Ground Source Heat Pump Design

A Technical Report submitted to the Department of Mechanical and Aerospace Engineering

Presented to the Faculty of the School of Engineering and Applied Science

University of Virginia • Charlottesville, Virginia

In Partial Fulfillment of the Requirements for the Degree

Bachelor of Science, School of Engineering

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Spring, 2022.

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On my honor as a University Student, I have neither given nor received unauthorized aid on this assignment as defined by the Honor Guidelines for Thesis-Related Assignments

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NOMENCLATURE

The following table delineates various acronyms and abbreviations utilized throughout the report. Provided page numbers locate the first appearance of each one. Common unit abbreviations are excluded.

Abbreviation	Description	Page
AASHTO	American Association of State Highway and Transportation Officials	11
ASTM	American Society for Testing and Materials	11
AWWA	American Water Works Association	11
CPVC	Chlorinated Polyvinyl Chloride	11
CSA	Canadian Standards Association	11
COP	Coefficient of Performance	6
GLHE	Ground Loop Heat Exchanger	10
GSHP	Ground Source Heat Pump	9
HDPE	High-Density Polyethylene	10
HVAC	Heating, Ventilation, and Air Conditioning	7
IGSHPA	International Ground Source Heat Pump Association	29
NSF	National Sanitation Foundation	11

Abbreviation	Description	Page
AASHTO	American Association of State Highway and Transportation Officials	11
OD	Outer Diameter	10
OSHA	Occupational Safety and Health Administration	11
PE	Polyethylene	11
PE-AL-PE	Composite Polyethylene Aluminum	11
PE-RT	Raised Temperature Polyethylene	11
PEX	Cross-Linked Polyethylene	11
PPI	Plastic Pipe Institute	11
PP-R	Polypropylene-Random	11
SDR	Standard Dimension Ratio	10
USDA	United States Department of Agriculture	9

PART ONE

OBJECTIVES

This technical team is committed to designing an innovative geothermal heating and cooling system as part of a net-zero residential home design initiative. This initiative is a class-wide endeavor, as two other teams will be working on new approaches to materials/insulation and energy generation/storage systems. When working with complex housing systems, simulation and computer analysis must be accompanied by hands-on experimentation, which is why all three teams will be utilizing the vacant reCOVER building on Milton Airfield at the University of Virginia for application and testing purposes.

The team will be focusing on designing a compact and efficient ground loop heat exchanger that will be connected to an existing heat pump. Using blueprints obtained from the Architecture school, the team will predict heating and cooling loads for the space to accurately determine what capacity is necessary for our system's heat pump. Once a reasonable unit is determined based on these predictions, the next step will be to identify the optimal conditions for the system such as working fluid, pipe material, and pipe dimensions. SolidWorks will be used to design and model varying ground-loop configurations, which will be imported to ANSYS Fluent to run simulations on each design with appropriate boundary conditions based on initial calculation. Each model will then be evaluated on effectiveness and cost, specifically pumping and material costs. After modeling various ground-loop configurations, the ground loop which minimizes excavation volume and maximizes heat transfer will be constructed and its performance will be evaluated against the model predictions.

From an educational perspective, the overarching goal of the technical project is to enable current and future students to address real-world environmental issues by applying the theoretical knowledge gained during their undergraduate studies at the University of Virginia. There is still plenty of room for advancement in residential energy technologies, especially with regards to efficiency and emissions. Who better to lead the charge for affordable and accessible sustainable energy than the next generation of homeowners and consumers?

BACKGROUND

EXISTING SYSTEMS

Current geothermal heat pumps are the most efficient heat pumps on the market. Studies have shown that ground source heat pumps use half the energy of a traditional air-to-air variable refrigerant flow system (Buehrer, 2021). We began our analysis of existing heat pumps with the Energy Star Most Efficient geothermal heat pumps, which is a list of the highest rated heat pumps by Energy Star for 2021. Top units on this list included the Bosch CDi Series SM and Carrier GC Series which had coefficients of performance (COP) nearing 5. This means that they can transfer five times more energy than they require as input. These units however are not cheap. According to the Bosch website, the cost of the unit including installation is between $12,000 and $40,000. Even the more affordable models, such as the Water Furnace 3 Series costs between $3,300 and $6,300, not including installation. With these high costs for units and installation we want to see how we can improve the heat transfer and design of the ground loop to increase efficiency and drive down cost.

Using the Energy Star list, we selected several two-ton models that spanned a range of costs and efficiencies. For each of the five units selected, two affordable units, one mid-range unit, and two high end units, we calculated the temperature change through the ground loop, our area of interest. The calculated values are summarized in the table below.

Heat Pump (Make/Model)	Heat Rejection (Btu/h)	Heat Extraction (But/h)	Water Flow Rate (GPM)	EWT Cooling	EWT Heating	Delta T Cooling	Delta T Heating	EER/COP
Water Furnace 3 Series	30000	18900	8	60	50	7.51	4.73	19.1/4.27
Trane T1GX Series	27900	15900	6	60	50	9.31	5.30	23.5/4.35
Geocomfort GRT Model	24600	17800	6	60	50	8.21	5.94	15.8/3.3 *
Carrier GC Series	35100	20500	6	60	50	11.71	6.84	24.1/4.2
Bosch CDi Series SM	35000	18400	6	60	50	11.68	6.14	21.6/3.86

Table I
Table with Select Heat Pump Data

Based on the five units we need to obtain a temperature change between 7°F and 12°F for cooling and between 4°F and 7°F for heating through the full loop. These values were all based around 60°F and 50°F entering water temperature for cooling and heating, respectively, which are values we think are reasonable for the climate in Charlottesville. Rules of thumb for design of the ground loop include that you need around 600 ft of pipe per ton of cooling and a separate ground loop for each ton of cooling so that there is no interference of heat transfer between pipe segments.

DESIGN CONSTRAINTS

reCOVER HOUSE THERMAL LOADS

The loads for the reCOVER house were calculated using Trace 3D Plus load design software, a commonly used tool in industry for accurately sizing HVAC systems. Trace's program uses the U.S. Department of Energy's open source EnergyPlus® simulation engine to model heat transfer through walls of buildings with multiple spaces and exterior exposures. The reCOVER house was modeled as shown below and the simulation was run to determine days of maximum heating and cooling for the location in Charlottesville, Virginia.

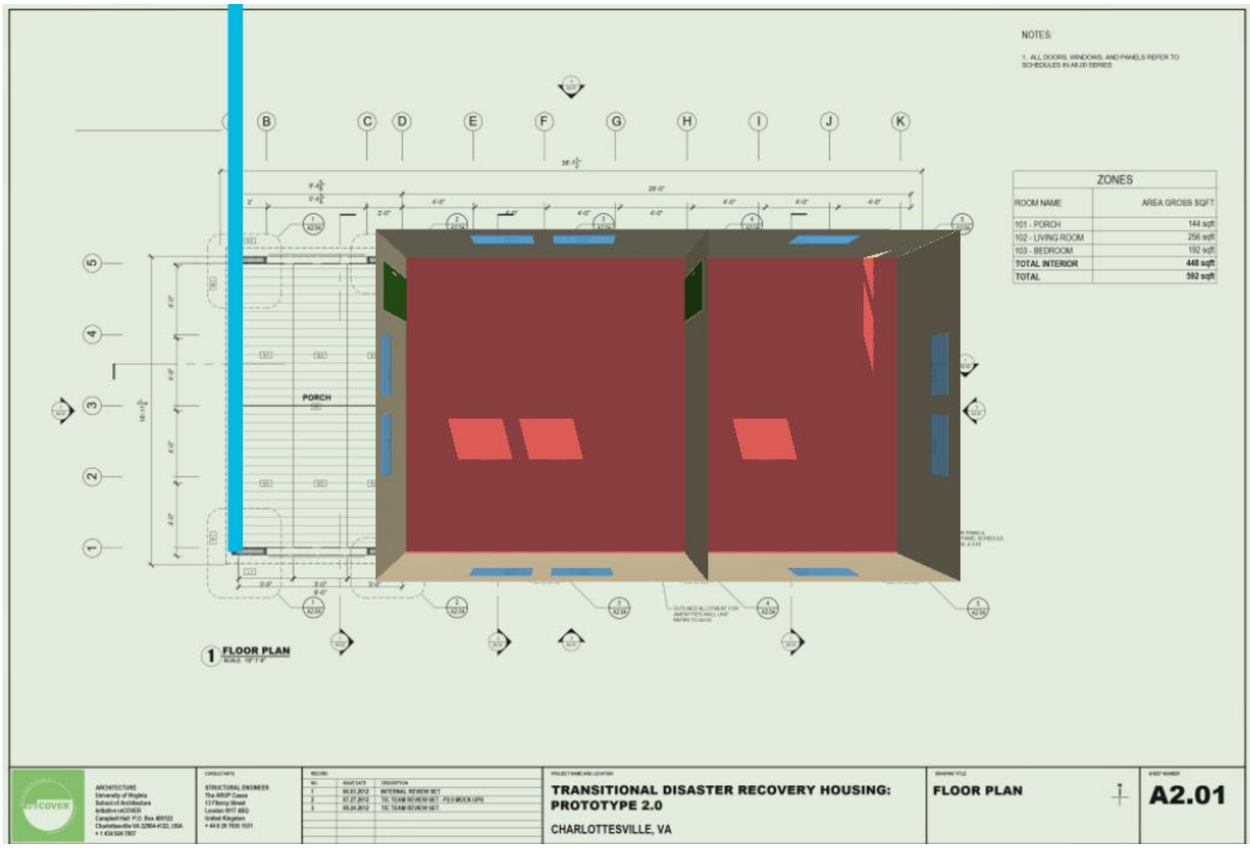

Figure 1: reCOVER Building Floor Plan in Trace3D+ Modeling Software

	Instant Sensible (Btu/h)	Time Delay Sensible (Btu/h)	Latent (Btu/h)	Total (Btu/h)	Percent of Total (%)
Roof	0	-1408	0	-1408	12.2
Other Roof	0	0	0	0	0.0
Glass	-2784	0	0	-2784	24.2
Door	0	0	0	0	0.0
Wall	0	-4700	0	-4700	40.9
Below-Grade Wall	0	0	0	0	0.0
Other Wall	0	0	0	0	0.0
Partition	0	0	0	0	0.0
Exterior Floor	0	0	0	0	0.0
Interior Floor	0	32	0	32	-0.3
Slab	0	153	0	153	-1.3
Other Floor	0	0	0	0	0.0
Infiltration	-2740	0	-42	-2782	24.2
Envelope Subtotal	-5524	-5923	-42	-11489	-
People	0	0	0	0	0.0
Lights	0	5	0	5	0.0
RA Sensible (Lights)	0	0	0	0	0.0
Miscellaneous Loads	0	0	0	0	0.0
Internal Subtotal	0	5	0	5	-
RA (All Other)	0	0	0	0	0.0
DOA Direct to Zone	0	0	0	0	0.0
Ceiling	0	-9	0	-9	0.1
Refrigeration	0	0	0	0	0.0
Service Water	0	0	0	0	0.0
HVAC Equipment Losses	0	0	0	0	0.0
Adj Air Transfer Heat	0	0	0	0	0.0
Sizing Factor Correction	0	0	0	0	0.0
Time Delay Correction	0	0	0	0	0.0
Grand Total	-5524	-5927	-42	-11493	100.0

