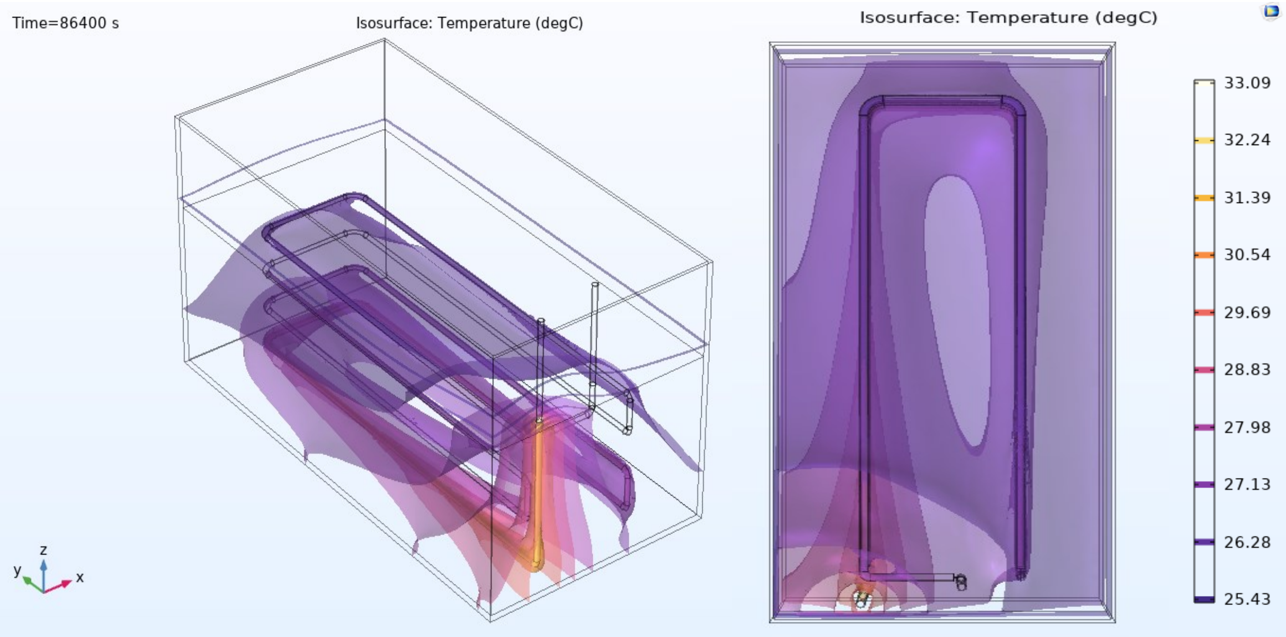


Figure 94: Isothermal Contours of prototype after 24 hours

Results have shown that the rate of heat energy (power) that has been stored in the ground – which can be used if considering the soil as thermal storage - can be described from the following curve:

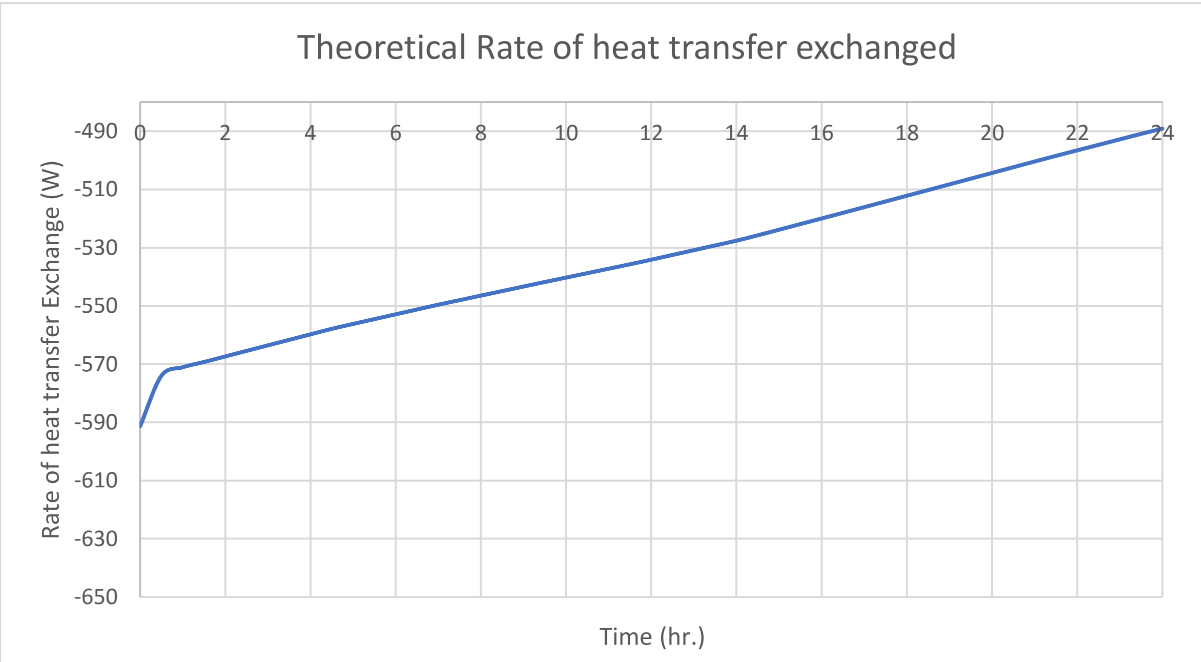


Figure 95: Rate of Heat transfer Exchange Vs Time - 24 hr. simulation

By using surface integration for power per unit area has a function of:

$(T - 306.65) * spf.U * spf.rho * comp1.mat1.def.Cp(T)$ as unit is in Watts (W)

Also, Results have shown that the outlet water temperature from the soil:

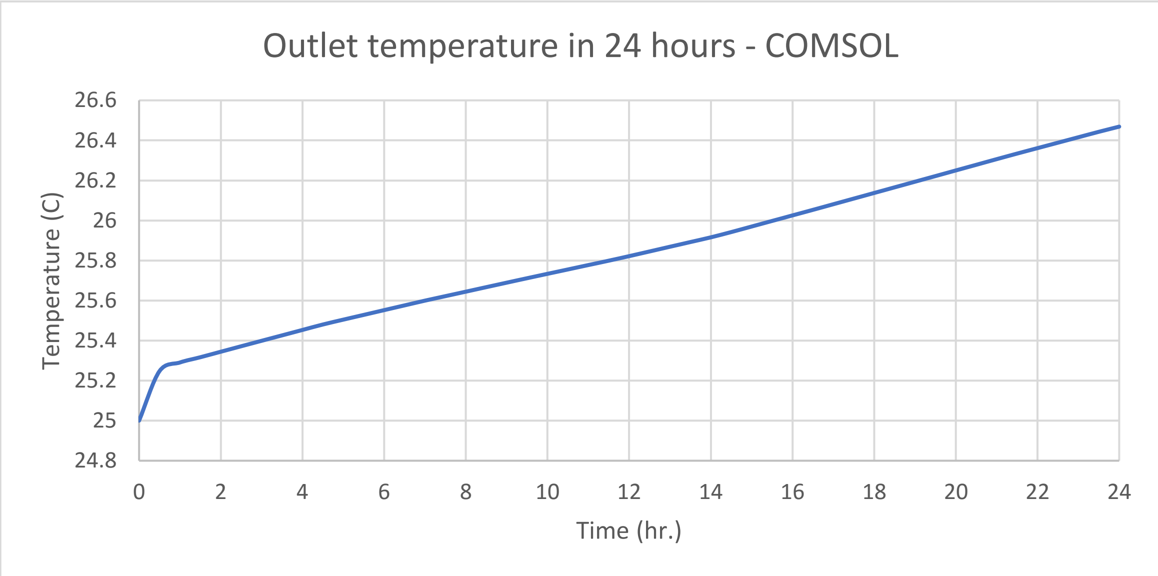
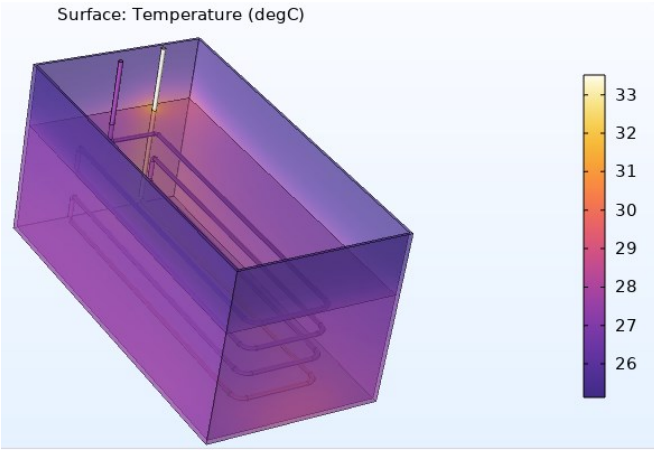
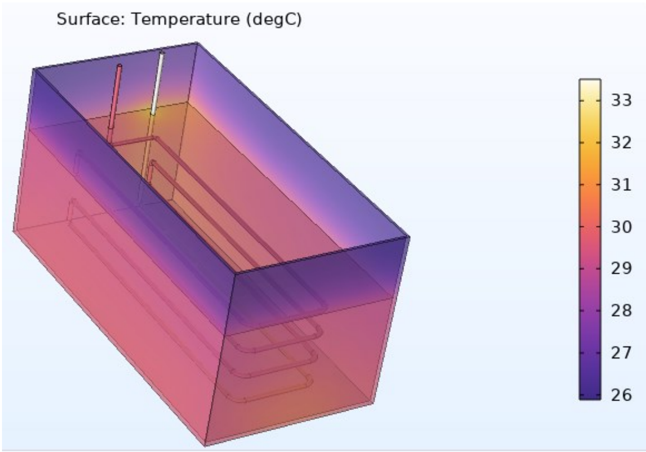


Figure 96: Temperature vs Time - 24 hr. simulation

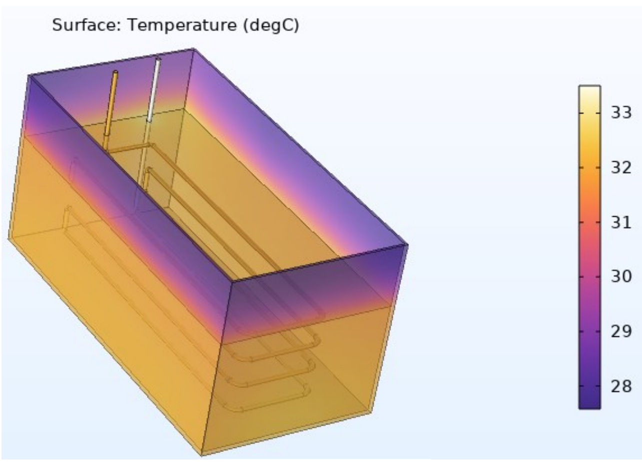
II. ***<u>30 Days' Time Interval Results:</u>***



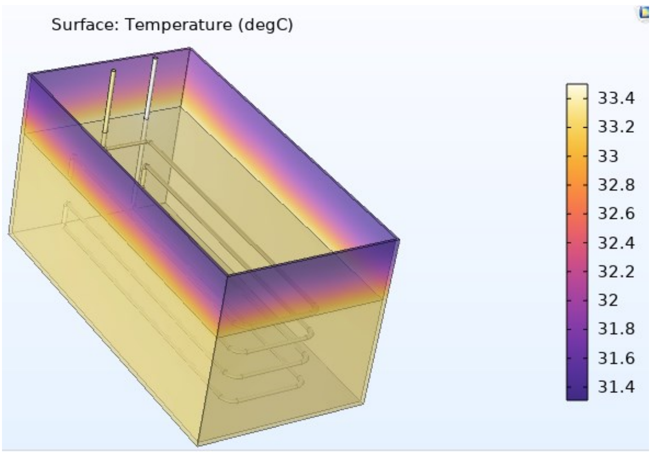
After 2 days



After 5 days



After 10 days



After 20 days

Figure 97: Transient solution at different times - Temperature distribution (°C)

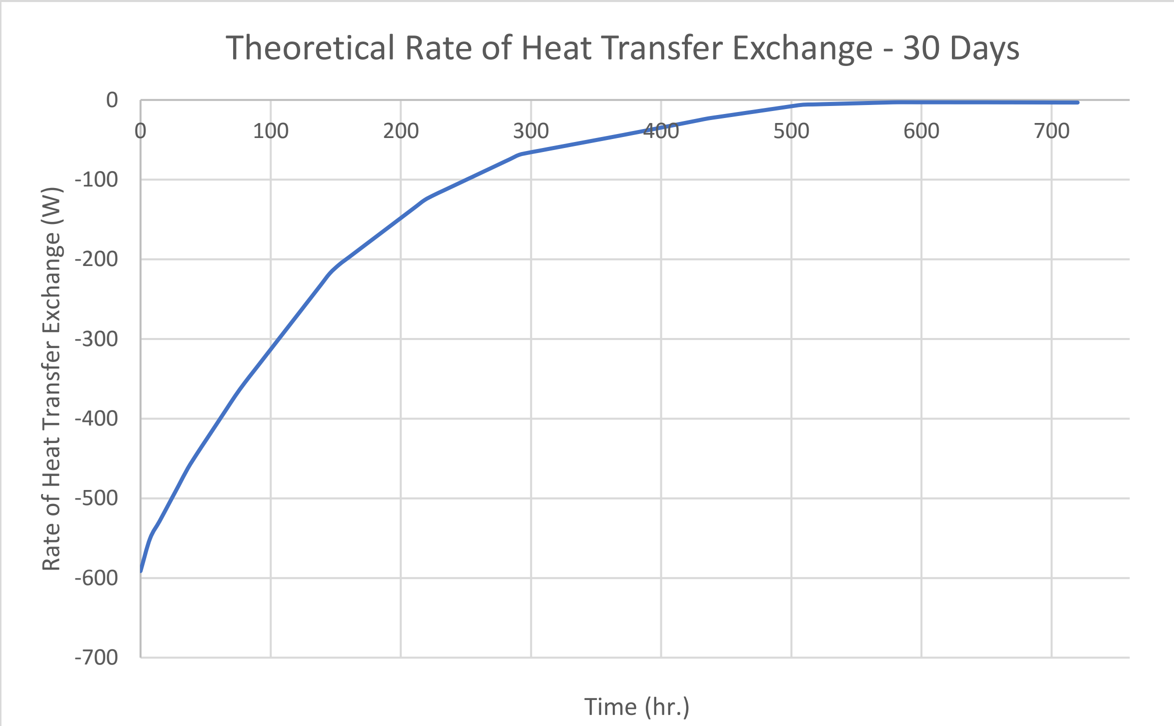


Figure 99: Rate of heat transfer Exchange Vs Time - 30 days' simulation

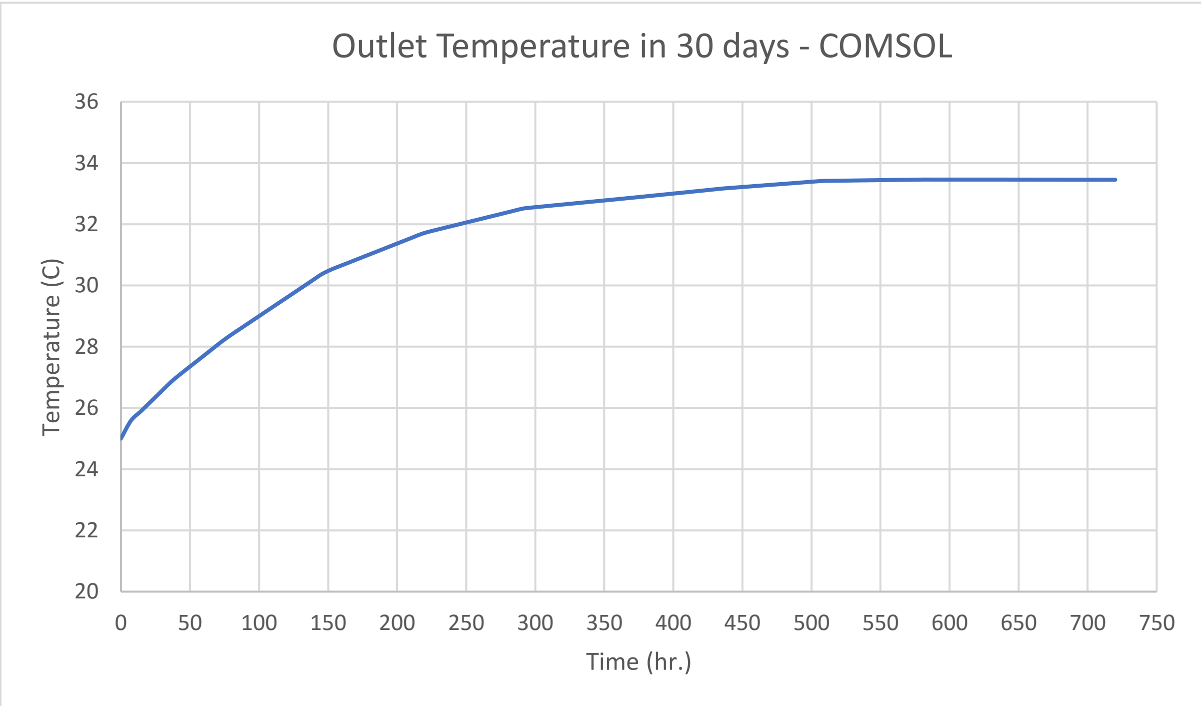


Figure 98: Temperature vs time - 30 days' simulation

Discussion:

Geothermal technology has been studied for its potential application in cooling and heating air conditioning systems. The study focused on implementing geothermal technology in the computer laboratory at Cairo University. The results of the study showed that to achieve the necessary load for cooling and heating, four boreholes at a depth of 35 meters, covering an area of 49 square meters, were required. The study found this to be a suitable solution for the available space at the university. The feasibility of implementing this project was also assessed, and the results showed that geothermal heat pumps could be installed instead of conventional air conditioning to achieve a payback period of 51 years. This indicates that the technology is not feasible in case of reducing costs over the operational lifetime of a small-scale system but can significantly reduce carbon emissions.

According to recent research, about 75% of carbon emissions produced by air conditioning systems can be attributed to the electrical consumption of the compressor. This consumption is typically generated by natural gas power plants, as most power plants in Egypt rely on natural gas. The building sector, and in particular the tertiary buildings of Cairo University, have a significant impact on climate change, as shown in Figure 38.

A study published in the journal Energy and Buildings found that carbon emissions per kWh for air conditioning in India ranged from 0.5 to 1.4 kgCO2/kWhth, depending on the type of air conditioning system used. In Singapore, the range was 0.28 to 0.88 kgCO2/kWhth, while in China, the range was 0.63 to 1.47 kgCO2/kWhth, depending on the region and the type of air conditioning system used (Kapshe & Garg, 2017; Wong et al., 2011; Li & Wang, 2018).

Another study published in the Journal of Cleaner Production found that the average carbon emissions per kWh for air conditioning in Europe ranged from 0.14 to 0.64 kgCO2/kWhth, depending on the region and the type of air conditioning system used.

An assessment of the heat pump system designed and implemented in the Cairo University rock laboratory showed that its GHG production was 0.1517 kgCO2eq./kWhth. This value is below the average for India, Singapore, and China and is within the range of European standards. These findings indicate that the heat pump system is environmentally friendly and has the potential to significantly reduce carbon emissions compared to conventional air conditioning systems.

The ground system was simulated using COMSOL software, and the results showed that its effectiveness was 66.72%. This is a promising result that can be improved through further research. Additionally, a life cycle assessment of the case study indicated that the application of geothermal technology for cooling and heating air conditioning significantly reduces carbon emissions, even when considering the negative impact on the environment from construction, drilling, and manufacturing.

A study was also conducted on the application of geothermal technology for cooling and heating air conditioning in one of the new Alamein towers, where the cooling and heating load was high. The study found that the area required for a horizontal ground loop heat pump was large, while the area required for a vertical ground loop heat pump was smaller but required greater drilling depths. The decision on which type of system to use ultimately depends on the project owner and available resources. However, in most cases, the vertical ground loop heat pump is the optimal solution and performs the best in large applications with high loads.

In the case of a villa unit in El-Alamein city, the vertical ground loop heat pump was chosen to achieve the least drilling area. The study found that six boreholes at a depth of 75 meters, covering an area of 98 square meters, representing 46% of the total building area, were required. After studying the feasibility of implementation, it was found that the payback period is 5.6 years if geothermal heat pumps are installed instead of conventional air conditioning. The payback period is 9.17 years if geothermal heat pumps and PV systems are installed instead of conventional air conditioning. The addition of PV systems to the geothermal technology makes it a more environmentally friendly option.

When applying geothermal technology for cooling and heating the acoustic structure, a hybrid system consisting of a water-cooled heat pump and a ground-source heat pump was studied. The study determined the area required for drilling and the required depths, as well as the full construction cost. The results were logical and suitable for implementation.

The prototype (experiment) showed promising results, demonstrating the benefits of using geothermal technology in our daily lives. The experiment showed that the water cooler consumes less energy when cooling the condenser with water from the soil compared to cooling using air. The goal was to cool 1.5 Liters of water from 31°C to 19°C. In the case of cooling with water, it took 46.2 minutes, while in the case of cooling with air, it took 56 minutes. This indicates energy savings, even after considering the energy consumed by the water pump. The soil performance in the box was also monitored in terms of temperature and distribution. The temperatures in the box were found to change and fluctuate over the operation of 24 hours. This is likely due to the hot weather conditions during the measurement period (35 to 40 degrees Celsius) and the relatively small amount of convection heat expelled in the soil, which does not significantly affect its temperature. However, over a longer period of operation, the soil temperature increases, which can affect the performance of the experiment. From COP results for being less than 1 shows that the system is not performing effectively as components used were salvage.

In GSHP designs, the prototype model proposed in the present work in COMSOL provides the temperature distribution surrounding the trench pipes so that the complete heat extraction/injection effects can be visualized. The model studied the effects of trench pipes interaction in a 3D manner. In this study, a 24-hour and a 30-day analysis were performed and as a result, heat accumulation was observed at the corners of the prototype near the inlet pipe (hot) which shows logic results.

Although the simulation results of the geothermal technology system using COMSOL software were promising, there are some limitations to the simulation that should be considered. First, the simulation model was based on assumptions and simplifications of the actual system, which may not accurately reflect the real-world conditions. For example, the simulation did not consider the effects of long-term operation on the system's performance, such as the potential for clogging or corrosion in the pipes, or the accumulation of sediment in the boreholes.

Second, the simulation was conducted with a fixed set of input parameters, such as the ground temperature, the flow rate of the heat transfer fluid, and the heat exchanger's effectiveness. In reality, these parameters may vary depending on several factors, such as the time of day, the season, or changes in the building's heating and cooling demand. Therefore, the simulation results may not represent the actual performance of the system under different operating conditions.

Moreover, the simulation did not account for the potential impact of external factors on the system's performance, such as the effects of nearby buildings or structures on the ground temperature or the potential for underground water flow to affect the performance of the boreholes.

Despite these limitations, the simulation provides useful insights into the potential performance of the geothermal technology system and can help inform the design and implementation of such systems in practice. However, further research is needed to address the limitations of the simulation and to validate the simulation's results with real-world data.

Furthermore, the study did not assess the social and economic impacts of implementing geothermal technology in buildings. Future studies should consider the potential social and economic benefits and drawbacks of using geothermal technology, such as job creation, energy cost savings, and increased energy independence.

Whilst the study assessed the environmental impact of geothermal technology, it did not consider the potential environmental impacts of the geothermal drilling process, such as land disturbance, noise pollution, and the potential for groundwater contamination. Future studies should evaluate the potential environmental impacts of geothermal drilling and explore ways to mitigate any negative effects.

Despite these limitations, the study provides valuable insights into the feasibility and cost-effectiveness of implementing geothermal technology for cooling and heating air conditioning systems in buildings (pilot study). Further research is needed to address the limitations of this study and to explore the potential of geothermal technology for other applications.

Conclusion:

Geothermal energy for cooling and heating applications using GSHPs is a promising renewable and sustainable source of energy which reflects its friendly impact on the environment and cost savings.

Through this study, it's found that the application of the geothermal technology for HVAC application depends entirely on the location of the project and the geological properties of the land that contains this project. It is shown that each project has its terms of ground loop sizing and the energy required to implement it. The two factors that showed the determination of the success of geothermal technology for HVAC application were the cost and its environmental impact. It can be said that the factor of preserving the environment and reducing emissions is already achieved in the case of using geothermal technology for HVAC application because it is sustainable and renewable energy. Therefore, the cost factor and the availability of sustainable resources and capabilities are the main drivers in geothermal cooling systems.

Recommendations:

It is worthy to fulfill the knowledge gap that this study presents, further study is needed to examine borehole separation distances and the potential of hybridization for more feasible systems for extreme heating/cooling cases for performance optimization.

In addition, borehole lengths should also be studied to determine potential benefits of varying borehole lengths in a configuration. Future work in this area also includes study of the use of a thermal storage medium with a GSHP.

It is important to implement a feasible geothermal cooling/heating plant, thermal response test of the project's ground should be made to know the temperature profile of the ground at different depths and based on the results the air conditioning system design will be made.

Future studies should explore the potential for geothermal technology to be used in other applications to further reduce the environmental impact of buildings such as space heating or hot water production.

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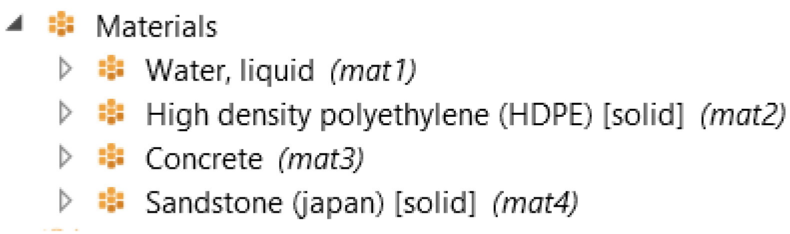
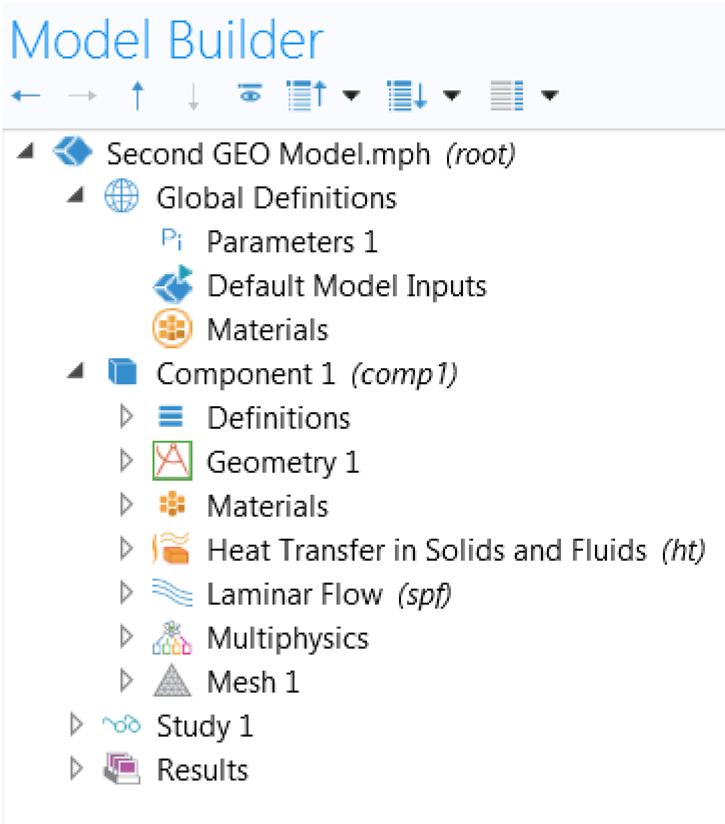
[27] Horizontal Ground Heat Exchanger Design for Ground-Coupled Heat Pumps