	Instant Sensible (Btu/h)	Time Delay Sensible (Btu/h)	Latent (Btu/h)	Total (Btu/h)	Percent of Total (%)
Roof	0	1947	0	1947	24.2
Other Roof	0	0	0	0	0.0
Glass	1031	629	0	1660	20.6
Door	0	0	0	0	0.0
Wall	0	2671	0	2671	33.2
Below-Grade Wall	0	0	0	0	0.0
Other Wall	0	0	0	0	0.0
Partition	0	0	0	0	0.0
Exterior Floor	0	0	0	0	0.0
Interior Floor	0	-96	0	-96	-1.2
Slab	0	-102	0	-102	-1.3
Other Floor	0	0	0	0	0.0
Infiltration	655	0	512	1166	14.5
Envelope Subtotal	1685	5048	512	7245	-
People	176	209	328	713	8.9
Lights	0	17	0	17	0.2
RA Sensible (Lights)	0	0	0	0	0.0
Miscellaneous Loads	348	404	75	826	10.3
Internal Subtotal	524	630	402	1556	-
RA (All Other)	0	0	0	0	0.0
DOA Direct to Zone	0	0	0	0	0.0
Ceiling	0	-735	0	-735	-9.1
Refrigeration	0	0	0	0	0.0
Service Water	0	0	0	0	0.0
HVAC Equipment Losses	0	0	0	0	0.0
Adj Air Transfer Heat	0	0	0	0	0.0
Sizing Factor Correction	0	0	0	0	0.0
Time Delay Correction	0	-22	0	-22	-0.3
Grand Total	2209	4921	914	8045	100.0

Figure 2: Right-Hand Side: Heating Load Summary; Left-Hand Side: Cooling Load Summary

Trace3D+ outputs a summary of the load for the whole building (only two rooms in this case), breaking down percentages of how heat enters and escapes the space as well as summarizing equipment, lighting, and people loads. The maximum cooling design load was 12000 btu/h and the maximum heating design load was 8100 btu/h.

Milton Airfield Site Properties

The thermal properties of the ground surrounding the ground-source heat pump's (GSHP) heat exchanger will greatly impact the effectiveness of the device; however, such properties are difficult to determine without a rigorous sampling and testing procedure. Because of limited time and resources, the team has instead opted to estimate values using equations from literature and a soil survey of the Milton Airfield conducted by USDA's Natural Resources Conservation Service (USDA, n.d.). Below is a map showing different soil regions in the area of the reCOVER house.

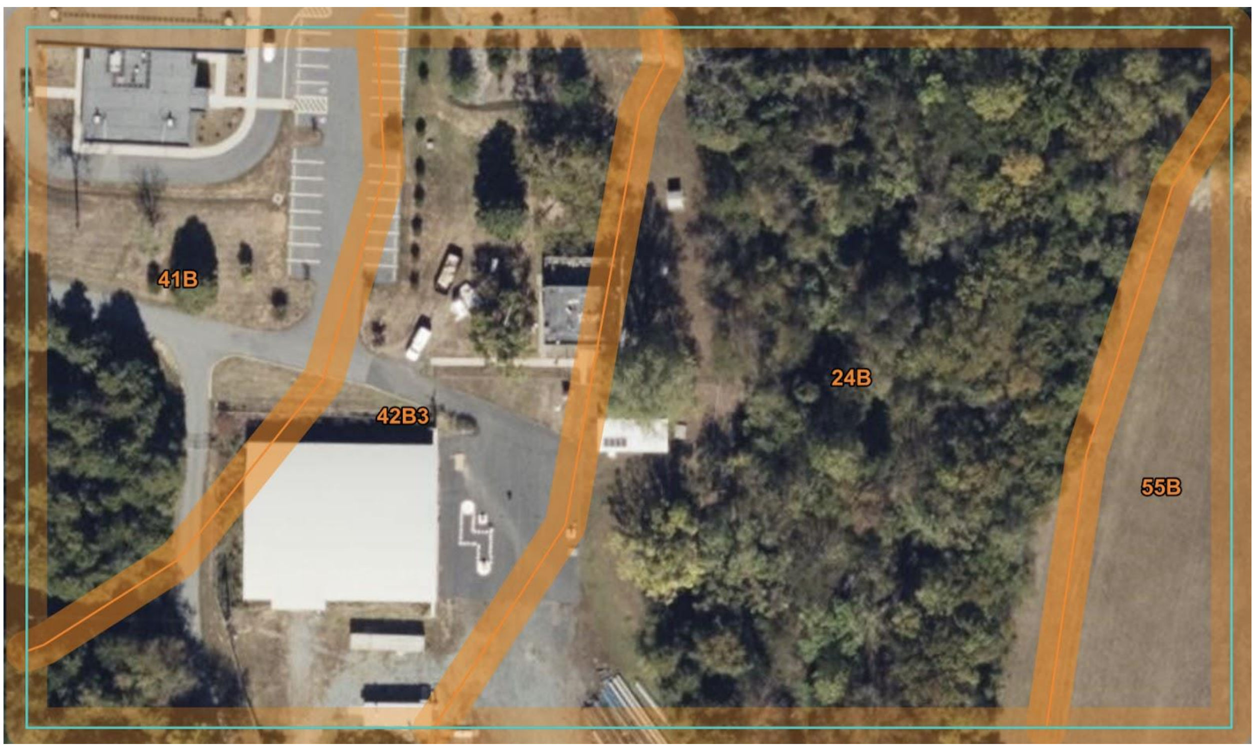

Figure 3: Aerial View of reCOVER House with Soil Region Delineation

The results of the soil survey show that region 24B takes up approximately 45.2% of the area of interest and is composed of banister silt loam. The second largest contributor to the area of interest is section 42B3 taking up 26.1% of the area of interest and composed of yadkin clay loam. With the soil texture defined, empirical equations can be used to estimate the thermal conductivity of the ground using the clay fraction (ʺΩ), the sand fraction (ʺΩ), the saturated volumetric water content (—), the volumetric water content (—), and the bulk density (ʺ). The following questions were used by Lu et al. In their 2014 model:

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| πф ʺΩ πϐ τ (3)

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Lu et al. calculated —ج values for three silt loam samples with various fractions of sand, silt, and clay as well as one clay loam sample (see table below). These values and the minimum volumetric water content for loam (35%) were used to calculate the following conductivity values (Tong, 2016):

Sample	**Sand (%)**	**Silt (%)**	**Clay (%)**	Ᵽ▼	ƚ ╥ ⅋□ ͻ╚
Silt loam 1	27	51	22	0.483	0.285
Silt loam 2	11	70	19	0.479	0.290
Silt loam 3	2	73	25	0.554	0.245
Clay loam	32	38	30	0.522	0.259

Table II
Soil Composition Table with Conductivity Values

In ANSYS modeling, the smallest thermal conductivity of 0.245 was chosen to represent the worst-case scenario. Additional thermal properties were found for general loam samples such as a bulk density of 1330 kg/m3 and a specific heat capacity of 1140 J/kg°C (USDA, n.d.; Alnefaie, 2020).

PIPE BENDING

While HDPE pipe is still considered relatively flexible, it does have a minimum allowable bend radius determined by the standard dimension ratio (SDR), which is the ratio of the nominal outside diameter, D_o, and the wall thickness of the tube, *t*.

$$SDR = \frac{D_o}{t}$$

Typical GSHP systems use a pipe with diameters between 0.75 in and 2 in. The team has selected 0.75 in OD HDPE pipe with SDR 11 as a constant design constraint across all heat exchanger designs. The minimum allowable bend radius can be calculated using:

$$R_a/D_0 > 25$$

Therefore, the minimum radius of curvature is 18.75 in (ISCO Pipe, n.d.). The bend radius ratio value of 25 includes a factor of safety of two. While this equation provides a more general value of the minimum allowable bend radius, McMaster Carr, the supplier we intend to purchase the HDPE pipe from, recommends a maximum bend radius of 15.2 in (McMaster-Carr, n.d.).

TRENCH SIZE

The team identified the large upfront investment and lengthy time frame of ground loop heat exchanger (GLHE) excavation as a major drawback for consumers. To reduce these barriers to entry, the team hopes to minimize trench size and have the GLHE operate at the shallowest possible depth. However, for the system to be operational the GLHE must be located below Virginia's freeze line of 18 in (World Population Review, n.d.). Therefore, the team has decided the GLHE must fit between the depths of 18 in and 72 in (the standard depth for horizontal GLHE units).

STANDARDS AND CODES

Ground source heat pump design and installation standards are laid out by the International Ground Source Heat Pump Association (IGSHPA) in ANSI/CSA/IGSHPA C448.

PIPE STANDARDS

Ground source loop pipe material can be CPVC, PEX, HDPE, PE-AL-PE, PP-R, PVC, or PE-RT. HDPE pipe material must follow ASTM D2737; ASTM D3035; ASTM F714; AWWA C901; CSA B137.1; CSA C448; NSF 358-1.

The Plastic Pipe Institute (PPI) recommends that the HDPE be a material with designation code PE 3608, PE 3710, PE 4608, PE 4708, or PE 4710 and color and ultraviolet stabilizer code of C or E per ASTM 3350. The minimum hydrostatic design stress should be 800 psi at 73°F. PPI also recommends that any piping material be able to withstand pressure changes of up to 60 psig due to thermal expansion/contraction of the heat transfer fluid and the pipe itself and to withstand temperature changes from 25°F to 115°F (Plastic Pipe Institute, n.d.).

PIPE FITTING STANDARDS

HDPE pipe fittings should follow ASTM D2683; ASTM D3261; ASTM F1055; CSA B137.1; CSA C448; NSF 358-1. IGSHPA recommends either a heat fusion process or a stab type mechanical fitting to provide a leak free union that is stronger than the pipe itself; however, all mechanical connections must be accessible. Therefore, a majority of the fittings of the heat exchanger will require the heat fusion method (Plastic Pipe Institute, n.d.).

PRESSURE TESTING

IGSHPA recommends that the heat exchanger be isolated and pressure tested before installation. The recommended testing procedure is operating the system at 150% of the pipe design pressure, or 300% of the system operating pressure (whichever is less), when measured from the lowest point in the loop being tested for a 30-minute test period. To ensure all air is removed from the system, IGSHPA recommends operating the system with a minimum flow rate of 2ft/min for 15 minutes (Plastic Pipe Institute, n.d.).

TRENCH STANDARDS

According to Albemarle County, pipeline construction must be made by open cut, and backfilling material must be piled in an orderly manner a sufficient distance from the banks of the trench. Trenches must be adequately shored and braced to comply with OSHA standards. Exposed ends of pipes should be fully closed by an appropriate stopper to prevent earth and other substances from entering the pipe. No more than 200 ft of trench shall be opened in advance of the completed pipe system.

Fill areas must be compacted to 95% of the optimum density determined by AASHTO T-99 before excavation begins. Certification is required in all fill areas and this certification must be signed by a professional geologist. Backfill must be deposited in 12 in layers in non-traffic areas or a thickness which will permit compaction to a density of at least 95% of the maximum density at optimum moisture content as determined by the AASHTO Standard Proctor test (AASHTO Designation T-99) (Albemarle County Service Authority, 2018).