Appendix A

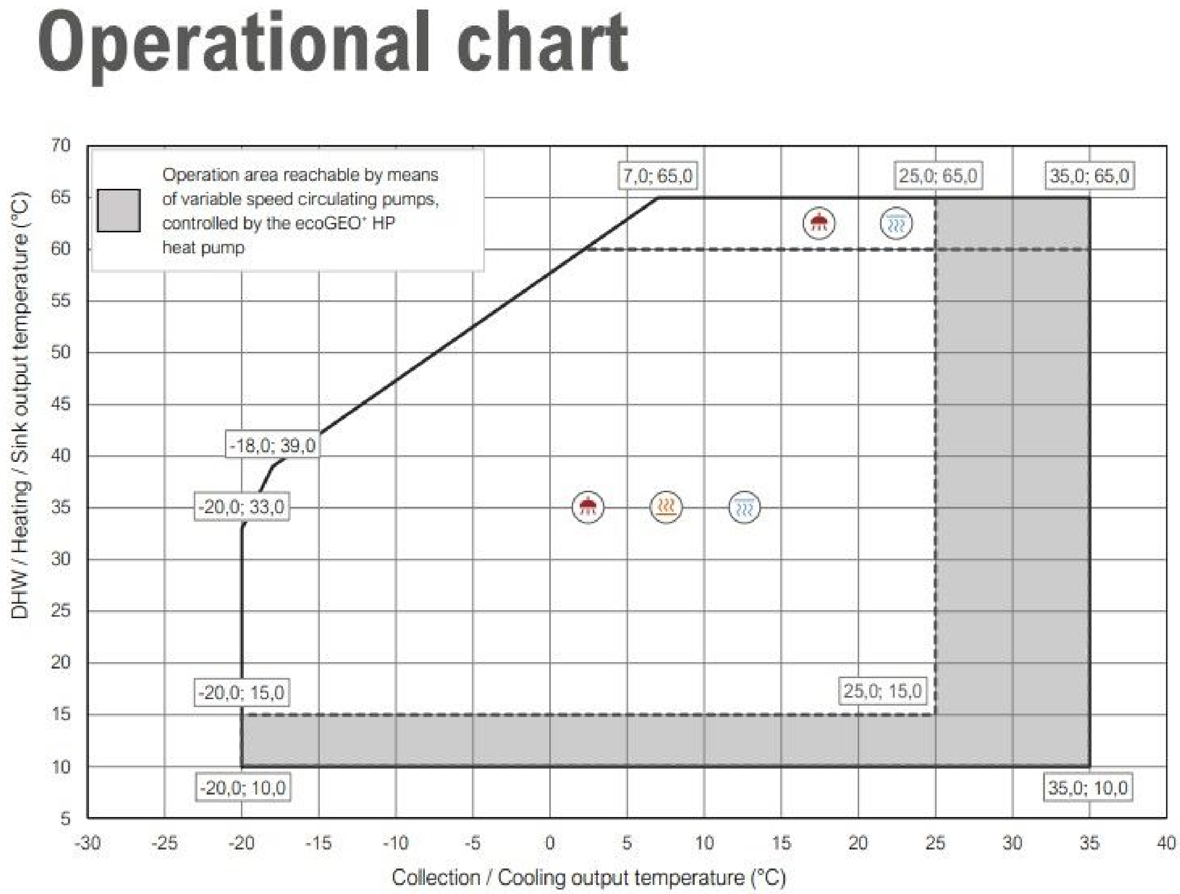
- ***CFD Model Physics and Materials***
  - ***Materials:***



  - ***Model Builder: (Case Study Physics)***



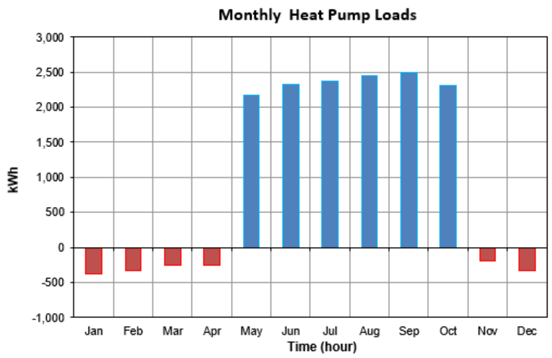
- ***Heat Pump Operational Chart:***



Max. entering fluid temperature and min. entering fluid temperature obtained from this chart

- ***GHX_Design_Toolbox Input data Layout:***
  - ***Cairo University Case Study:***

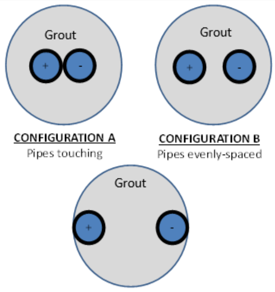
				Building Loads Input								Monthly Heat Pump Loads Corrected for Supplemental Heating				
Supplemental Heat =	0	kW		Hrs/Mo.	Start Hr.	End Hr.	Tot. Heating	Tot. Cooling	Peak Ht	Peak Cl		Tot. Heating	Tot. Cooling	Peak Ht	Peak Cl	
Peak Bldg. Heat =	2	kW					(kWh)	(kWh)	(kW)	(kW)		(kWh)	(kWh)	(kW)	(kW)	
			Jan	744	1	744	381	0	2	0		381.0	0.0	1.9	0.0	
Total ht	1,764	kWh	Feb	672	745	1416	339	0	2	0		339.0	0.0	1.7	0.0	
Total cl	14,110	kWh	Mar	744	1417	2160	258	0	1	0		258.0	0.0	1.4	0.0	
			Apr	720	2161	2880	255	0	1	0		255.0	0.0	1.3	0.0	
Peak Ht	2	kW	May	744	2881	3624	0	2,166	0	5		0.0	2,165.8	0.0	5.1	
Pk Cool	6	kW	Jun	720	3625	4344	0	2,321	0	5		0.0	2,320.8	0.0	5.4	
			Jul	744	4345	5088	0	2,368	0	6		0.0	2,368.5	0.0	5.5	
Month Load Factor (Htg.)	0.270	--	Aug	744	5089	5832	0	2,452	0	6		0.0	2,452.3	0.0	5.6	
Month Load Factor (Clg.)	0.000	--	Sep	720	5833	6552	0	2,494	0	6		0.0	2,494.0	0.0	5.8	
			Oct	744	6553	7296	0	2,309	0	5		0.0	2,308.6	0.0	5.4	
Annual Load Factor (Htg.)	0.106	hr	Nov	720	7297	8016	198	0	1	0		198.0	0.0	1.4	0.0	
Annual Load Factor (Clg.)	0.278	hr	Dec	744	8017	8760	333	0	2	0		333.0	0.0	1.8	0.0	
				8760	8761		1,764	14,110	1.9	5.8		1,764.0	14,110.0	1.9	5.8	
Cool/Heat Ratio	8.00	--														
Heat/Cool Ratio	0.13	--														



Click Here to Return to Main Input Sheet

Input Data in Yellow Fields — SI Units

Pipe Details:		
Nominal Diameter:	DN32	mm
SDR:	SDR11	--
Pipe k:	0.4	W/(m-K)
Pipe OD:	45.000	mm
Pipe ID:	37.000	mm
u-tube center to borehole center Spacing:	22.50	mm
Borehole Details:		
Borehole Dia.	152.00	mm
Borehole Depth	35	m
Fluid Details:		
Heat Transfer Fluid:	Water	
Flow Rate per Borehole:	5.25	Lpm
Design Temperature of Fluid:	24.5	°C
Freeze Point:	-7.3	°C
Density:	1025.80	kg/m^3
Specific Heat:	3929.00	J/(kg-°C)
Thermal Conductivity:	0.61	W/(m-K)
Viscosity:	4.05E-03	kg/(m-s)
Grout Details:		
Thermal Conductivity:	2.50	W/(m-K)
Ground Details:		
Thermal Conductivity:	2.50	W/(m-K)



CONFIGURATION A
Pipes touching

CONFIGURATION B
Pipes evenly-spaced

CONFIGURATION C
Pipes against borehole wall

Select Pipe Configuration: Configuration A

R'_b = 0.1659 m-°C/W

Calculations

Mass Flow Rate =	8.98E-02	kg/s
Re =	763	-
Pr =	26.24	-
friction factor (f) =	0.07703	-
Nu =	4.360	-
h =	71.5	W/(m^2-°C)
$R'_{p\_convective\_component}$ =	0.12040	m-°C-/W
$R'_{p\_conductive\_component}$ =	0.07788	m-°C-/W
R'_p =	0.19829	m-°C-/W
β =	3.115	-
σ =	0.000	-
R'_1 =	0.50360	
R'_2 =	0.15926	
$R'_{1,2}$ =	2.4040	
R'_{eff1} =	0.1659	
R'_{eff2} =	0.1659	

START | Vertical GHX | Single u-tube | Borehole Field Configuration | +

Job Description: **PROJECT NAME**

(ENTER DATA IN COLORED FIELDS ONLY)

Select Units: **SI**

Ground Parameters			EXPLANATION
$T_{undisturbed}$ =	24.5	°C	Average underground earth temperature
bore diameter =	152.00	mm	Borehole diameter
k =	2.5	W/(m-°C)	Average ground thermal conductivity
Vρcp ground =	2800	kJ/(°C-m^3)	Average ground volumetric heat capacity
Bore Thermal Resistance =	0.166	m-°C/W	Calculate single u-tube
Specific heat of fluid =	3.93	kJ/(kg-°C)	The heat transfer fluid in the borehole (4.19 kJ/(kg-°C) for water)

Click Here to Size Vertical GHX

Select Borehole Field Configuration: Uniform Grid / User-Defined

Currently Input Configuration: Uniform Grid of 2 x 2 boreholes, 7-m bore-to-bore spacing.

Heat Pump			
Max EWT =	35	°C	Max design heat pump entering fluid temp.
Min EWT =	-25	°C	Min design heat pump entering fluid temp.
COPh =	6.3		COP at design conditions
COPc =	5.5		COP at design conditions
Total Fluid Flow Rate =	21	L/min.	Design flow rate (usually 2.4-3.4 Lpm/kW)

Loads Information	Htg.	Clg.	
Peak Loads (kW)	2	6	Peak hour loads
Peak Load Duration (hours)	4	4	Expected duration of peak load on the peak day
Monthly Load Factor	0.43	0.31	Effectively, the fraction of the peak month that the system runs
Annual Load Factor	0.21	0.11	Effectively, the fraction of the year that the system runs
Design Life	20	yrs	Design life (to size for long-term effects)

START | Vertical GHX | Single u-tube | Borehole Field Configuration

- ***El-Alamein Case study:***

  1) El-Alamein Gate Towers:

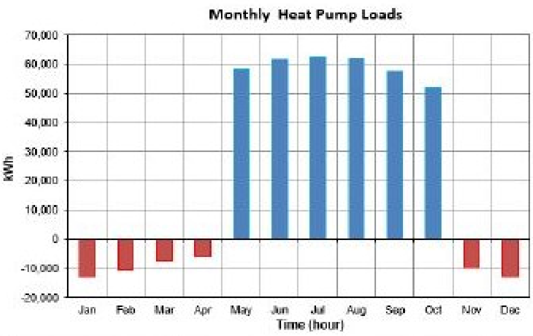
Supplemental Heat =	0	kW
Peak Bldg. Heat =	36	kW
Total ht	59,526	kWh
Total cl	353,904	kWh
Peak Ht	36	kW
Pk Cool	108	kW
Month Load Factor (Htg.)	0.478	--
Month Load Factor (Clg.)	0.000	--
Annual Load Factor (Htg.)	0.187	hr
Annual Load Factor (Clg.)	0.375	hr
Cool/Heat Ratio	5.95	--
Heat/Cool Ratio	0.17	--

Building Loads Input

	Hrs/Mo.	Start Hr.	End Hr.	Tot. Heating (kWh)	Tot. Cooling (kWh)	Peak Ht (kW)	Peak Cl (kW)
Jan	744	1	744	12,948	0	36	0
Feb	672	745	1416	10,524	0	32	0
Mar	744	1417	2160	7,281	0	21	0
Apr	720	2161	2880	5,901	0	19	0
May	744	2881	3624	0	58,182	0	101
Jun	720	3625	4344	0	61,881	0	108
Jul	744	4345	5088	0	62,319	0	105
Aug	744	5089	5832	0	61,962	0	108
Sep	720	5833	6552	0	57,648	0	104
Oct	744	6553	7296	0	51,912	0	97
Nov	720	7297	8016	9,759	0	28	0
Dec	744	8017	8760	13,113	0	36	0
	8760	8761		59,526	353,904	36.4	107.8