ORGANIZATIONAL TIMELINE

At the beginning of September 2021, the team explored various ways of improving residential building sustainability. After Greg Linteris's lecture on his net-zero home, the team decided to focus on improving geothermal HVAC systems. While Dr. Linteris's geothermal system required drilling hundreds of ft into the ground; the team decided to pursue systems that would not require such specialty equipment and focus on shallow geothermal heat exchangers. After doing extensive background research on geothermal system exergy efficiency, the team identified the GLHE as the most impactful component that could be improved given the limited timeframe of the project. Having narrowed the topic from geothermal systems overall to GLHE, the team then performed a sensitivity analysis to determine the impact of flow rate, turbulence, and working fluid on heat transfer and pressure drops. Additionally, after it was determined the reCOVER house would act as the test load for the system, the team calculated the expected heating and cooling requirements. At the same time the team explored different ways to estimate the thermal properties of the soil on the site.

Beginning in mid-October and into November the team used SolidWorks to develop preliminary heat exchanger designs. It is around this time that the team decided to focus on four common GLHE configurations: straight piping, a traditional zig-zag design, a horizontally oriented slinky, and a vertically oriented slinky. In November, the team started exploring the ANSYS software and performed conjugate heat transfer simulations. Using these simulations, the team calculated design metrics such as temperature drop per unit length and pressure drop per unit length across each design. These metrics as well as overall trench volume and an ease of installation measure were used in a decision matrix to converge on a final design. The final design was then evaluated at full length in ANSYS to determine the systems expected operating values.

	September	October	November	December
Opportunity Exploration	■			
Sensitivity Analysis	■	■		
Site Specifications	■	■	■	
CAD & ANSYS Modeling		■	■	■
Final Design Evaluation			■	■

DESIGN SPECIFICATIONS

To evaluate and rank the 4 different loop configurations we established a list of five parameters that are most important to the success of the design:

1. Temperature drop per length of pipe
2. Trench dimensions
3. Cost of materials

4. Pressure loss
5. Ease of installation (Weight, volume of unit, additional equipment required, time of install)

Using the Fluent software, we can calculate each of these parameters for a small section of pipe modeled in each design.

The first parameter, temperature drop per length of pipe, will indicate how effective the configuration is at transferring heat. This is the most important factor as a greater heat transfer means our unit will be more efficient. It may also have a direct relationship with the second parameter, the trench dimensions. If one of the designs is more efficient at transferring heat, less pipe will be needed to get the desired temperature drop through the full loop, meaning a smaller trench may be possible. Ideally the trench dimensions should be minimized because a smaller trench means less work needs to be done to dig the trench and a smaller area is needed. Moreover, digging the trench is one of the most expensive parts of installing a ground source heat pump. Minimizing the trench could dramatically drive down the cost of installation. Cost will not only be influenced by the trench dimensions, but also the amount of pipe needed, the labor needed to construct each design, and any framing needed for each unit. Ideally, we would like to make our design more affordable than models and designs currently on the market.

Pressure loss must be manageable by a conventional pump used for a geothermal heat pump. Many turns in the piping might allow for a smaller footprint, but if the pressure loss is too great for the pump to handle, the design is useless. ANSYS Fluent will calculate the pressure loss through a segment of pipe and we can determine if the pressure loss through the whole loop will be manageable. Initially these will be rated relative to each other, but once we have the specifications of the final heat pump we will be using, we can compare the simulation results to analytical results.

Finally, the installation will be evaluated by the weight of each unit of piping, the packability and transportability of the units, additional equipment required, and the amount of time required to install. These may be more subjective measurements but will still be essential to determine which design to go with. Using the results from ANSYS Fluent simulations, each of the designs are rated based on these factors. Whichever design scores the highest in all categories will be the design the team moves forward with for further testing and installation at the Milton Airfield.

Analysis

1D Analysis ï Axial Flow

The team first approached the ground heat exchanger (GLHE) design by performing a sensitivity analysis on a simplified heat pump system to identify areas which could be leveraged to increase heat transfer. In this sensitivity analysis the team considered the impact of pipe geometry (outer diameter and associated wall thicknesses), the flow regime (laminar vs. turbulent), and the working fluid on the temperature drop experienced across a 300 ft straight HDPE pipe GLHE under heating conditions. The calculations assumed a 57°F ground temperature and a water inlet temperature of 44.7°F as these values reflect the required temperature drop to accommodate a 1-ton heating load. The calculations were performed using a Python script with realistic and validated fluid properties from an installed CoolProp module. The following equations were taken from Chapter 8 of *Fundamentals of Heat and Mass Transfer* by Incopera and DeWitt and were used to determine the effective heat transfer coefficient and from that the expected temperature drop across the system:

[illegible] (laminar) (5)

[illegible] (turbulent) (6)

FLOW REGIME IMPACT

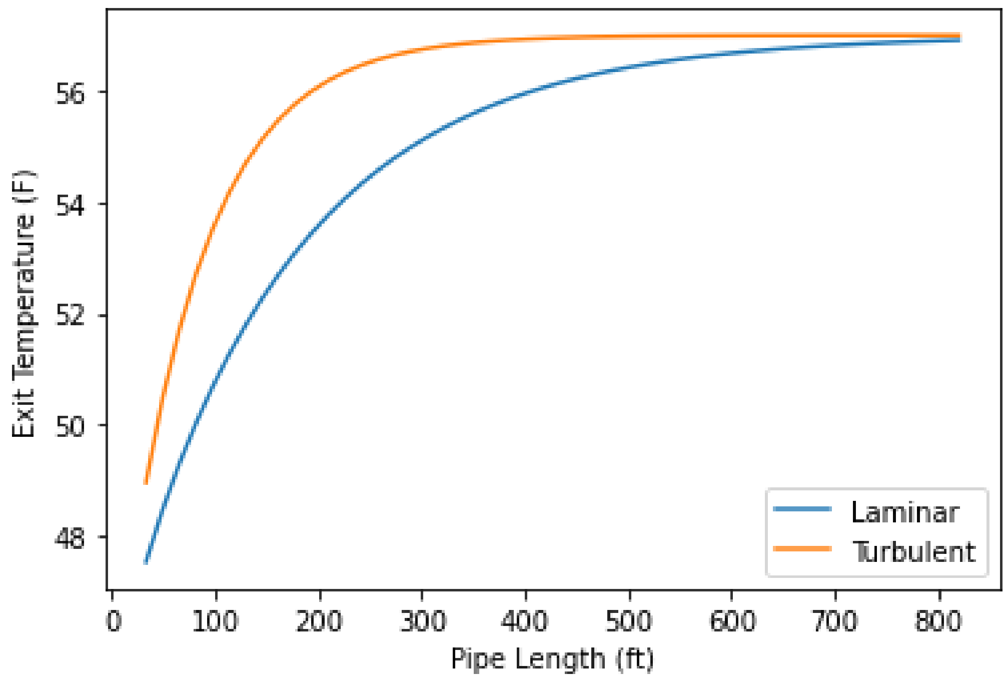

Figure 4: The Relationship Between Exit Temperature over Pipe Length for Laminar and Turbulent Flow Regimes

From the plot above, it is evident that a turbulent flow regime results in greater heat transfer compared to that of a laminar regime. This makes intuitive sense since turbulent flow is characterized by enhanced mixing. It should be noted that the mass flow rate used was 0.056 kg/s, which corresponds to the transition Reynold's number of 2300 for pipe flow. Because of the enhanced heat transfer built into the turbulent flow correlation, the turbulent flow regime requires nearly half the amount of pipe as the laminar one.

PIPE GEOMETRY

Next the team explored the impact of pipe geometry on the pressure drop and the length required to achieve the desired temperature increase of 5.3°F across the system. Pipe geometry values were taken from the EngineeringToolbox website and are for standard ASTM D3035 PE pipe which are commonly used in GLHE applications. The cost of each pipe geometry was found from hdpesupply.com; however, the cost for the ½ in nominal size pipe could not be found (“IPS DR11 HDPE straight length pipe”, n.d.). The following equations from *Fundamentals of Heat and Mass Transfer* were used to calculate the friction factor, pressure drop, and required length are as follow (Gerhart, 2019):

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Nominal Size (in)	Outer Diameter (in)	Thickness (in)	Flow Regime	Length =f 5.3°F (ft)	Pressure Drop (psi)	Cost ($)
½	0.84	0.076	Turbulent	208	24.5	-
¾	1.050	0.095	Turbulent	210	8.51	109.20
1	1.315	0.12	Turbulent	215	2.95	172.00
1 ¼	1.660	0.151	Turbulent	217	0.98	284.27
1 ½	1.900	0.173	Turbulent	220	0.52	358.60
2	2.375	0.216	Turbulent	225	0.18	380.25

Table III
Tabular Analysis of Cost, Pressure Drop, and Required Length for Varying Nominal Size

The table above indicates that pipes with smaller nominal sizes are more cost effective and require a shorter length to achieve the desired temperature drop. On the other hand, the smaller pipes also result in a much larger pressure drop across the system. Seeing as most GHSP systems are able to handle 60 psi across the heat exchanger, the increased pressure drop due to the smaller nominal size is not a concern.

WORKING FLUID

Next the team decided to consider the impact of different working fluids on the temperature drop experienced across the 300 ft system. Since many GLHE systems incorporate either ethylene glycol or propylene glycol to prevent pipe freezing, the working fluids investigated where pure water and then mixtures of water and these chemicals in different ratios. Below is a summary table of the results:

Fluid	Specific Heat Capacity [illegible]	Viscosity [illegible]	Maximum T (F)	Cost ($/gal)
Water	4200	0.0014	5.645	-

Diluted Ethylene Glycol (15%)	3959	0.0021	5.802	$10.90
Diluted Ethylene Glycol (30%)	3679	0.0033	5.965	
Diluted Propylene Glycol (15%)	4007	0.0026	5.71	$20.72
Diluted Propylene Glycol (30%)	3821	0.0051	5.641	

Table IV

Table with Cost and Performance Analysis for Varying Working Fluids

The table above indicates that a mixture of 30% ethylene glycol will be the most effective working fluid as it resulted in the greatest temperature drop. Additionally, ethylene glycol is nearly two times less expensive than propylene glycol. While ethylene glycol may be the preferred working fluid, many GSHPs come with prespecified working fluids; therefore, the working fluid can only be determined after the unit has been finalized.

3D Analysis with Fluent

ANSYS Fluent software was used to determine the conjugate heat transfer of the different GLHE designs and generate important evaluation metrics like the fluid temperature change and the pressure drop across the systems. While the GLHE designs utilize different geometries to reduce the heat exchanger's footprint, all designs use the same SDR 11 pipe dimensions for a 0.75 in outer diameter. The pipe dimensions were found from McMaster Carr. All solutions were generated using Fluent standard meshing settings and each were calculated using 1,000 iterations. The common operating conditions were 0.9655 m/s water flow rate (to replicate the 3 GPM mass flow rate of the WaterFurnace 500A11), entering water temperature of 300K, and ground wall temperature of 287K (to replicate Charlottesville average ground temperature of 57°F). The solid materials were defined using the GRANTA database. The pipe was set to plastic HDPE and the ground was set to the properties determined by our soil analysis. The imposed boundary conditions were the inlet flow rate and an outlet gauge pressure of 0 Pa. A second order solution method was specified, and hybrid initialization was enabled.

Straight Pipe

Two simple straight pipe models were developed to inform dimensions for other GLHE designs, mainly reducing reaction between pipes in closer proximity. The first model, an infinite line source, was used to validate ground temperature results from the second three dimensional ANSYS model.

Radial Heat Transfer Effects

The infinite line source model provides solutions for radial heat transfer from a line with constant heat flux into an infinite medium by conduction. This model was used to determine minimum radius from buried pipe, such that ground temperature had returned to a nominal value (57°F as outlined in Model Development above). The model is mathematically given as follows and the solution of this problem is found in (Carslaw and Jaeger, 1959; Ingersoll et al., 1954) Where E_1 is the exponential integral:

[illegible] (9)

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The plot below illustrates results for the infinite line source model with soil conditions matching that of limestone. We see a general decrease in ground temperature the further from the center of the pipe or line source. Times ranging from 10 to 600 minutes, show a nominal radius of around 8 in or 0.2 meters before heat transfer into the ground cannot be seen for this straight pipe arrangement.

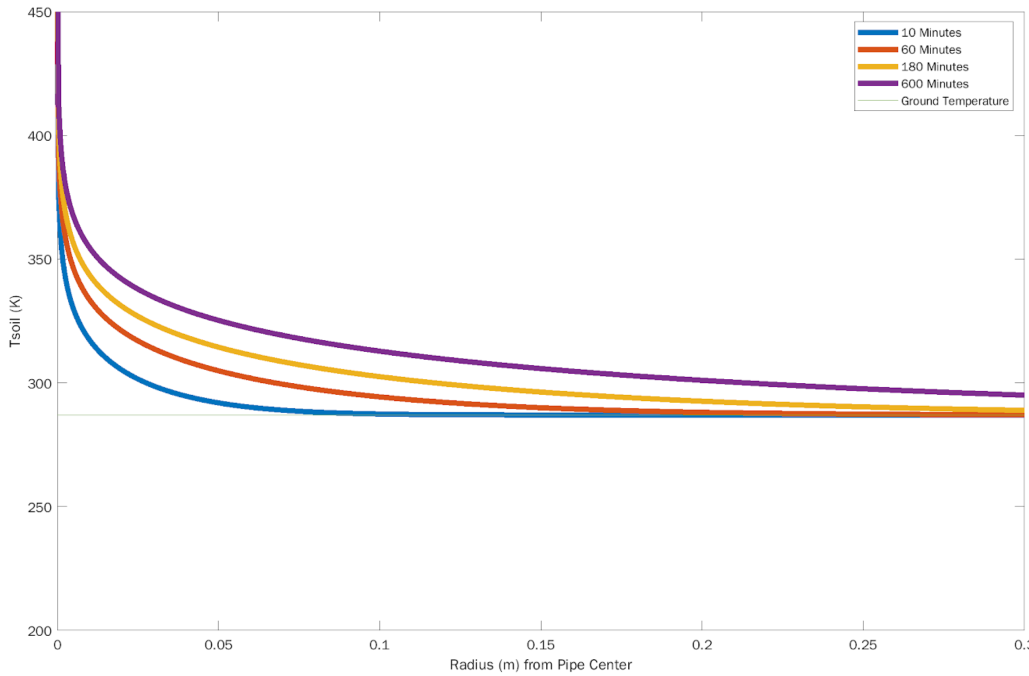

Figure 5: Plot of Soil Temperature vs. Pipe Radius for Varying Time Durations - Straight Pipe

Fluent 3D Analysis

The ANSYS modeled straight pipe uses the aforementioned pipe dimensions and has a length of 10 ft through an earth medium of at least a half meter on each side. Traditional systems typically use 500-600 ft of pipe; however to reduce computational feasibility the length was decreased. All metrics used to evaluate the designs will be on a per unit length basis to generate comparable results. Additionally, the overall volume of the trench required for each design will be considered and used as a metric for the ease of installation. For the straight pipe segment, the occupied volume was calculated assuming a trench of depth 19 in (just below the Virginia freeze line), a width of 2 ft (the average backhoe width), and the length equal to that of the pipe length.

From the fluent model the temperature and pressure drop across the 10 ft long system was determined to be 1.37K and 0.40 psi, respectively. Additionally, using the probe feature it was determined that the impacts of the flowing fluid were felt up to 0.194 m or roughly 7.64 in away from the center of the pipe.

While a pipe spacing of at least 8 in would optimize heat transfer, it will significantly increase the footprints of the GLHEs. For the team to achieve our goal of creating a GLHE that is easy to install and does not require a large excavation area, our following designs will sacrifice some heat transfer effectiveness in order to achieve a smaller overall envelope.

Below is a Fluent screenshot showing the ground temperature profile from the center of the pipe on a plane located 0.5 m behind the inlet. This cut was used to determine the 7.64 in depth or nominal radius. These results are corroborated by the infinite line source model discussed earlier. Because the thermal properties of the ground at Milton Airfield can only be approximated, the team expects that the approximated thermal conductivity is greater than that of the actual ground. Therefore, the team expects a realistic radius even greater than 8 in will be required for the ground temperature to reach its initial value of 57°F (total distance between two pipes to be >16 in as the pipes will also radiate 8 in.).

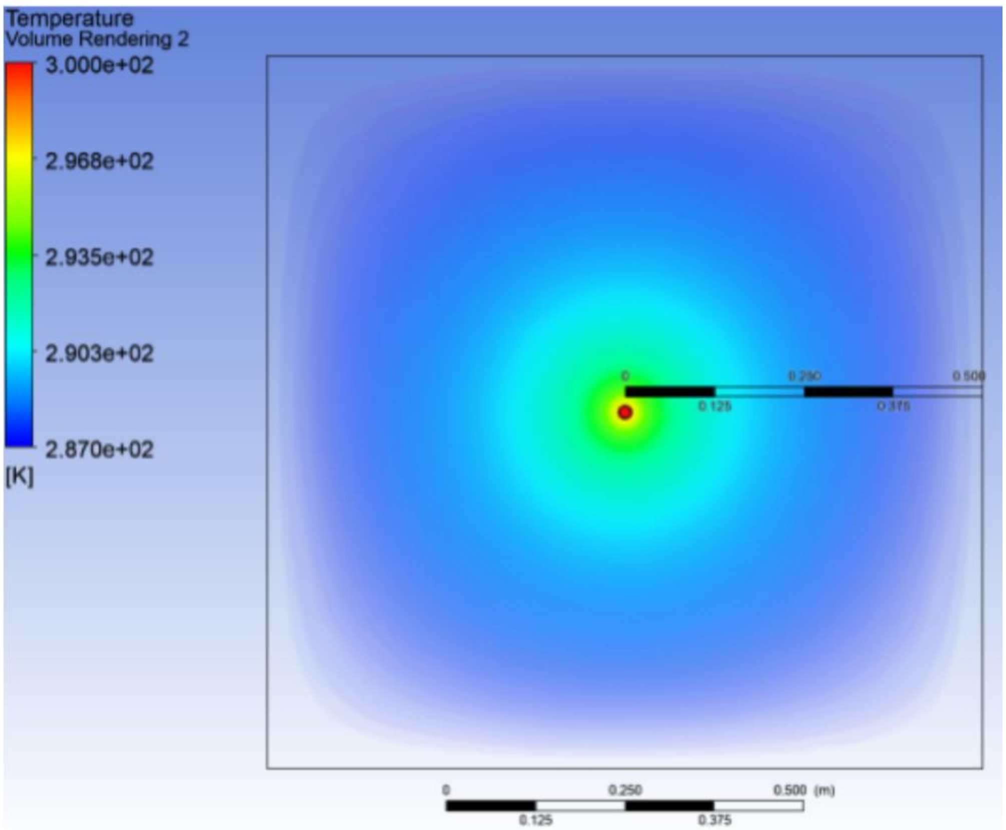

Figure 6: Ground Temperature Contour Plot - Straight Pipe

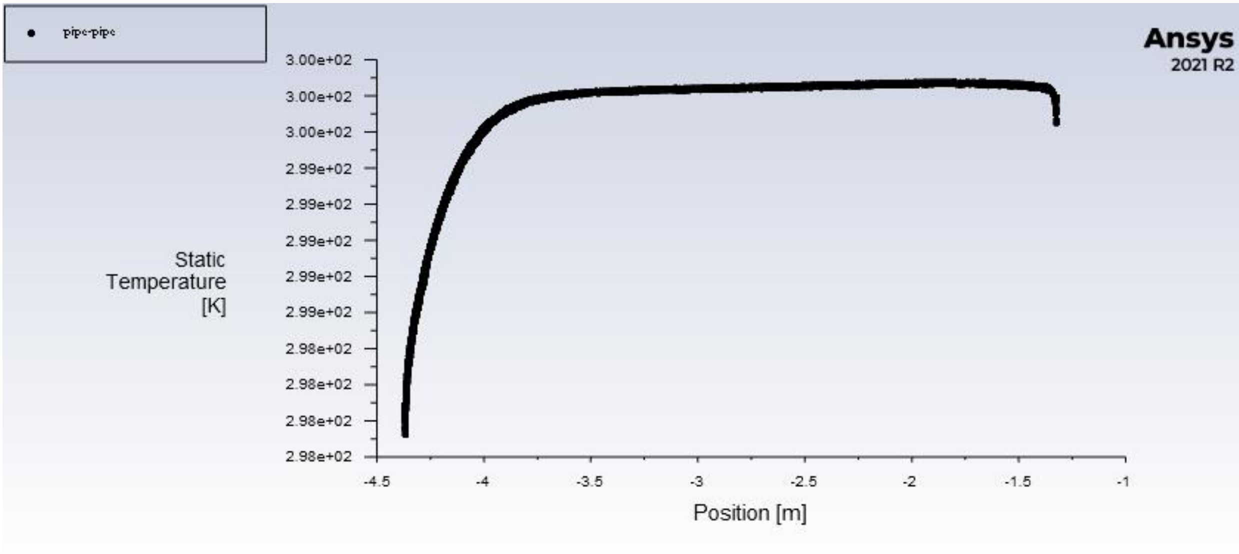

Figure 7: Static Temperature vs. Position Plot w/ Low Turbulence - Straight Pipe

To draw further conclusions from the straight pipe and corroborate results from the sensitivity analysis of flow regime, the same model was run with an increased inlet turbulence intensity. Through the velocity inlet boundary condition, “turbulent intensity” was increased from 5% to 20% prior to initialization. The new conditions yielded an increased temperature delta of 1.43K and a similar pressure delta of 0.4 psi along the 10 ft pipe length. The plot below illustrates the difference in temperature along the length of the pipe compared to the less turbulent case.