Monthly Heat Pump Loads Corrected for Supplemental Heating

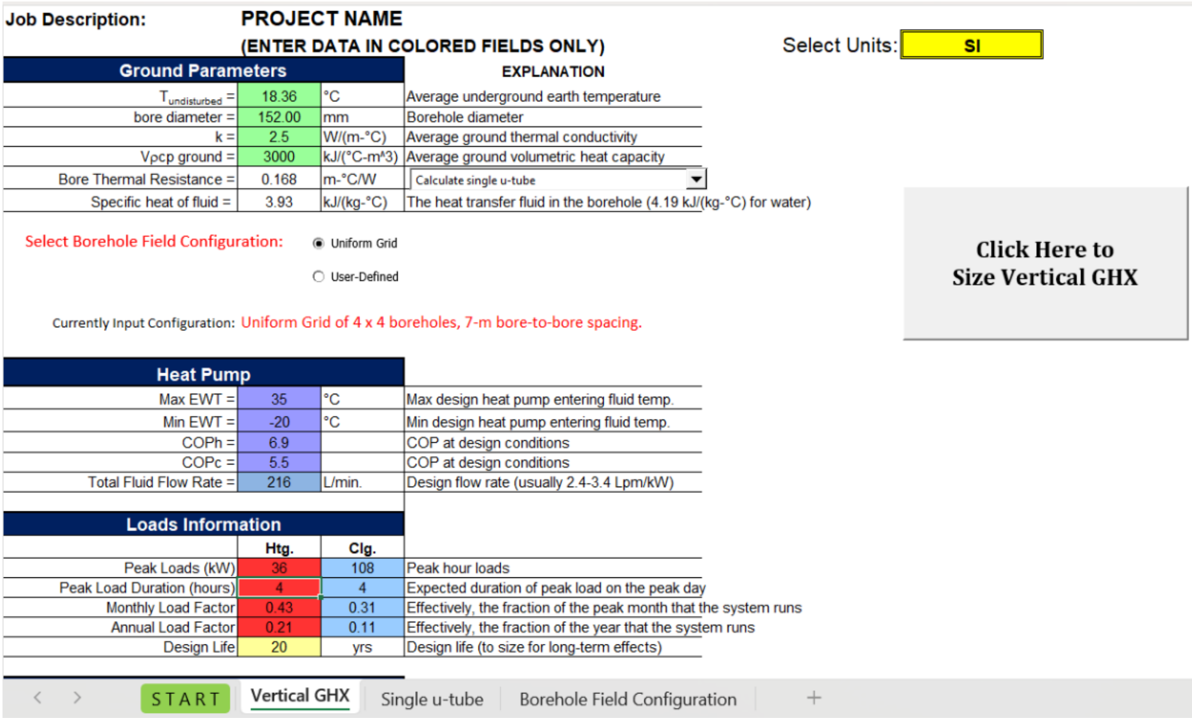
Tot. Heating (kWh)	Tot. Cooling (kWh)	Peak Ht (kW)	Peak Cl (kW)
12,948.0	0.0	35.4	0.0
10,524.0	0.0	31.7	0.0
7,281.0	0.0	21.4	0.0
5,901.0	0.0	19.0	0.0
0.0	58,182.0	0.0	100.5
0.0	61,881.0	0.0	107.8
0.0	62,319.0	0.0	104.8
0.0	61,962.0	0.0	107.6
0.0	57,648.0	0.0	103.5
0.0	51,912.0	0.0	96.5
9,759.0	0.0	27.9	0.0
13,113.0	0.0	35.9	0.0
59,526.0	353,904.0	35.4	107.8



Click Here to Return to Main Input Sheet

START | Vertical GHX | Borehole Field Configuration | Heat Pump Input | Monthly Loads | Output

I. Vertical



Job Description: **PROJECT NAME**

(ENTER DATA IN COLORED FIELDS ONLY)

Select Units: **SI**

Ground Parameters			EXPLANATION
$T_{undisturbed}$ =	18.36	°C	Average underground earth temperature
bore diameter =	152.00	mm	Borehole diameter
k =	2.5	W/(m-°C)	Average ground thermal conductivity
Vρcp ground =	3000	kJ/(°C-m^3)	Average ground volumetric heat capacity
Bore Thermal Resistance =	0.168	m-°C/W	Calculate single u-tube
Specific heat of fluid =	3.93	kJ/(kg-°C)	The heat transfer fluid in the borehole (4.19 kJ/(kg-°C) for water)

Select Borehole Field Configuration: ◉ Uniform Grid ○ User-Defined

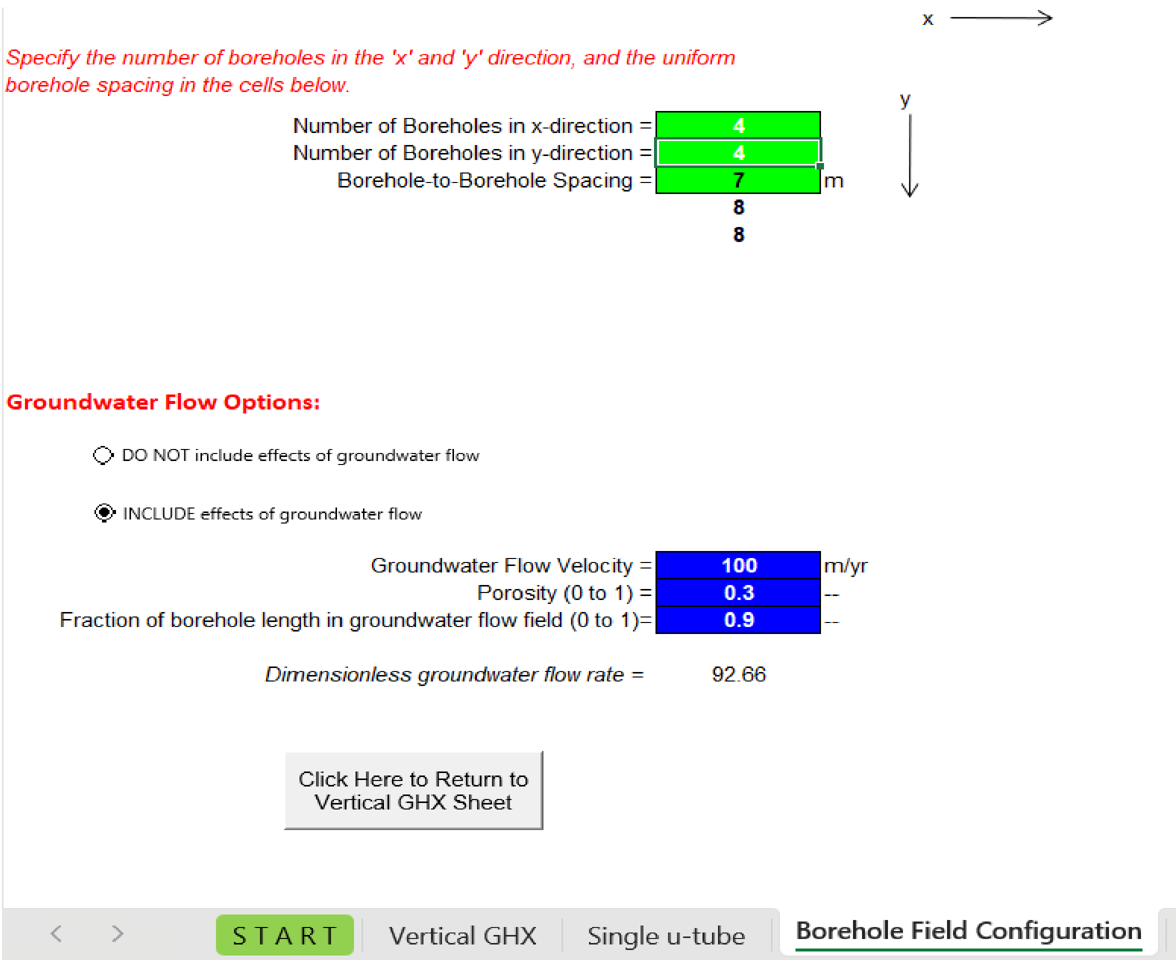
Click Here to Size Vertical GHX

Currently Input Configuration: Uniform Grid of 4 x 4 boreholes, 7-m bore-to-bore spacing.

Heat Pump			
Max EWT =	35	°C	Max design heat pump entering fluid temp.
Min EWT =	-20	°C	Min design heat pump entering fluid temp.
COPh =	6.9		COP at design conditions
COPc =	5.5		COP at design conditions
Total Fluid Flow Rate =	216	L/min.	Design flow rate (usually 2.4-3.4 Lpm/kW)

Loads Information	Htg.	Clg.	
Peak Loads (kW)	36	108	Peak hour loads
Peak Load Duration (hours)	4	4	Expected duration of peak load on the peak day
Monthly Load Factor	0.43	0.31	Effectively, the fraction of the peak month that the system runs
Annual Load Factor	0.21	0.11	Effectively, the fraction of the year that the system runs
Design Life	20	yrs	Design life (to size for long-term effects)

START | Vertical GHX | Single u-tube | Borehole Field Configuration



x

y

Specify the number of boreholes in the 'x' and 'y' direction, and the uniform borehole spacing in the cells below.

Number of Boreholes in x-direction =	4	
Number of Boreholes in y-direction =	4	
Borehole-to-Borehole Spacing =	7	m
	8	
	8	

Groundwater Flow Options:

○ DO NOT include effects of groundwater flow

◉ INCLUDE effects of groundwater flow

Groundwater Flow Velocity =	100	m/yr
Porosity (0 to 1) =	0.3	--
Fraction of borehole length in groundwater flow field (0 to 1)=	0.9	--

Dimensionless groundwater flow rate = 92.66

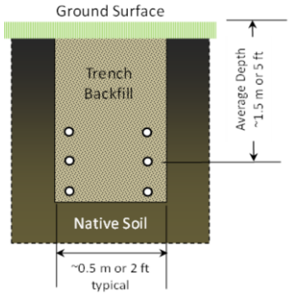
Click Here to Return to Vertical GHX Sheet

START | Vertical GHX | Single u-tube | Borehole Field Configuration

II. Horizontal

Input Data in Yellow Fields

	SI Units	
Pipe Details:		
Nominal Diameter:	DN32	mm
SDR:	SDR11	--
Pipe k:	0.4	W/(m-K)
Pipe OD:	42.164	mm
Pipe ID:	34.498	mm
Pipe center-to-center spacing:	450.00	mm
Trench Details:		
Width:	5000.00	mm
Fluid Details:		
Heat Transfer Fluid:	Water	
Flow Rate per Circuit:	10.80	Lpm
Design Temperature of Fluid:	18.4	°C
Freeze Point:	0	°C
Density:	998.53	kg/m^3
Specific Heat:	4188.06	J/(kg-°C)
Thermal Conductivity:	0.60	W/(m-K)
Viscosity:	1.04E-03	kg/(m-s)
Trench Backfill:		
Thermal Conductivity:	2.50	W/(m-K)



R'_{trench} = 0.1714 m-°C/W

Calculations		
Mass Flow Rate =	1.80E-01	kg/s
Re =	6,366	-
Pr =	7.33	-
friction factor (f) =	0.03588	-
Nu =	62.292	-
h =	1075.7	W/(m^2-°C)
$R'_{p\_convective\_component}$ =	0.00858	m-°C-/W
$R'_{p\_conductive\_component}$ =	0.07984	m-°C-/W
R'_p =	0.08842	m-°C-/W
eccentricity parameter (b) =	0.045	-
R'_{sf} =	0.1633	m-°C-/W
R'_a =	0.1128	m-°C-/W

START | HorizontalGHX | Six-Pipe Trench

File Home Insert Page Layout Formulas Data Review View Automate Help

B30

Job Description: **Project Description**

(ENTER DATA IN COLORED FIELDS ONLY)

Select Units: SI

Ground Parameters			EXPLANATION
Trench width or Bore diameter =	5000.00	mm	Width of trench or diameter of horizontal borehole
k =	2.5	W/(m-°C)	Average ground thermal conductivity
ρcp ground =	3000	kJ/(°C-m^3)	Average ground volumetric heat capacity
Trench Thermal Resistance =	0.171	m-°C/W	Calculate six-pipe trench resistance
Mean Earth temperature =	18.36	°C	
Surface Amplitude =	10	°C	Temperature amplitude at ground surface (~10 °C)
to =	90	days	Number of days to minimum ground surface temperature
Earth Temperature at 35 days =	17.8	°C	
Earth Temperature at 215 days =	18.6	°C	
Specific heat of fluid =	4.19	kJ/(kg-°C)	The heat transfer fluid in the borehole (4.19 kJ/(kg-°C) for water)
# Trenches or Bores =	20		The number of horizontal trenches or bores
Average Depth =	1.5	m	Average depth of heat exchange pipes
Trench/Bore Spacing =	3.0	m	<- Recommended Max value is 2x the pipe depth

Heat Pump			
Max EWT =	35	°C	Max design heat pump entering fluid temp.
Min EWT =	-20	°C	Min design heat pump entering fluid temp.
COPh =	6.9		COP (heating) at design conditions
COPc =	5.5		COP (cooling) at design conditions
Flow Rate =	216.0	L/min.	Design flow rate (usually 2.4-3.4 Lpm/kW)

Loads Information	Htg.	Clg.	
Peak Loads (kW)	36	108	Peak hour loads
Peak Load Duration (hours)	4	4	Expected duration of peak load on the peak day
Monthly Load Factor	0.43	0.31	Effectively, the fraction of the peak month that the system runs
Annual Load Factor	0.21	0.11	Effectively, the fraction of the year that the system runs
Design Life	20	yrs	Design life (to size for long-term buildup)

Click Here to Size Horizontal GHX

Microsoft Excel

RESULTS:

Trench Configuration: 20 @ 3.0 m spacing
Calculated Length for Each Trench: 78.9 m

Total Length Requirement for Trenches:
1579 m, driven by the cooling loads (-80 m are driven by the heating loads)

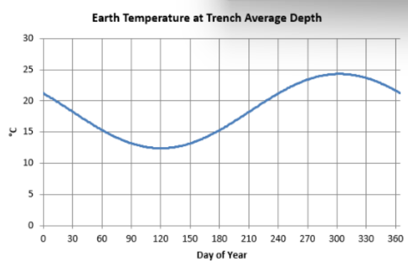
Relative Impact on GHX Design Length Due to the Ground Thermal Resistances Related to:

Trench Resistance:	67.3%
Daily Pulse:	.5%
Monthly Pulse:	3.4%
Annual Pulse:	28.9%

Residual: 8.429E-06

Calculation Time: 6.797E-01 seconds

OK



START | HorizontalGHX | Six-Pipe Trench

2) El-Alamein Villa:

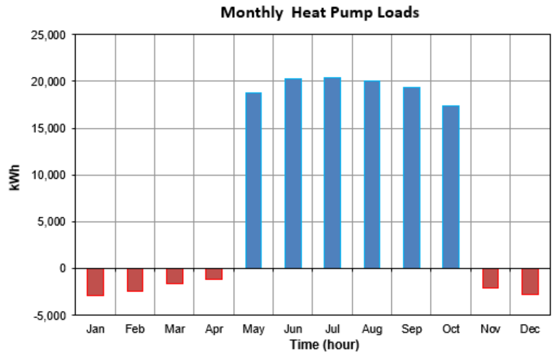
Supplemental Heat =	0	kW
Peak Bldg. Heat =	17	kW
Total ht	12,801	kWh
Total cl	116,301	kWh
Peak Ht	17	kW
Pk Cool	52	kW
Month Load Factor (Htg.)	0.226	--
Month Load Factor (Clg.)	0.000	--
Annual Load Factor (Htg.)	0.085	hr
Annual Load Factor (Clg.)	0.255	hr
Cool/Heat Ratio	9.09	--
Heat/Cool Ratio	0.11	--