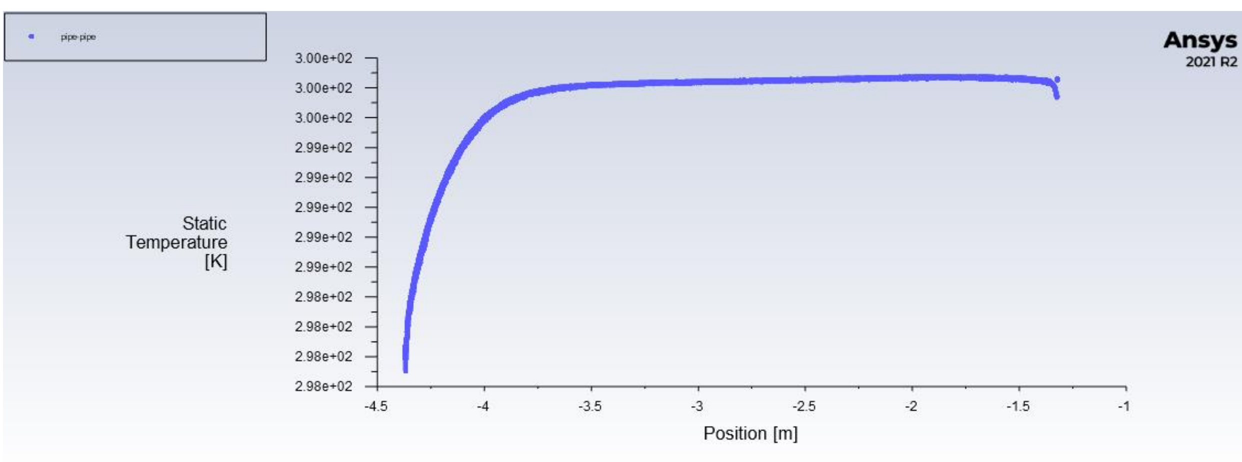

Figure 8: Static Temperature vs. Position Plot w/ High Turbulence - Straight Pipe

TRADITIONAL GHLE SYSTEMS

The traditional system attempts to maximize heat transfer by laying the pipe in a zig-zag design to increase the pipe length without needing additional space. This system is commonly used in current geothermal systems because of its easy install and increased pipe length relative to the straight system. Dimensions of the traditional design were determined by following the 16 in radius of curvature constraint of HDPE and by limiting the size so two people could carry and place the unit. This resulted in a section that was 9 ft long by 4 ft wide containing 17.5 ft of ¾ in HDPE pipe. This design is simple and proven to work in existing systems. The flat design makes it easy to transport on a truck and it is easy to place in a basic trench.

Fluent 3D Analysis

The CFD analysis of the traditional system was completed using the same initial conditions as the straight pipe described above. Boundary conditions were set to a .9655 m/s inlet velocity at 300K for the fluid and a 287K initial temperature for the ground. After loading the solidworks model into ANSYS workbench and meshing, 1,000 iterations were completed. Figure 9 shows the temperature distribution of the fluid along the flow of the pipe. The total temperature drop across the 17.5 ft of pipe was 3.1K, with a pressure loss of only 0.82 psi. Figure 10 shows the temperature distribution within the ground and demonstrates that the snaking pipe segments are placed sufficiently far apart that they do not interact with each other. Figures 9 and 10 below show the temperature change of the fluid through the pipe as well as the ground temperature profile from above.

It may be helpful to refine the calculation by improving the mesh quality in ANSYS. In many areas the tetrahedral mesh collapsed to triangular shapes and it was not optimized to fit around the curved surface of the pipe and fluid. This could cause any number of different convergence problems and will be addressed in following simulations.

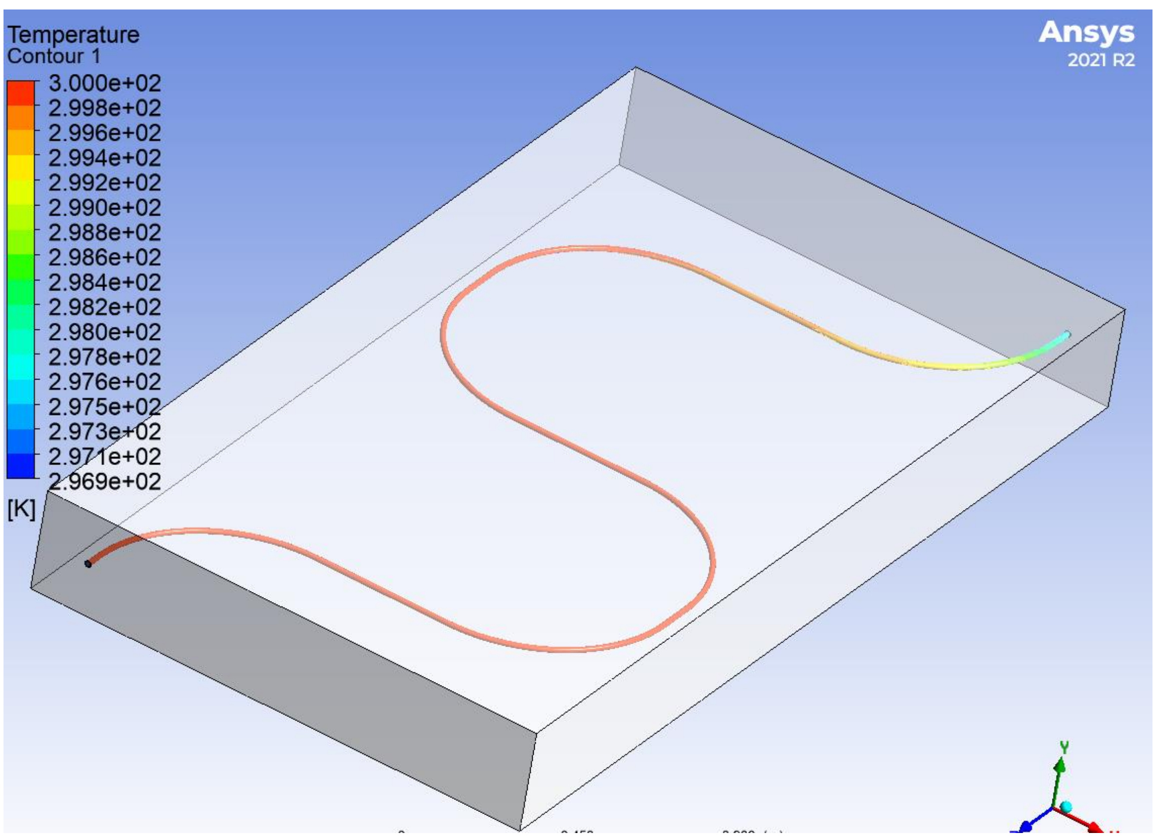

Figure 9: Temperature Distribution Contour Plot of Fluid Within Pipe - Traditional System

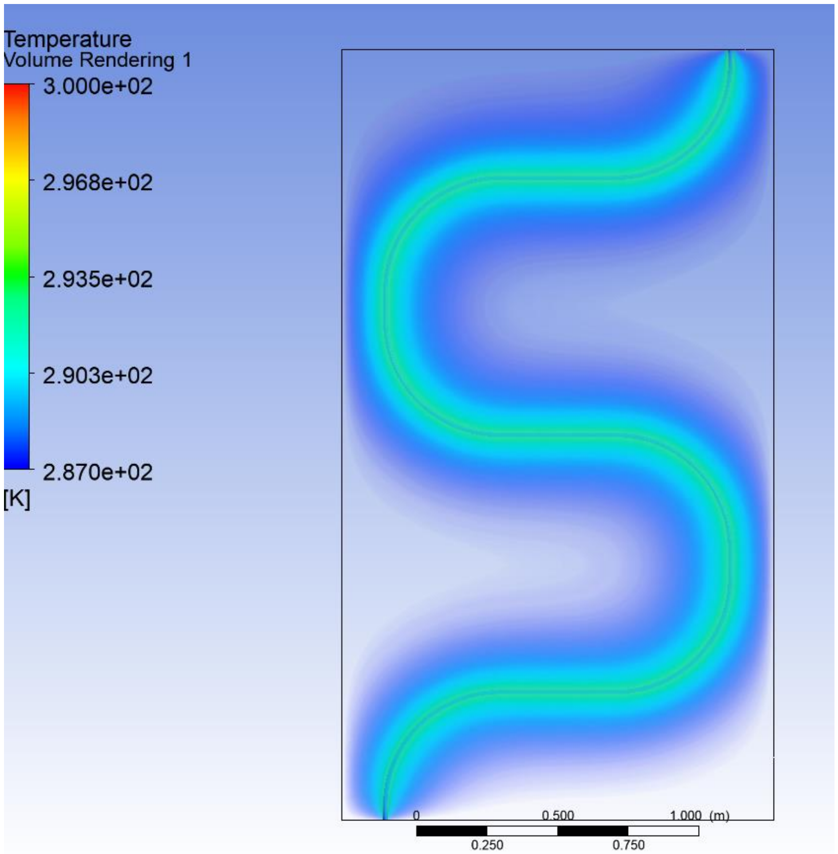

Figure 10: Temperature Distribution Contour Plot of Surrounding Ground - Traditional System

HORIZONTAL SLINKY

The horizontal slinky design is modeled after conventional horizontal ground loop heat exchangers with an overlapping single-helix design. Coil spacing and loop pitch are the primary concerns with regards to efficiency of the heat exchanger. The design was constrained by the minimum bend radius of 18.75 in for the 0.75 in OD HDPE SDR 11 piping. Additionally, the total envelope is constrained by the expected trench width of approximately 3 ft. This initial design has a coil radius of 18 in with a loop pitch of 2 in.

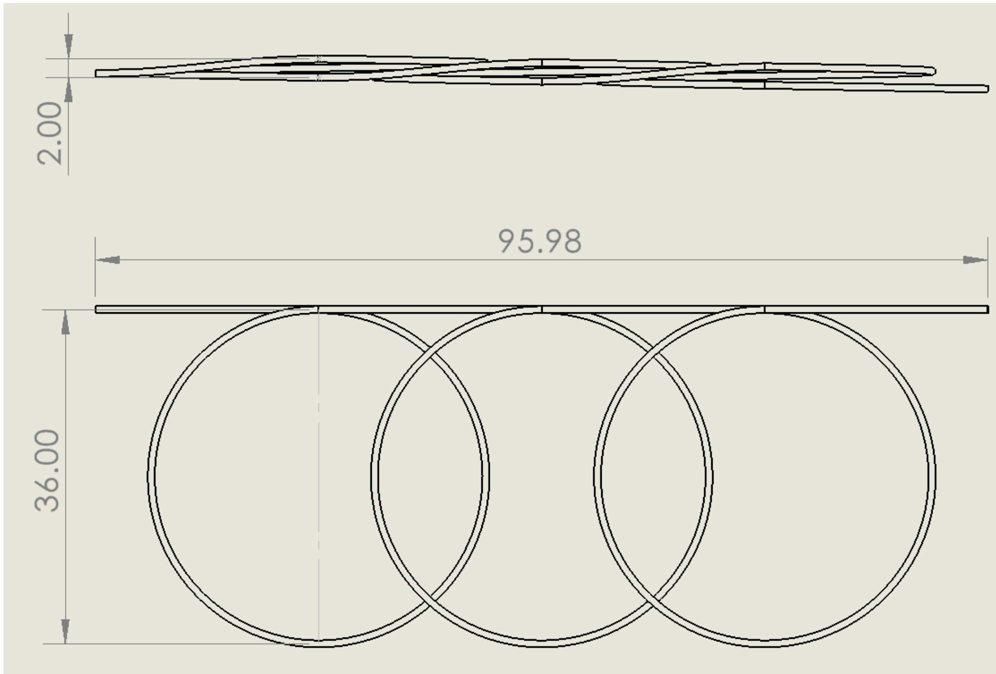

Figure 11: Top and Side Views of Horizontal Slinky-Loop Design

Fluent 3D Analysis

The CFD analysis for the slinky was completed using the same boundary and initial ground temperature conditions as outlined in the model development section. Across the slinky's total unfurled length of 36 ft, temperature dropped 5.05K and pressure dropped 1.62psi. The two volume renderings below illustrate the temperature results for the fluid in the pipe as well as the surrounding earth. Spots where the pipes do interact can be observed as the hotter areas in the ground temperature profile. This design's coil spacing, and loop pitch can be further adjusted to optimize temperature and pressure drop.