Building Loads Input

	Hrs/Mo.	Start Hr.	End Hr.	Tot. Heating (kWh)	Tot. Cooling (kWh)	Peak Ht (kW)	Peak Cl (kW)
Jan	744	1	744	2,880	0	17	0
Feb	672	745	1416	2,376	0	15	0
Mar	744	1417	2160	1,608	0	10	0
Apr	720	2161	2880	1,185	0	8	0
May	744	2881	3624	0	18,765	0	48
Jun	720	3625	4344	0	20,328	0	51
Jul	744	4345	5088	0	20,418	0	51
Aug	744	5089	5832	0	20,073	0	52
Sep	720	5833	6552	0	19,314	0	51
Oct	744	6553	7296	0	17,403	0	45
Nov	720	7297	8016	2,031	0	12	0
Dec	744	8017	8760	2,721	0	17	0
	8760	8761		12,801	116,301	17.1	52.1

Monthly Heat Pump Loads Corrected for Supplemental Heating

Tot. Heating (kWh)	Tot. Cooling (kWh)	Peak Ht (kW)	Peak Cl (kW)
2,880.0	0.0	17.1	0.0
2,376.0	0.0	15.3	0.0
1,608.0	0.0	10.1	0.0
1,185.0	0.0	8.4	0.0
0.0	18,765.0	0.0	47.9
0.0	20,328.0	0.0	51.1
0.0	20,418.0	0.0	51.4
0.0	20,073.0	0.0	52.1
0.0	19,314.0	0.0	50.6
0.0	17,403.0	0.0	45.3
2,031.0	0.0	12.3	0.0
2,721.0	0.0	16.5	0.0
12,801.0	116,301.0	17.1	52.1

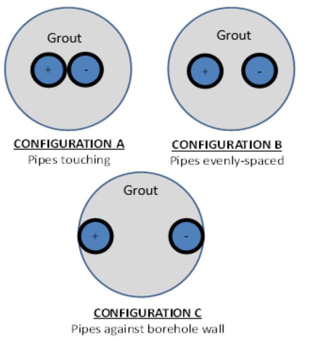


Click Here to Return to Main Input Sheet

START | Vertical GHX | Single u-tube | Borehole Field Configuration | Heat Pump Input | Monthly Loads | Output

Input Data in Yellow Fields — **SI Units**

Pipe Details:		
Nominal Diameter:	DN32	mm
SDR:	SDR11	--
Pipe k:	0.42	W/(m-K)
Pipe OD:	45.000	mm
Pipe ID:	37.000	mm
u-tube center to borehole center Spacing:	53.50	mm
Borehole Details:		
Borehole Dia.	152.00	mm
Borehole Depth	100	m
Fluid Details:		
Heat Transfer Fluid:	Water	
Flow Rate per Borehole:	1000.00	Lpm
Design Temperature of Fluid:	18.4	°C
Freeze Point:	-7.3	°C
Density:	1025.80	kg/m^3
Specific Heat:	3929.00	J/(kg-°C)
Thermal Conductivity:	0.60	W/(m-K)
Viscosity:	4.05E-03	kg/(m-s)
Grout Details:		
Thermal Conductivity:	2.50	W/(m-K)
Ground Details:		
Thermal Conductivity:	2.50	W/(m-K)



Select Pipe Configuration: Configuration C

R'_b = 0.0652 m-°C/W

Calculations

Mass Flow Rate =	1.71E+01	kg/s
Re =	145,266	-
Pr =	26.71	-
friction factor (f) =	0.01665	-
Nu =	1442.635	-
h =	23225.1	W/(m^2-°C)
$R'_{p\_convective\_component}$ =	0.00037	m-°C-/W
$R'_{p\_conductive\_component}$ =	0.07418	m-°C-/W
R'_p =	0.07455	m-°C-/W
β =	1.171	-
σ =	0.000	-
R'_1 =	0.34807	
R'_2 =	0.06524	
$R'_{1,2}$ =	-1.0427	
R'_{eff1} =	0.0652	
R'_{eff2} =	0.0652	

START | Single u-tube

Job Description: **PROJECT NAME**

(ENTER DATA IN COLORED FIELDS ONLY)

Select Units: **SI**

Ground Parameters			**EXPLANATION**
$T_{undisturbed}$ =	18.36	°C	Average underground earth temperature
bore diameter =	152.00	mm	Borehole diameter
k =	2.5	W/(m-°C)	Average ground thermal conductivity
Vρcp ground =	2800	kJ/(°C-m^3)	Average ground volumetric heat capacity
Bore Thermal Resistance =	0.073	m-°C/W	Calculate single u-tube
Specific heat of fluid =	3.93	kJ/(kg-°C)	The heat transfer fluid in the borehole (4.19 kJ/(kg-°C) for water)

Click Here to Size Vertical GHX

Select Borehole Field Configuration:

- ◉ Uniform Grid
- ○ User-Defined

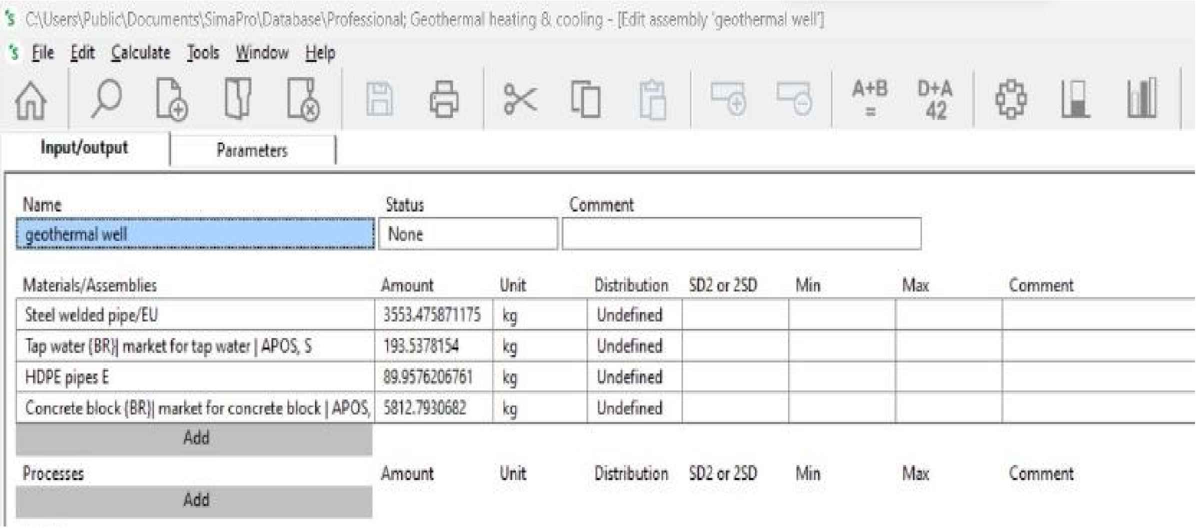
Currently Input Configuration: Uniform Grid of 2 x 3 boreholes, 7-m bore-to-bore spacing.

Heat Pump			
Max EWT =	35	°C	Max design heat pump entering fluid temp.
Min EWT =	-20	°C	Min design heat pump entering fluid temp.
COPh =	6.8		COP at design conditions
COPc =	5.5		COP at design conditions
Total Fluid Flow Rate =	125	L/min.	Design flow rate (usually 2.4-3.4 Lpm/kW)

Loads Information			
	Htg.	**Clg.**	
Peak Loads (kW)	17	52	Peak hour loads
Peak Load Duration (hours)	4	4	Expected duration of peak load on the peak day
Monthly Load Factor	0.23	0.00	Effectively, the fraction of the peak month that the system runs
Annual Load Factor	0.09	0.25	Effectively, the fraction of the year that the system runs
Design Life	25	yrs	Design life (to size for long-term effects)

START | Vertical GHX | Single u-tube | Borehole Field Configuration | +

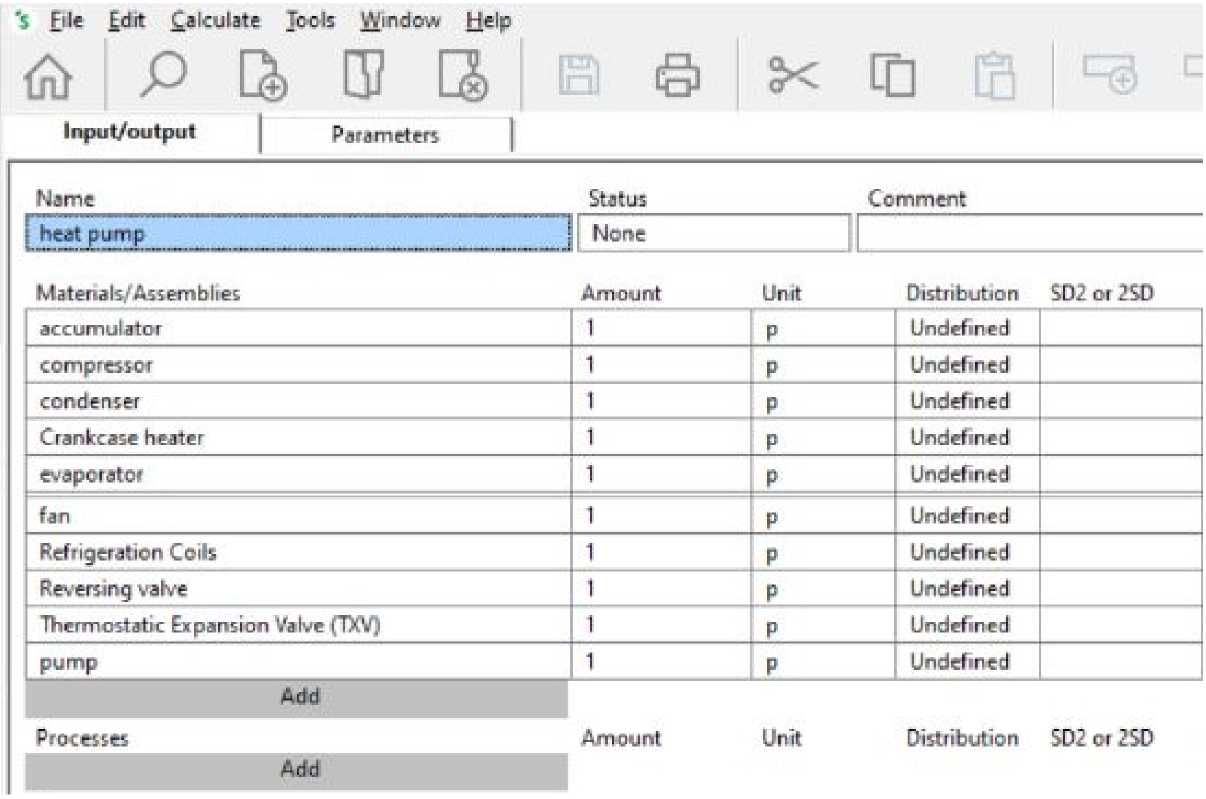
- ***Screenshots of Input Data on Simapro:***



Materials/Assemblies	Amount	Unit	Distribution	SD2 or 2SD	Min	Max	Comment
Steel welded pipe/EU	3553.475871175	kg	Undefined				
Tap water {BR}| market for tap water | APOS, S	193.5378154	kg	Undefined				
HDPE pipes E	89.9576206761	kg	Undefined				
Concrete block {BR}| market for concrete block | APOS,	5812.7930682	kg	Undefined				
Add							

Processes	Amount	Unit	Distribution	SD2 or 2SD	Min	Max	Comment
Add							

Figure 101: Geothermal Well Material Input Data using Simapro



Materials/Assemblies	Amount	Unit	Distribution	SD2 or 2SD
accumulator	1	p	Undefined	
compressor	1	p	Undefined	
condenser	1	p	Undefined	
Crankcase heater	1	p	Undefined	
evaporator	1	p	Undefined	
fan	1	p	Undefined	
Refrigeration Coils	1	p	Undefined	
Reversing valve	1	p	Undefined	
Thermostatic Expansion Valve (TXV)	1	p	Undefined	
pump	1	p	Undefined	
Add				