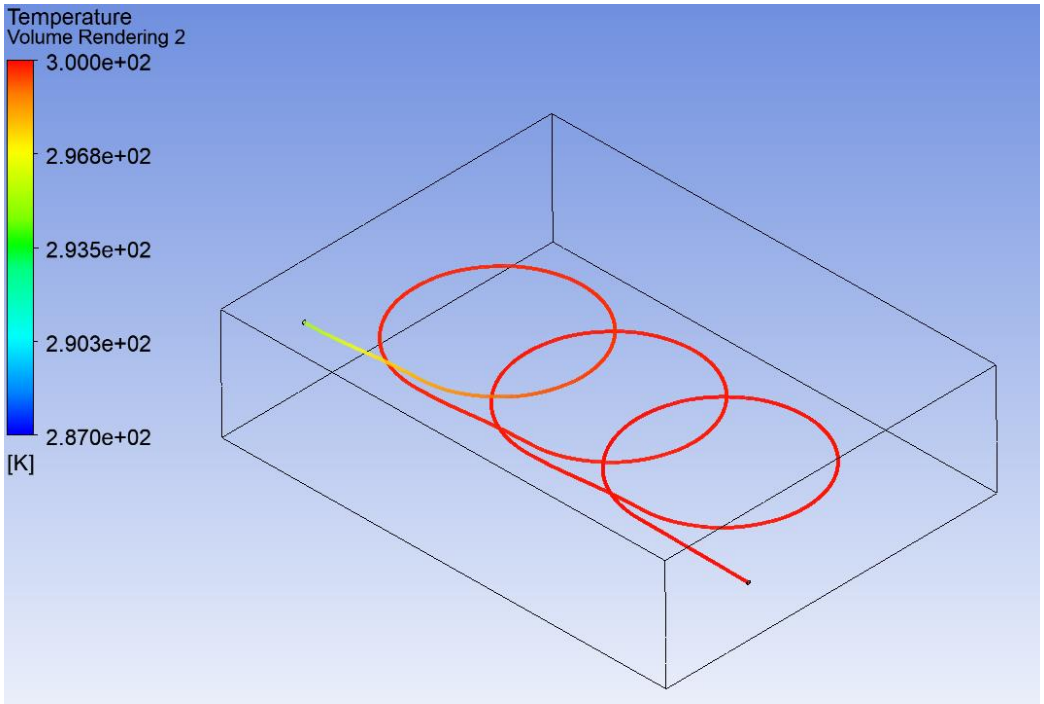

Figure 12: Temperature Distribution Contour Plot of Fluid Within - Horizontal Slinky

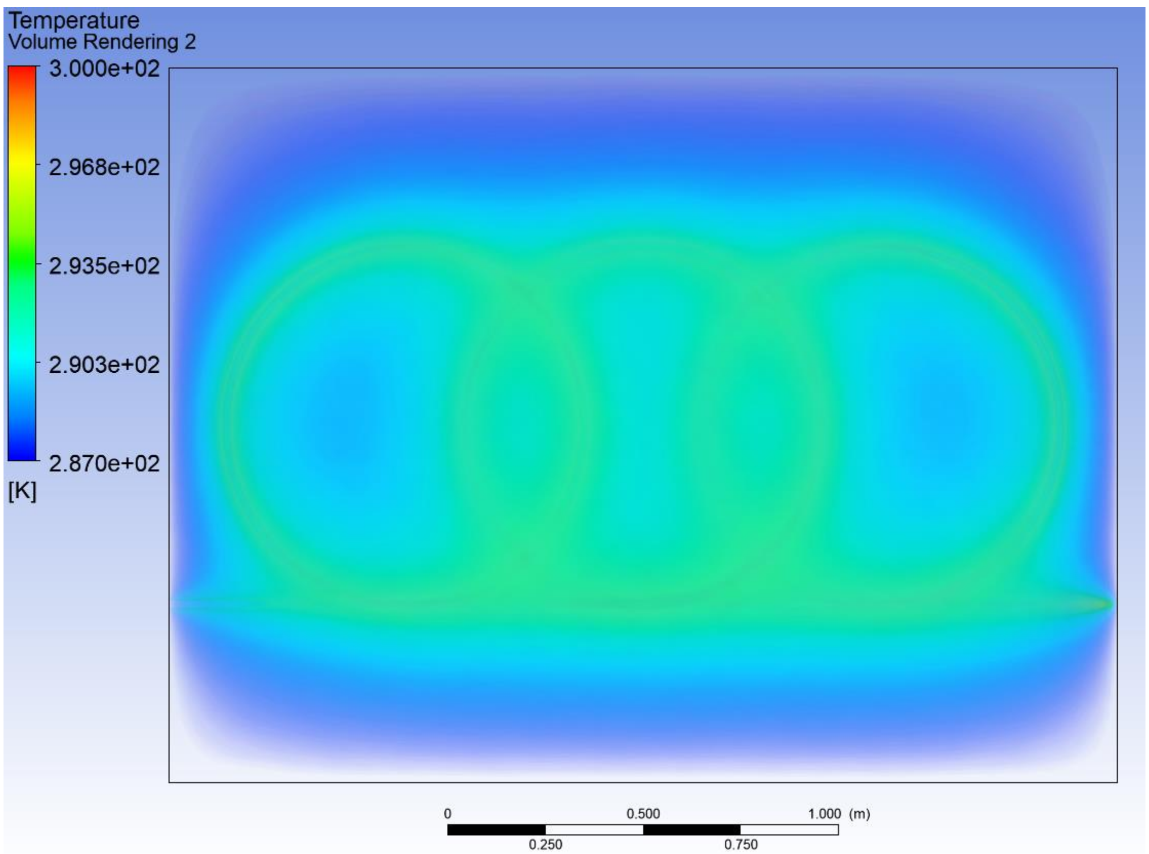

Figure 13: Temperature Distribution Contour Plot of Surrounding Ground - Horizontal Slinky

VERTICAL SLINKY

The vertical slinky design aims to minimize the horizontal trench size by orienting slinky loops downwards. To minimize the risk of pipe freezing and to reduce the installation timeline, the team constrained the design to operate between the depths of 18 in and 72 in, therefore limiting the slinky's overall height to roughly 54 in. Additionally, the design was further constrained by the minimum bend radius of 18.75 in for the 0.75 in OD HDPE SDR 11 piping. While the pipes would require a 17 in buffer zone to prevent thermal interaction, the team decided to perform an initial analysis using a 9 in pitch to ensure that each unit had a significant length of pipe. With these constraints in mind the team developed a 3D CAD model in SolidWorks, which was then imported to ANSYS Fluent to perform CFD analysis. The overall unit has a length of 62.6 ft with the primary slinky component occupying a volume of 43.9 ft^3.

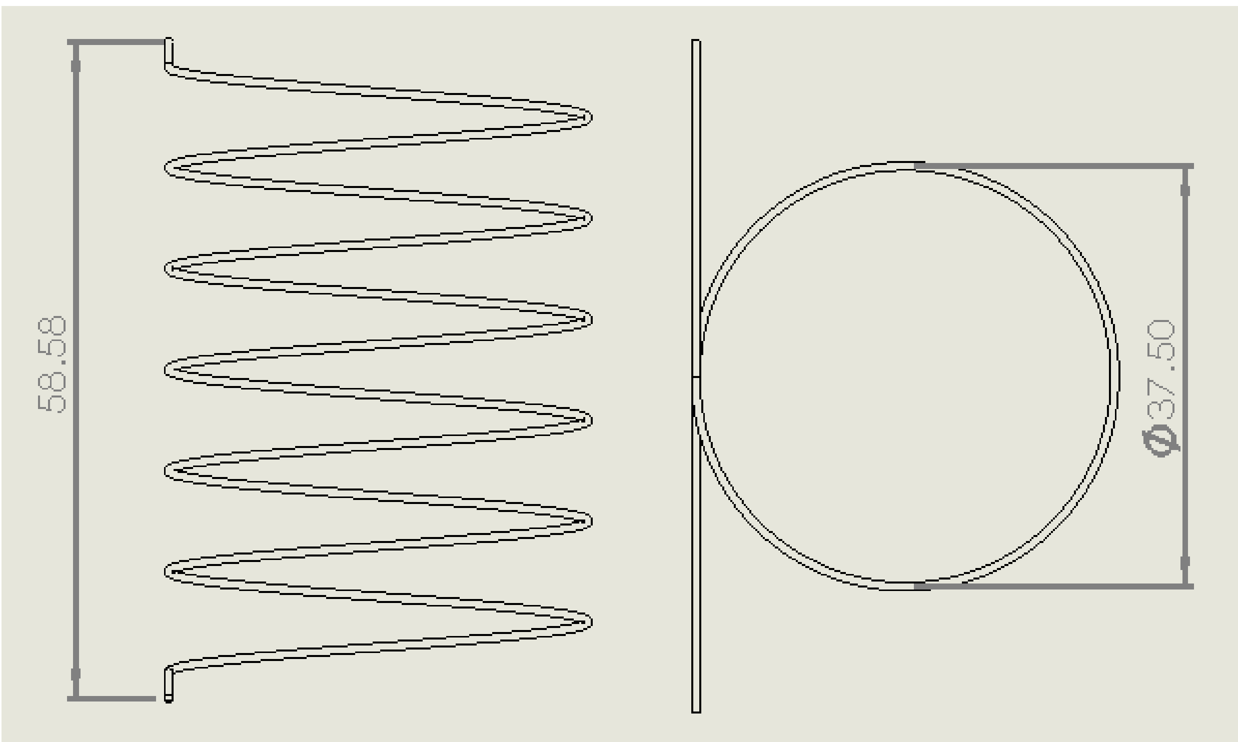

Figure 14: Top and Side Views of Vertical Slinky-Loop Design

Fluent 3D Analysis

After developing the CAD model in SolidWorks, the geometry was loaded into ANSYS Workbench and CFD analysis was performed under the same conditions as described in the straight pipe section. Because of the complicated nature of the geometry and limited computing power the solution was found using only 500 iterations; however, from the residuals plot it was clear the solution had converged. The design resulted in an 11K temperature drop with the water entering at 300K and exiting at 289K. This temperature drop is more than 5 times greater than that experienced in the straight pipe segment. Additionally, the vertical slinky experienced a 2.92 psi pressure drop which when normalized by the overall length of the design is nearly identical to that of the straight pipe. However, because these units are intended for a modular design, there may be additional pressure losses after combining units in series which are currently not reflected in these numbers.