Processes	Amount	Unit	Distribution	SD2 or 2SD
Add				

Figure 100: GSHP Components for LCA

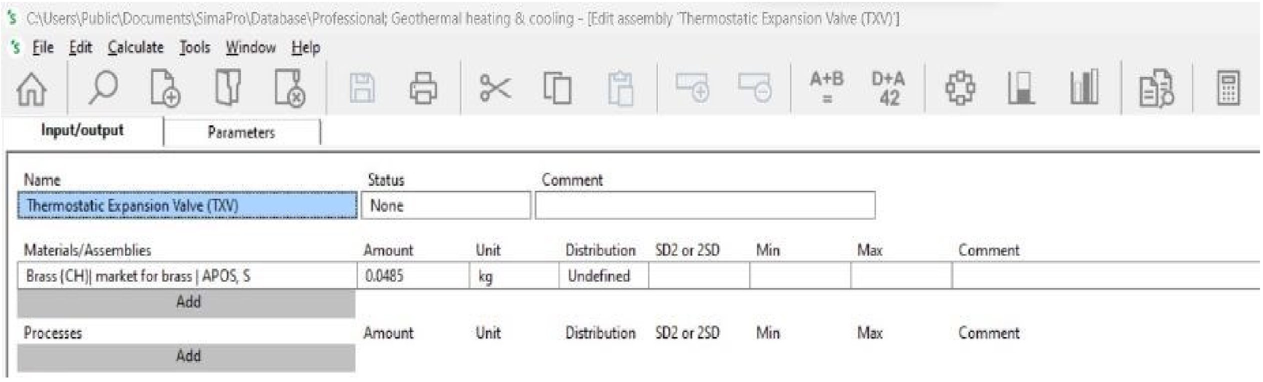


Figure 105: Thermostatic Expansion Valve (TXV) Material Input Data

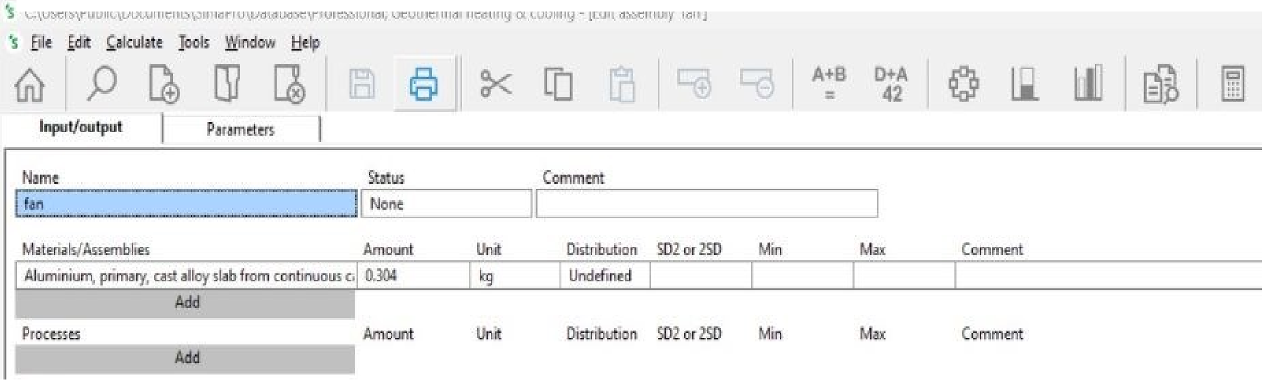


Figure 104: Fan Material Input Data

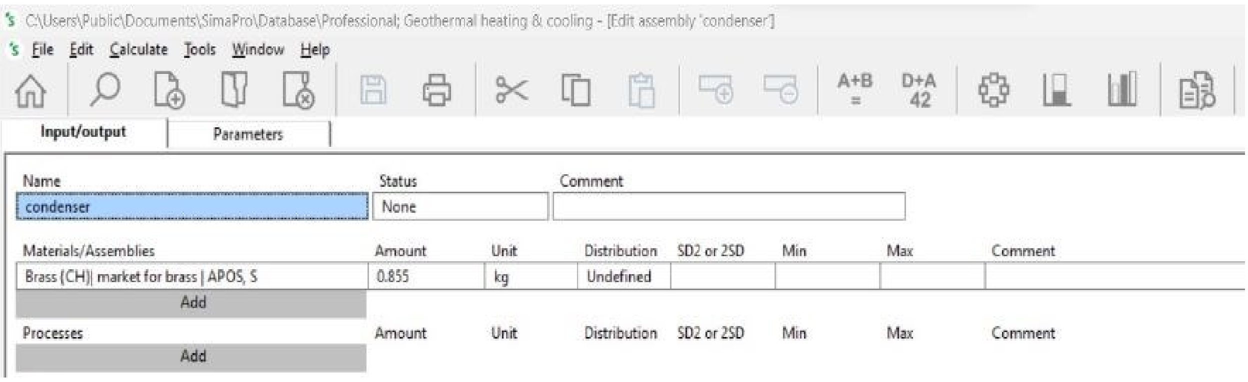


Figure 103: Condenser Material Input Data

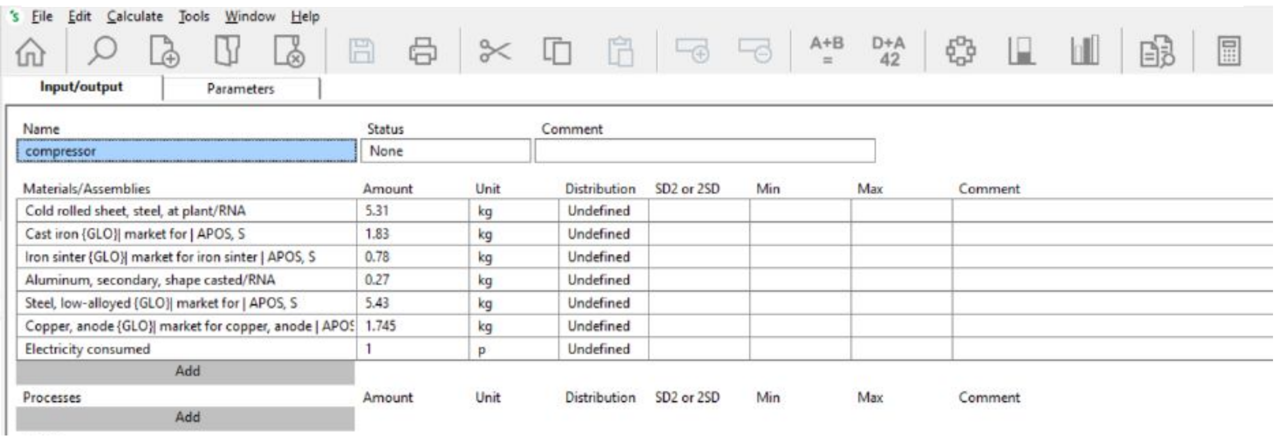


Figure 102: Compressor Material Input Data

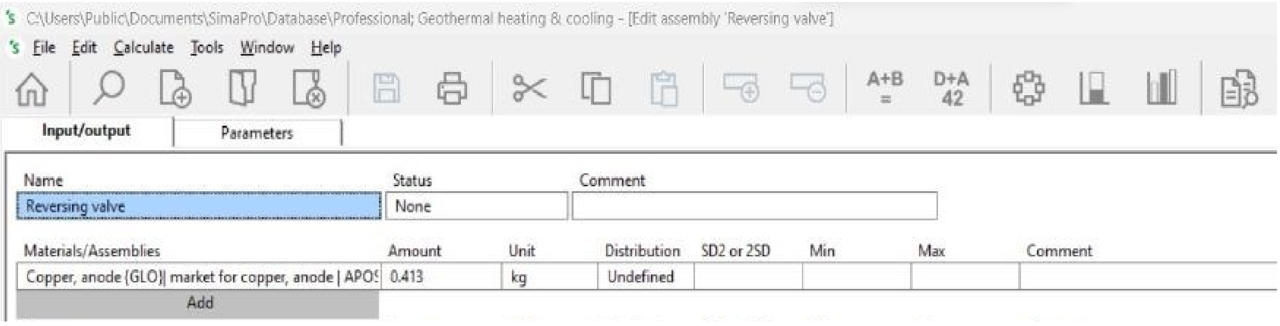


Figure 110: Reversing Valve Material Input Data

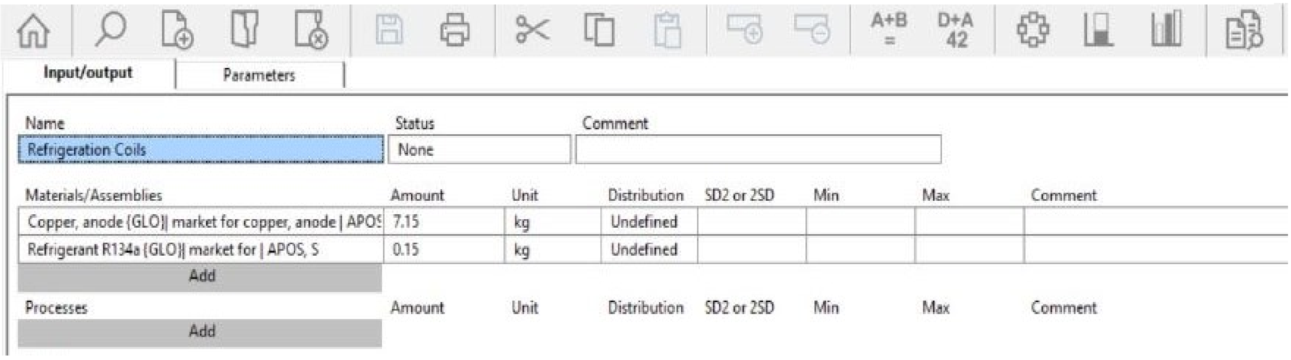


Figure 109: Refrigeration Coils Material Input Data

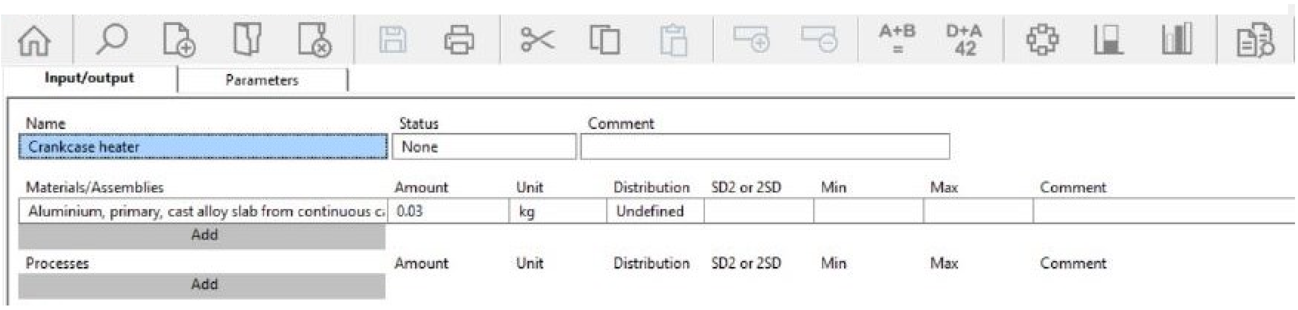


Figure 108: Crankcase heater Material Input Data

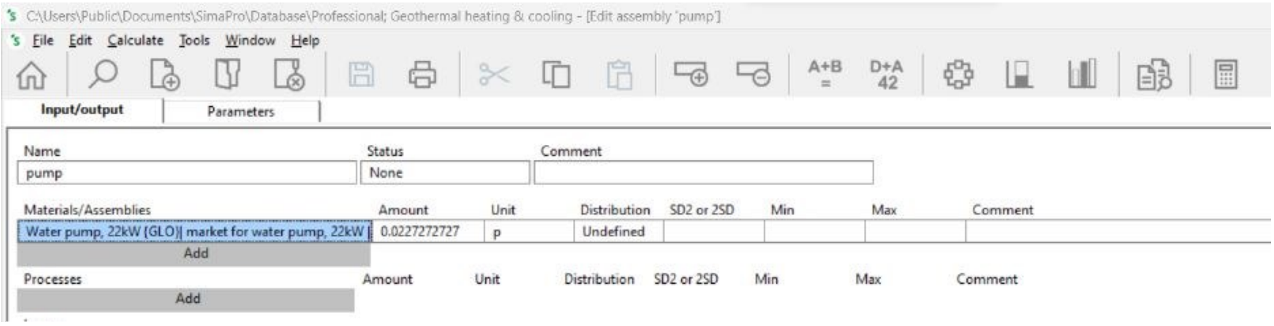


Figure 107: Pump Material Input data

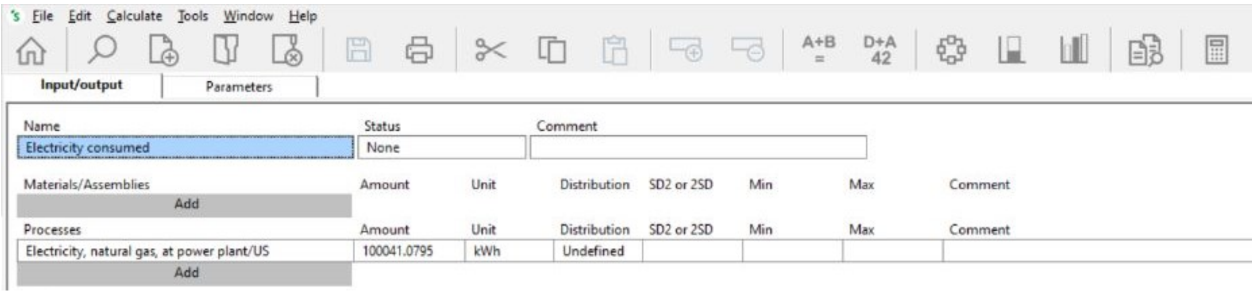
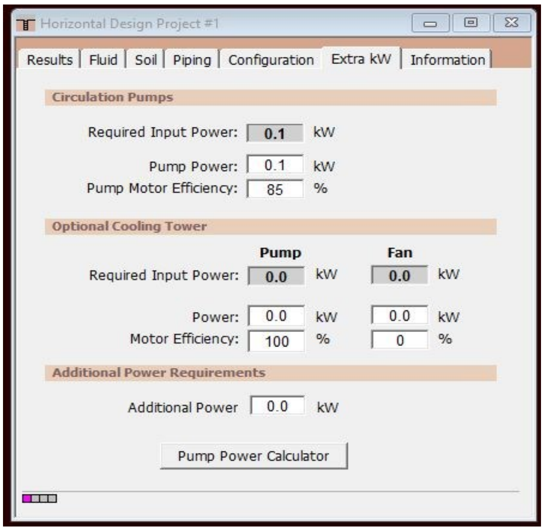
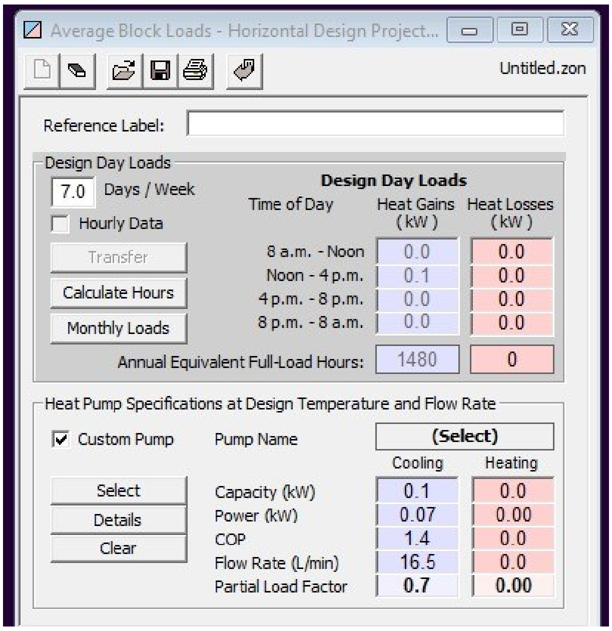
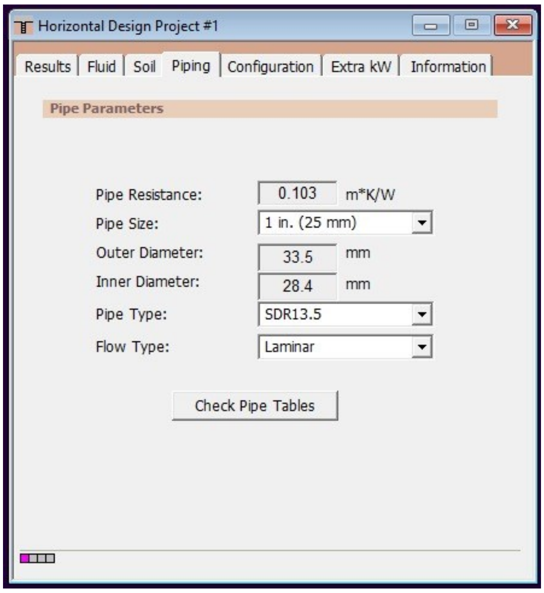


Figure 106: Electrical Consumption Input Data for LCA

- ***<u>Screenshots of Input Data on GLD Software:</u>***

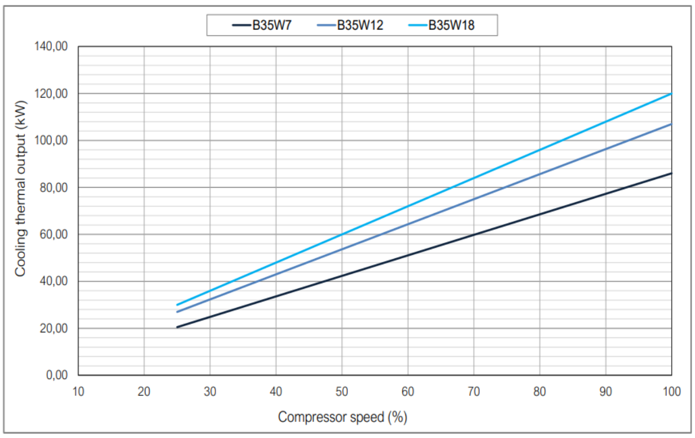


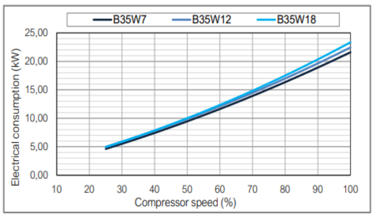


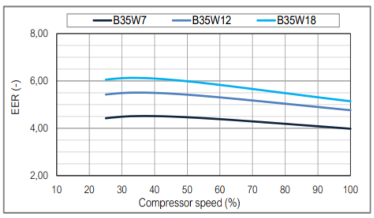


- ***<u>Cost Assessment: (Electrical Consumption)</u>***

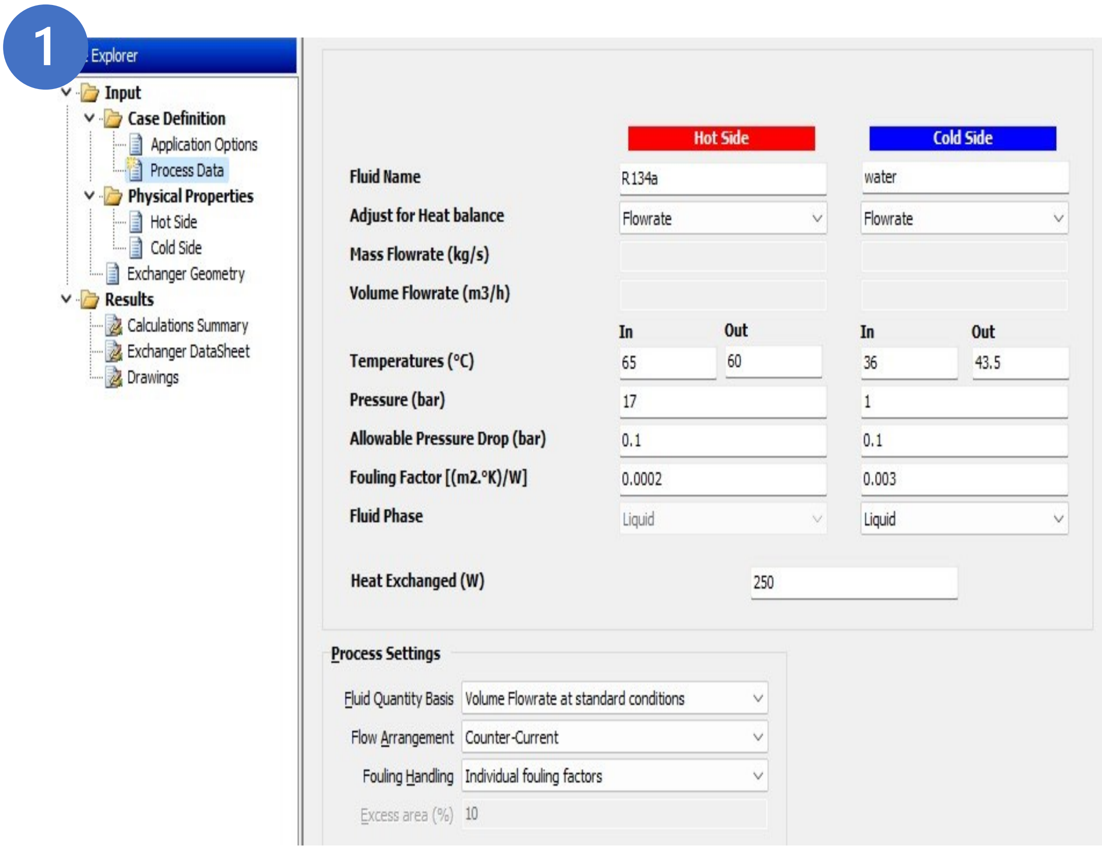
By using the performance curves in the Eco Forest catalog, a relation was obtained between the Cooling load and electrical consumption (function in compressor speed).