Figure 15 below shows the temperature contours of the working fluid on the left-hand side and the temperature contours of the ground on the right-hand side. From these figures it is clear that the pipes are still thermally interacting due to the 9 in spacing. Additionally, the most significant cooling impact is felt in the bottom coil as only one side interacts with another pipe segment. Given that the majority of the cooling occurs only in the bottom segment of the structure, the significant temperature drop of 11K is dubious since this segment is very similar to that of the straight pipe previously modeled. One explanation for this significant temperature drop could be introduced or increased turbulence from the turns of the helical coil, which then enhances heat transfer. Further exploration of this design is required to understand the compounding effects of arranging units in series. Additionally, if greater computing power is available these findings should be evaluated using a 1,000 iteration solution.

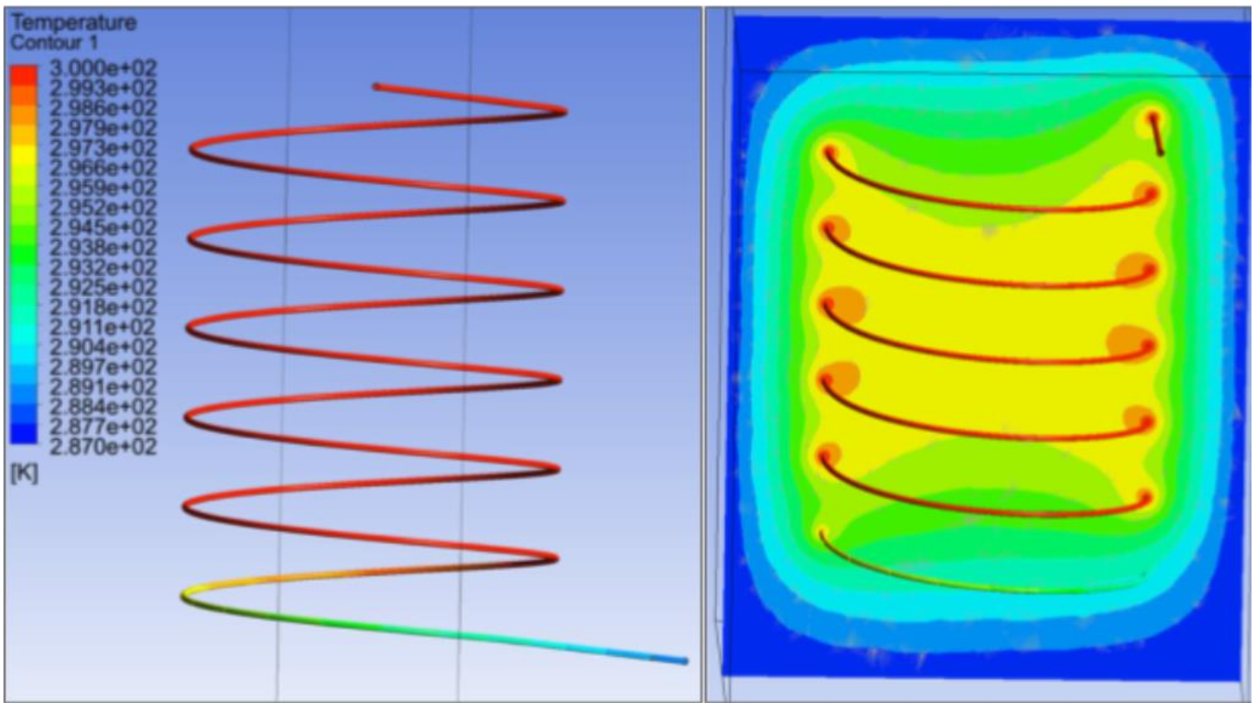

Figure 15: Temperature Contour Plots of Working Fluid (Left) and Ground (Right) - Vertical Slinky

DESIGN SELECTION

Design	Temperature Drop per Unit Length (K/ft)	Pressure Drop per Unit Length (psi/ft)	Required Trench Volume (ft3)
Straight Pipe	0.137	0.040	317
Traditional System	0.177	0.0468	54
Horizontal Slinky	0.140	0.0450	37.5

Vertical Slinky	0.176	0.0466	49.3

Table V
Technical Evaluation Table for Four Pipe Configurations

Decision Matrix
This table translates the technical values into a ranking (1 to 4) and the sum of the rankings across each category will be used to determine the best design.

Design	Temperature	Pressure	Trench Volume	Ease of Installation	Total Score
Straight Pipe	4	1	4	4	13
Traditional System	1	4	3	3	11
Horizontal Slinky	3	2	1	2	8
Vertical Slinky	2	3	2	1	8

Table VI
Decision Matrix for Four Pipe Configurations

Cost Analysis

Item	Unit Type	Units Required	Cost/Unit	Total
HDPE piping	100 ft	5	$50.87	$254.35
Backhoe	Rental	1	$400.00	$400.00
Electrofuser	n/a	1	$500.00	$500.00
8020 no.1515	1 ft	200	$9.94	$1,988.00
Sum Total:			$3,142.35	

Table VII
Cost Analysis for Trench Excavation and Pipe Installation

Conclusion

This design report presented the results of analytical simulation for four ground-loop heat exchanger configurations. ANSYS flow simulation results provided temperature and pressure drop values per unit length, and dimensioned drawings from each Solidworks model provided data for required trench volume. These factors were analyzed and compared through the use of a decision matrix, which converted data into raw ranking scores and also added a category for ease of installation. These scores were tallied in order to determine which configuration best meets the criteria determined for this project.

Since cost is a major concern for residential applications, minimizing the size of the total system footprint is a priority. As shown in the results, both the straight pipe and traditional systems had the worst scores for trench volume and ease of installation, which would indicate a relatively high upfront cost. As expected, both slinky configurations scored high in these categories. Using unoptimized coil dimensions, both the horizontal and vertical slinky models had worse temperature efficiency scores than the traditional model. The vertical slinky slowed a slightly higher temperature drop during simulation than its horizontal counterpart, while pressure drop results were slightly lower. As there is little separation in these categories, volume becomes the most significant category in determining the best configuration to meet the objectives. Thus, the horizontal slinky appears to be the best option based on initial simulation results. Smaller scale testing and additional simulation with varying coil dimensions will need to be performed to verify these results.

PART TWO

DESIGN BUILD

ISSUES WITH BUILDING RESTRICTIONS

After having identified the horizontal slinky as the optimal ground loop design which maximized heat transfer and minimized installation time and cost, the group planned to install a full-scale system at the reCOVER house. This would entail the digging of a U-shaped trench with a length of 75 ft, width of 16 ft, and depth of 3 ft on one side of the reCOVER house. Within the trench, 17 horizontal slinky units would be placed in series and form the ground loop network. Below is a dimensioned drawing of the proposed ground loop trench and slinky configuration. The total length of the buried tubing would be 613 ft.

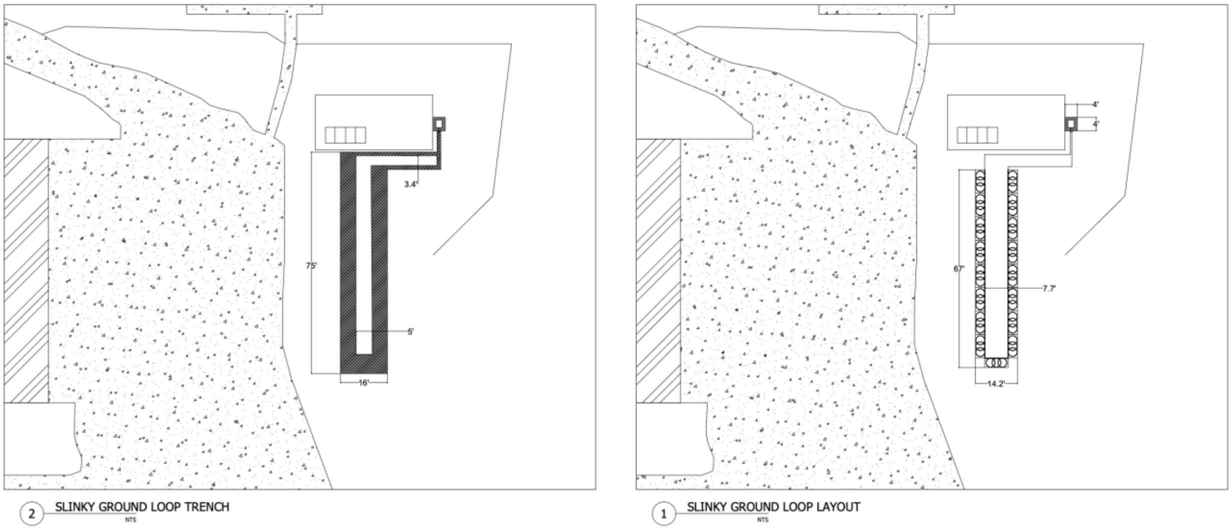

Figure 16: Left-hand side shows the trench configuration; right-hand side shows the horizontal slinky configuration

The limited space of the reCOVER house meant the team also intended to build a shelter and concrete pad for the GSHP. The shelter and pad are represented in Figure 16 by the 4 ft adjacent to the building on the right-hand side. With the installation plan in place, the team was in the process of acquiring an Enertech GeoComfort Element ZS/ZT unit from a local company. Below are diagrams showing a horizontal slinky unit and the proposed shelter for the GSHP unit.

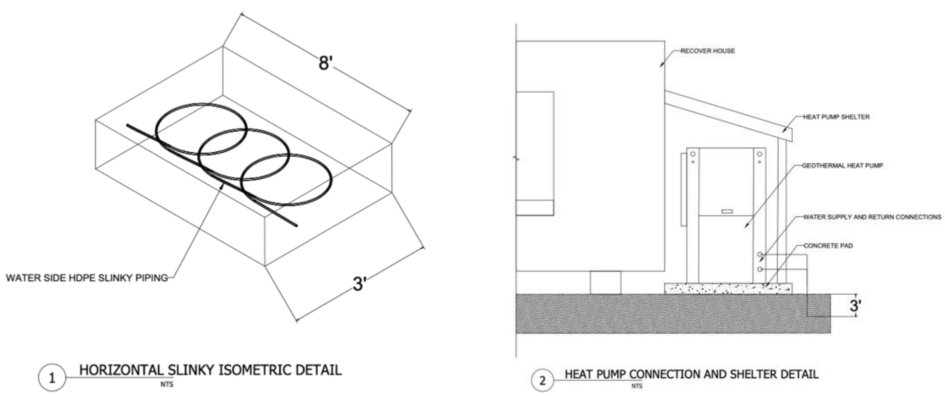

Figure 17: Left-hand side shows the horizontal slinky geometry; right-hand side shows the design for the GSHP shelter

Despite having these plans in the place, the team was unable to go forward with a full-scale installation of the ground loop design due to restrictions from UVA facilities. During the planning stages, the team primarily communicated with the UVA School of Architecture who oversaw the reCOVER house and were supportive of the building plan. However, because the build required digging on the Milton Airfield site, the installation plan needed to also be approved by UVA facilities. After lengthy discussions with UVA facilities, it became clear that team would be unable to install a full-scale design this semester. With this set-back, the team then pivoted to pursue a scaled design which could then be used to validate our ANSYS flow simulations and provide a firm, data-backed foundation for future groups to build from.