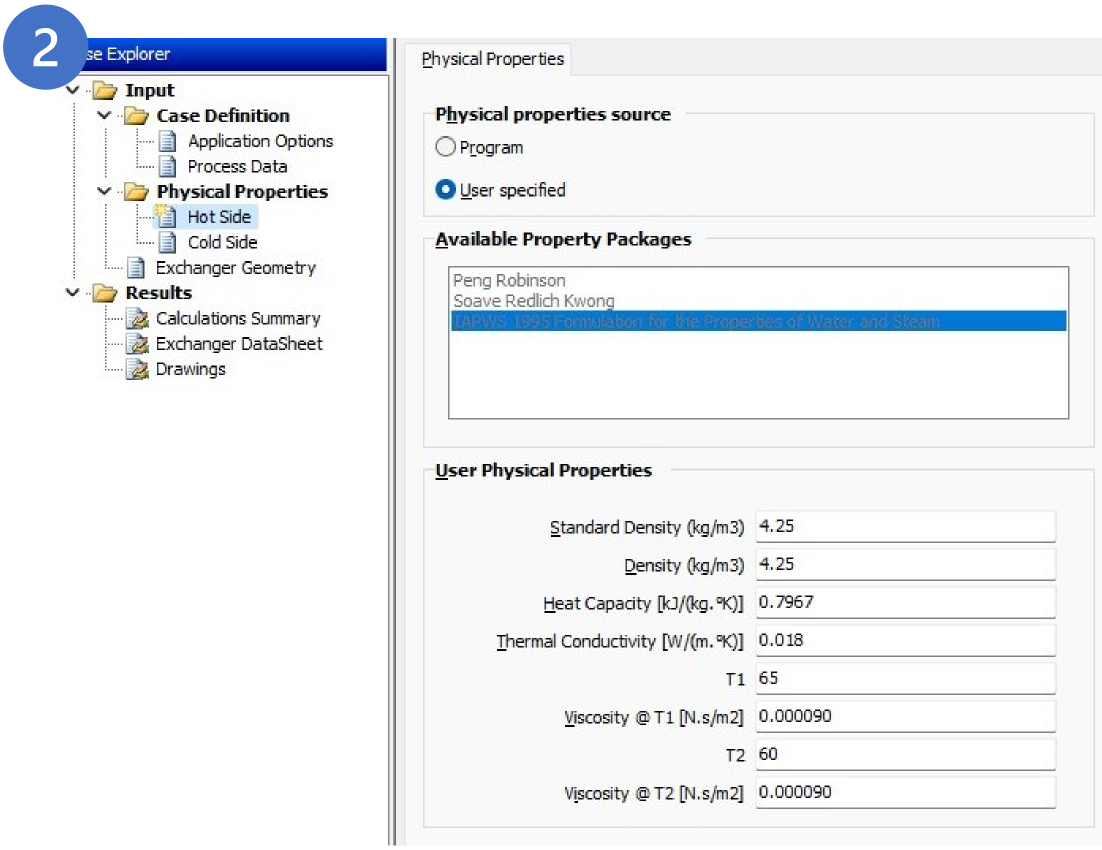


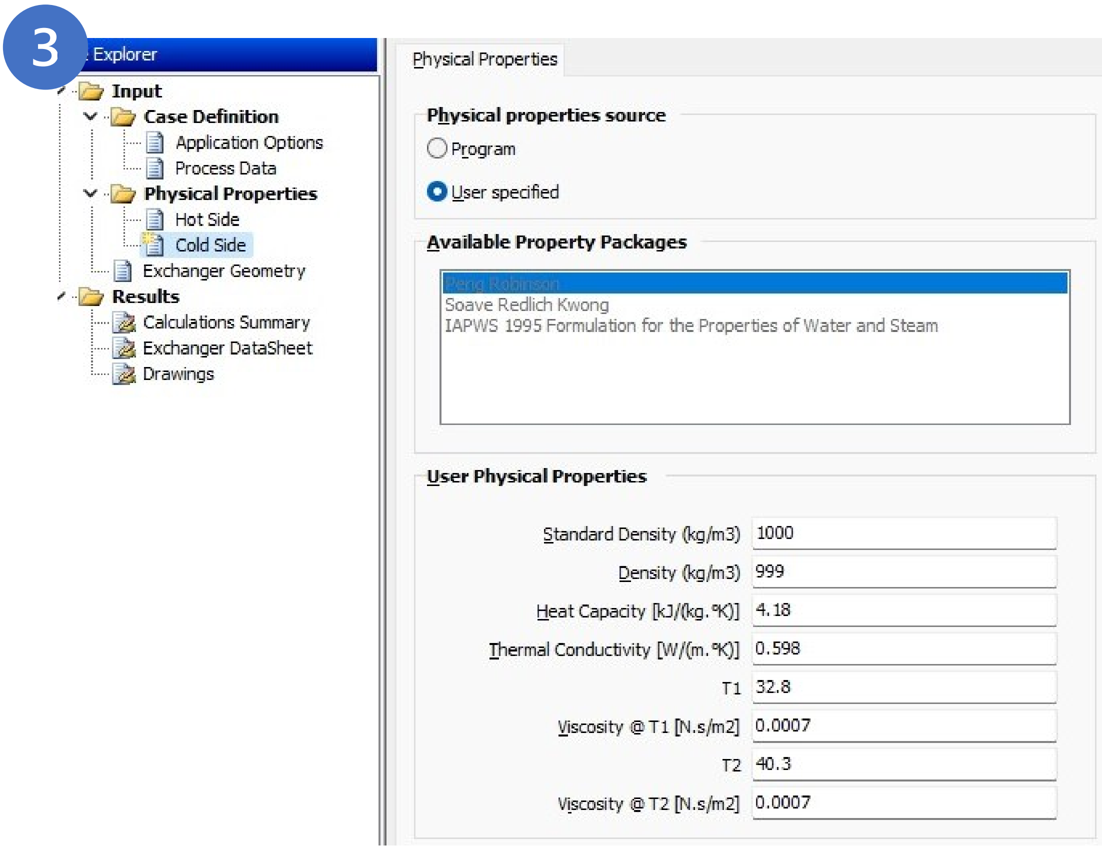


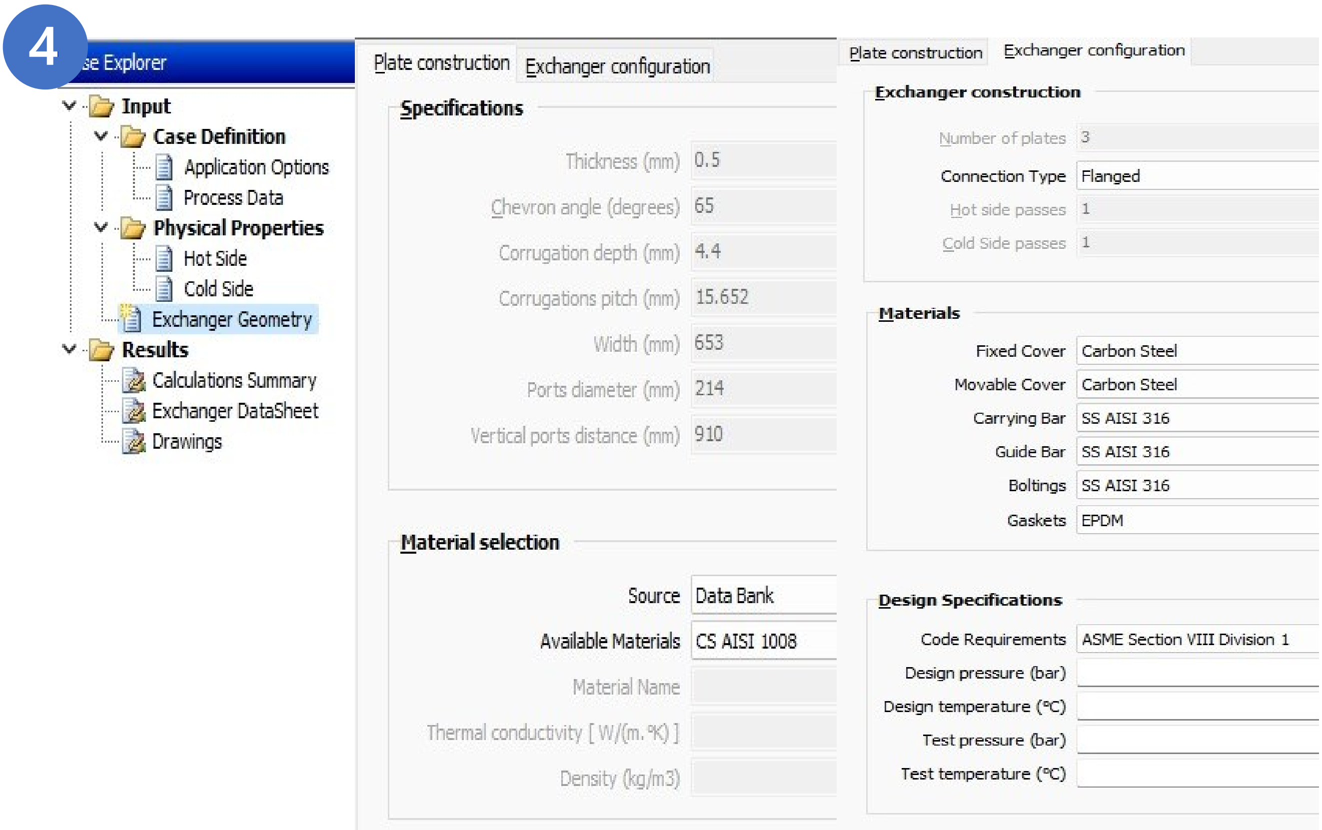


- ***Plate Heat Exchanger calculations:***
  - ***Input Data:***






3
Explorer
Input
Case Definition
Application Options
Process Data
Physical Properties
Hot Side
Cold Side
Exchanger Geometry
Results
Calculations Summary
Exchanger DataSheet
Drawings
Physical Properties
Physical properties source
Program
User specified
Available Property Packages
Peng Robinson
Soave Redlich Kwong
IAPWS 1995 Formulation for the Properties of Water and Steam
User Physical Properties
Standard Density (kg/m3) 1000
Density (kg/m3) 999
Heat Capacity [kJ/(kg.°K)] 4.18
Thermal Conductivity [W/(m.°K)] 0.598
T1 32.8
Viscosity @ T1 [N.s/m2] 0.0007
T2 40.3
Viscosity @ T2 [N.s/m2] 0.0007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se Explorer
Input
Case Definition
Application Options
Process Data
Physical Properties
Hot Side
Cold Side
Exchanger Geometry
Results
Calculations Summary
Exchanger DataSheet
Drawings
Plate construction
Exchanger configuration
Specifications
Thickness (mm) 0.5
Chevron angle (degrees) 65
Corrugation depth (mm) 4.4
Corrugations pitch (mm) 15.652
Width (mm) 653
Ports diameter (mm) 214
Vertical ports distance (mm) 910
Material selection
Source Data Bank
Available Materials CS AISI 1008
Material Name
Thermal conductivity [W/(m. °K)]
Density (kg/m3)
Plate construction
Exchanger configuration
Exchanger construction
Number of plates 3
Connection Type Flanged
Hot side passes 1
Cold Side passes 1
Materials
Fixed Cover Carbon Steel
Movable Cover Carbon Steel
Carrying Bar SS AISI 316
Guide Bar SS AISI 316
Boltings SS AISI 316
Gaskets EPDM
Design Specifications
Code Requirements ASME Section VIII Division 1
Design pressure (bar)
Design temperature (°C)
Test pressure (bar)
Test temperature (°C)

- ***Results:***

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Heat Transfer Analysis	
Required Heat Duty (W)	250
Actual Heat Duty (W)	772.696
LMTD (°C)	22.727
Clean Heat Transfer Coefficient [W/(m2.°K)]	79.955
Service Heat Transfer Coefficient [W/(m2.°K)]	63.666
Installed Heat Transfer Area (m2)	0.534
Required Heat Transfer Area (m2)	0.173
% Excess Area	0

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Streams Properties	Hot Side	Cold Side
Name	Hot stream	Cold stream
Mass Flowrate (kg/s)	0.063	0.008
Volumetric Flowrate (m3/h)	***	***
Number of Transfer Unit (NTU)	0.68	1.02
Mean Temperature (°C)	62.5	39.75
Density (kg/m3)	4.25	999
Heat Capacity [kJ/(kg.°K)]	0.797	4.18
Viscosity [N.s/m2]	9E-05	7E-04
Thermal Conductivity [W/(m.°K)]	0.018	0.598
Reynolds Number Re	1807.292	29.526
Prandth Number Pr	3.984	4.893
Wall Temperature Tw (°C)	49.286	49.272
Viscosity @ Tw [N.s/m2]	9E-05	7E-04
Mass Velocity [kg/(m2.s)]	21.843	2.775
Heat Transfer Coefficient [W/(m2.°K)]	112.066	279.77

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Pressure drop Analysis		
Velocity Through Channels (m/s)	5.139	0.003
Velocity Through Ports (m/s)	0.411	2.219E-04
Mass flowrate per Channel (kg/s)	0.063	0.008
Friction Pressure Drop (bar)	0.04	1.549E-05
Ports Pressure Drop (bar)	5.373E-06	3.69E-10
Total Pressure Drop (bar)	0.04	1.549E-05
Hydraulic Diameter (mm)	7.447	7.447
Number of channels per pass	1	1

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Construction of One Unit			
Design Pressure (bar)		Code Requirements	ASME Section VIII Division 1
Design Temperature (°C)		Connections Type	Flanged
Test Pressure (bar)		Gaskets Material	EPDM
Test Temperature (°C)		# Passes (Hot / Cold)	1 / 1
Plate Specifications			
Plate Thickness (mm)	0.5	Plate Material	CS AISI 1008
Chevron Angle (deg)	65	Plate Pack Length (mm)	14.7
Ports Diameter (mm)	214	Corrugation Depth (mm)	4.4
Plate Width (mm)	653	Corrugations Pitch (mm)	15.652
Vertical Ports Distance (mm)	910	Plate Pitch (mm)	4.9
Number of plates	3	Thermal Conductivity [W/(m.°K)]	53.302

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Performance of One Unit				
	Hot Side		Cold Side	
Fluid Name	Hot stream		Cold stream	
Mass Flowrate (kg/s)	0.063		0.008	
Temperatures (In/Out) (°C)	65	60	36	43.5
Density (kg/m3)	4.25		999	
Heat Capacity [kJ/(kg.°K)]	0.797		4.18	
Viscosity [N.s/m2]	9E-05		7E-04	
Thermal Conductivity [W/(m.°K)]	0.018		0.598	
Inlet Pressure (bar)	17		1	
Fouling Factors [(m2.°K)/W]	2E-04		0.003	
Heat Transfer Coefficient [W/(m2.°K)]	112.066		279.77	
Velocity (m/s)	5.139		0.003	
Pressure Drop (Calculated/Allowed) (bar)	0.04	0.1	1.549E-05	0.1
Heat Exchanged (W)	772.696	LMTD (°C)	22.727	
Transfer Rate, Service [W/(m2.°K)]	63.666	Clean [W/(m2.°K)]	79.955	

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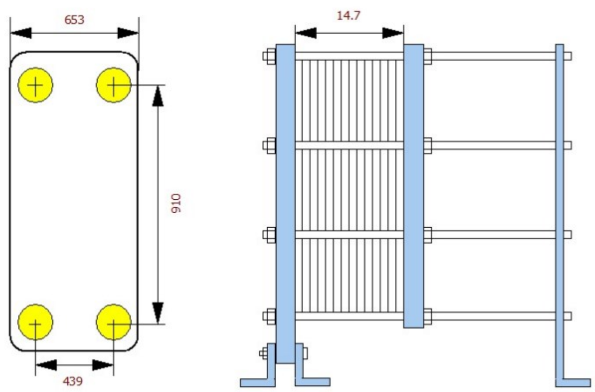

PLATE HEAT EXCHANGER MATERIAL LIST	
PART NAME	MATERIAL
FIXED COVER	Carbon Steel
MOVABLE COVER	Carbon Steel
CARRYING BAR	SS AISI 316
GUIDE BAR	SS AISI 316
PLATES	CS AISI 1008
GASKETS	EPDM
BOLTING	SS AISI 316

CODE REQUIREMENTS	ASME Section VIII Division 1
DESIGN PRESSURE (bar)	
TEST PRESSURE (bar)	
DESIGN TEMPERATURE (°C)	
NUMBER OF PLATES	3 Plates
NUMBER OF PASSES (HS/CS)	1 / 1
CONNECTIONS TYPE	Flanged
HEAT TRANSFER AREA (m2)	0.534
PORTS DIAMETER (mm)	214
CHEVRON ANGLE (DEG)	65
PLATE PACK WEIGHT (kg)	2.808

Ain shams university

Cairo-Egypt

- ***<u>Comment:</u>***

Dimensions (length and width) of plate heat exchanger may differ from calculations as when selecting plate heat exchanger from the market, the important parameter is to have the same surface area even if having different dimensions from calculations.

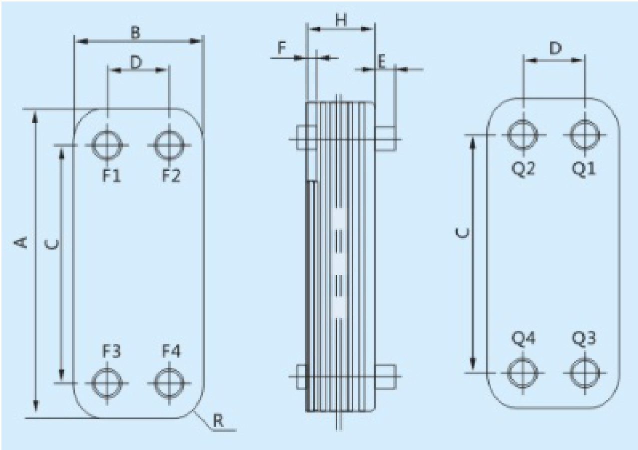

Plate Heat Exchanger Specifications by Model

Model	A	B	C	D	F	H	Weight	Volume per Channel	Unit Heat Exchanger Area	Design Pressure	Refrigeration Capacity	Max Number of Plates	Max Connection Size
	mm						kg	Liters	m^2	Mpa	KW		
B3-5A	135	40	104	18	5	4.7+1.8xNP	0.04+0.0021xNP	0.006	0.005	3.0	1-5	30	1/4" NPT
B3-12A	199	85	154	40	7	13+2.34xNP	0.4+0.044xNP	0.018	0.012	1.0	1-6	60	3/4" NPT 7/8" Welded
B3-14DW	196	86	156	46		7+2.35xNP	0.4+0.096xNP	0.022	0.014				
B3-16A	220	90	180	52		7+2.24xNP	1.1+0.06xNP	0.029	0.016		2-8		
B3-23A	315	73	278	40			1.2+0.07xNP	0.040	0.023		3-10		
B3-27A	286	116	234	63		7+2.4xNP	1.3+0.125xNP	0.050	0.027	3.0	5-20	120	1-1/4" NPT
B3-32A			243	72			1.3+0.126xNP		0.032				1" NPT 1-1/8" Welded
B3-32DW						10+2.42xNP	1.3+1.52xNP						
B3-36A	466	73	432	40		7+2.24xNP	1.3+0.106xNP	0.063	0.036		4-15		3/4" NPT 7/8" Welded
B3-52A	523	107	466	50		15+2.4xNP	1.5+0.23xNP	0.094	0.052		10-70		1" NPT 1-1/8" Welded
B3-60A	529	124	478	73		7+2.4xNP	2.7+0.26xNP	0.111	0.060			160	
B3-63A	390	200	298	120		7+2.55xNP	5.5+0.27xNP	0.128	0.063				
B3-95A	616	191	519	92		7+2.4xNP	7.8+0.42xNP	0.250	0.095		30-200	200	2" Threaded 2-1/8" Welded
B3-105A	528	246	430	148			9.5+0.46xNP	0.290	0.105	2.0		280	3" Threaded
B3-115A	535	253	456	174		15+2.4xNP	14+0.424xNP	0.250	0.115	3.0			2" Threaded 2-1/8" Welded
Conversion	1" = 25.4mm						1 kg = 2.2 lb	1 Liter = 0.264 Gallons	$1m^2 = 10.76ft^2$	1 Mpa = 145 psi	1 KW = 3412 Btu/h		

- ***Circulating Diaphragm DC pump calculations:***
  - ***Input Data:***
    - **Pressure**

SI Units		Input Data in Yellow Fields																
Heat Transfer Fluid:																		
Water																		
Pipe System Branch	Nominal Diameter	SDR	Flow Rate	Length	Fluid Temperature	Pipe OD	Pipe ID	Fluid Freeze Point	Fluid Density	Fluid Viscosity	Fluid Velocity	Reynolds Number (Re)	Friction Factor (f)	Pressure Drop in Pipe		Minor Losses (fittings, valves)		Totals	
	(mm)	(--)	(Lpm)	(m)	(°C)	(mm)	(mm)	(°C)	(kg/m^3)	(kg/(m-s))	(m/s)	(--)	(--)	(m)	(Pa)	(m)	(Pa)	(m)	(Pa)
1	DN 25	SDR11	1	27.0	32.0	33.40	27.33	0	9.95E+02	7.64E-04	0.03	1.21E+03	5.27E-02	0.00	30.29	0.00	11.83	0.00	42.11

Input Data in Yellow Fields																		
Pipe System Branch	Fitting	Qty.	Pressure Drop		Fitting	Qty.	Pressure Drop		Fitting	Qty.	Pressure Drop		Fitting	Qty.	Pressure Drop		Total Pressure Drop in Fittings	
			(m)	(Pa)			(m)	(Pa)			(m)	(Pa)			(m)	(Pa)	(m)	(Pa)
1	90° Elbow	10	0.00	5.23	90° Elbow	10	0.00	5.23	Tee-Piece (in-line flow)	1	0.00	0.20	180° U-Bend	2	0.00	1.16	0.00	11.83

$$Pipe\ system\ >>\ pf\ =\ 42.11\ pa$$

$$Head\ =\ 1\ m\ \&\ P\ =\ 9810\ pa$$

From pressure drop analysis of plate heat exchanger input data – [BHEx]:

$P\ =\ 0.04\ bar\ =\ 4000\ pa$

Total Pressure Drop (bar)	0.04

- ***Results:***

$$\boxed{P_{max}\ =\ 125\ PSI\ =\ 861{,}844.66\ Pa\ (accepted)}$$