PIVOT TO A SCALED DESIGN

To determine the effectiveness of the horizontal slinky design, the loop's heat transfer, $\dot{Q}$, must be evaluated. The following equation was used to determine the heat transfer rate:

$$\dot{Q} = \dot{m}(h_o - h_i) = \dot{m}C(T_o - T_i), \tag{11}$$

Where $\dot{m}$ represents to the mass flow rate of the fluid in the loop (the working fluid), C corresponds to specific heat capacity of the working fluid, and ΔT represents the difference between the inlet and outlet temperature of the ground loop. Thermocouples were placed along the length of the ground loop to evaluate the temperature change and heat rejection across various segments of the loop. Additionally, thermocouples were placed around the ground loop to determine the excess ground temperature, θ, which was calculated with the following equation:

$$\theta = T_c - T_s \tag{12}$$

Where T_c represents the current ground temperature during a given test run and T_s represents the starting ground temperature. Using the heat transfer rate and excess temperature data from these scaled trials, the team aims to validate the results of previous ANSYS flow simulations and evaluate the effectiveness of the horizontal slinky ground loop.

HEAT SINK SELECTION

Without the ability to dig at Milton Airfield and use the Earth as a heat sink, the team had to identify a suitable heat sink to test a small section of the horizontal slinky ground loop. Initially a large water tank was considered as a potential heat sink as it would leverage the large specific heat capacity of water; however, because tanks of a sufficient size were costly and had long lead times (six to eight weeks) this option was quickly eliminated. The team eventually decided to pursue a sandbox heat sink as it would most closely mimic the heat transfer conditions of the ground surrounding the reCOVER house. As shown in the ground composition investigation earlier in the paper, the ground around the reCOVER house is primarily banister silt loam, which has a composition of roughly 30% to 40% sand (Wikipedia, n.d.). Additionally, the sandbox design also allows for measurements of thermal penetration in a solid medium and ensures that heat transfer is not enhanced by convective effects which would be present in a water bath.

GROUND LOOP PARAMETERS

After identifying the heat sink, the team then focused on scaling down the original horizontal ground loop design to one which would not require an excessively large heat sink. Initially the team pursued a one third the scale design. Below is a table showing specifications of the original ground loop and the initial scaled ground loop.

Model	Embedded Tube Length	Center-to-Center	Loop Diameter (in)	Tube Outer	Standard Dimension Ratio	Tube Inner Diameter	Flow Rate (GPM)	Reynolds Number

	(ft)	Distance (in)		Diameter (in)	(SDR)	(in)		
Full Scale Design	36	24	36	0.750	11.7	0.622	3.00	15155
One Third Scale Design	22.7	6	12	0.250	5.3	0.157	0.757	15155

Table VIII

Design Specifications for the Full Scale and One Third Scale Horizontal Slinky Ground Loops

Some important differences between the full-scale and the scaled design include the center-to-center distance and the standard dimension ratio (SDR), which is the ratio of the nominal outside diameter and the wall thickness of the tube. In the scaled design, the team opted to reduce the center-to-center distance to 6 in rather than 8 in to minimize the overall length of the slinky and therefore allow for a smaller heat sink. As mentioned previously, for HDPE pipe to be buried underground it must have an SDR of 11; however, because of limited options for 0.25-inch outer diameter piping the team had to settle for an SDR of only 5.3. While the team had to compromise on some aspects of the ground loop design, maintaining the same Reynolds number across the two designs was made a priority (Linquip, 2021). The Reynolds number for a given flow rate was kept constant to ensure dynamic similarity. Dynamic similarity is when two items of the same shape and boundary conditions are of a different size but have equal dimensionless numbers and therefore experience the same fluid flows. Maintaining the same fluid flow from the full-scale to the scaled model was imperative to ensure that any heat transfer data collected from scaled testing could be translated back to the full-scale design. Below is a schematic of the ground loop and a picture of the constructed ground loop.

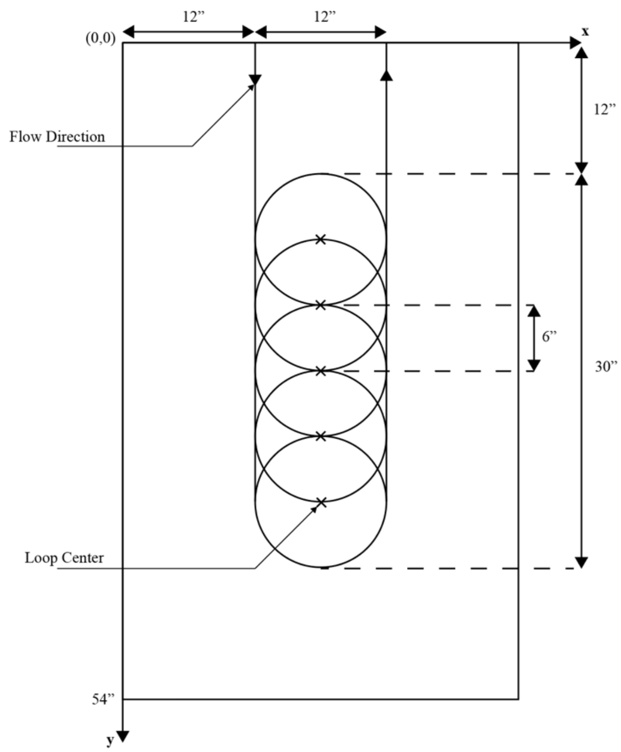

Figure 18: Right-hand side shows the dimensions of the ground loop; left-hand side shows the constructed ground loop

ISSUES WITH HEAD LOSS AND NEW SCALED DESIGN

Quickly after implementing the one third scale design, the team determined that the head loss resulting from the small cross-sectional area of the 0.25 in tubing was too large for the pumps the team had available. Without the time or access to a significantly larger pump, the team opted to increase the size of the tubing to reduce the strain on the pump and allow the system to reach flowrates like that of a real GSHP. Below is table showing how the final design compares to the initial horizontal slinky design:

Model	**Embedded Tube Length (ft)**	**Center-to-Center Distance (in)**	**Loop Diameter (in)**	**Tube Outer Diameter (in)**	**Standard Dimension Ratio (SDR)**	**Tube Inner Diameter (in)**	**Flow Rate (GPM)**	**Reynolds Number**
Full Scale Design	36	24	36	0.750	11.7	0.622	3.00	15155
Final Design	22.7	6	12	0.500	8	0.375	1.81	15155

Table IX

Design Specifications for the Full Scale and Final Design Horizontal Slinky Ground Loops

IMPLEMENTATION OF THE FINAL SCALED DESIGN

SANDBOX DESIGN

After deciding to pursue the sandbox heat sink, the team determined the necessary sandbox dimensions using ANSYS flow simulation data. From ANSYS, it was ascertained that the maximum thermal penetration from the ground loop was 7.64 in; therefore, using a factor of safety of 1.5 the team decided to use a 12 in buffer surrounding the ground loop. With this buffer, the sandbox dimensions were set to a 2 ft height, 3 ft width, and 4.5 ft length. The total volume of the box is 27 ft^3 and therefore holds approximately 2,700lbs of sand. Because of the large weight of the sand, the sandbox was reinforced along its length to ensure stability. Additionally, the bottom of the sandbox was raised off the floor using 4x4s so that the sandbox could be moved with a pallet jack if required.

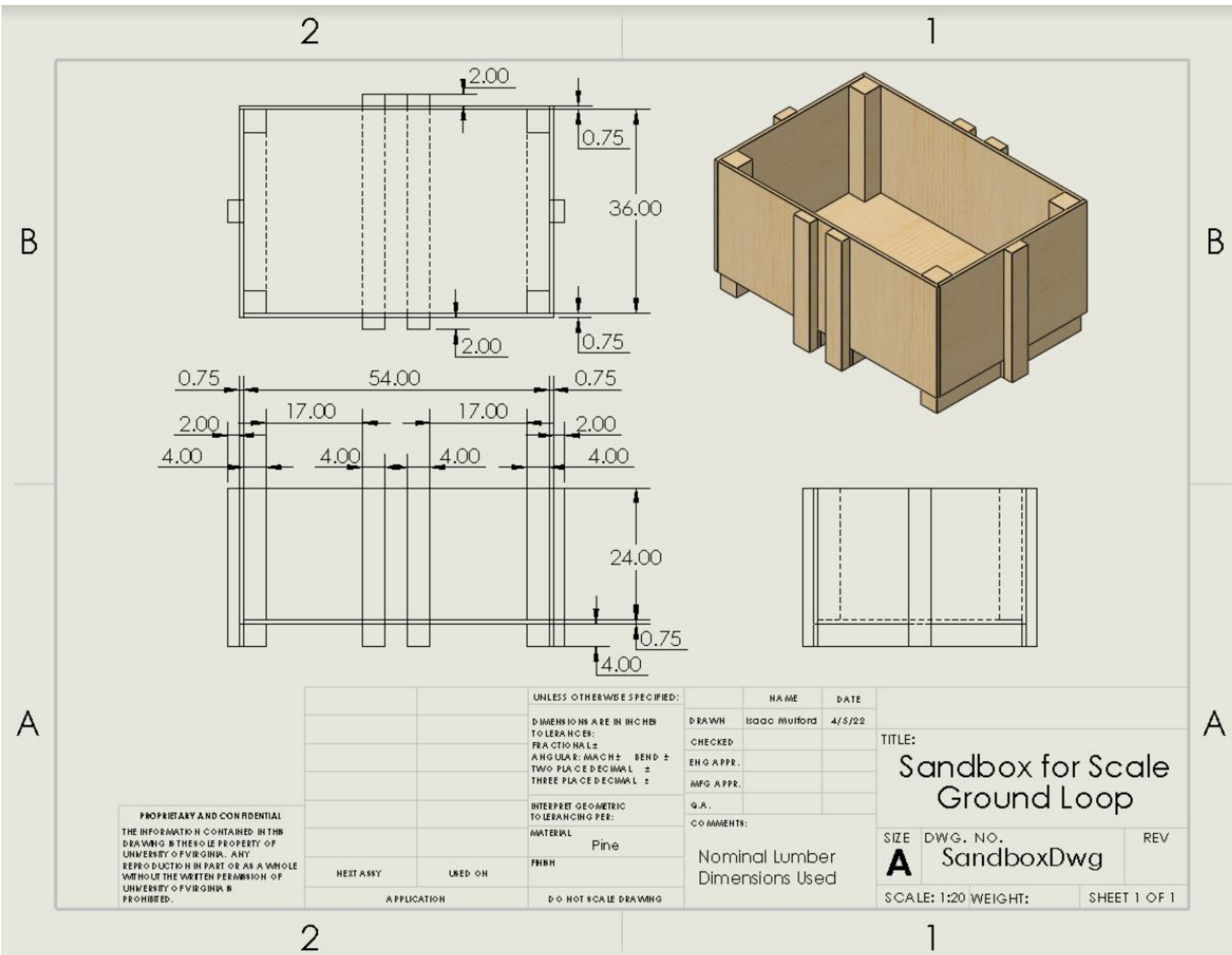

Figure 19: Dimensioned drawing of sandbox heat sink (nominal dimensions)

SYSTEM LAYOUT

Key Components:

- 1300W Immersion Bucket Water Heater

- ¾ HP Water Transfer Pump
- High-Accuracy Panel-Mount Flowmeter [0.5 gpm to 5 gpm]
- 50 Gallon Rainwater Collection Barrel
- 54 x 36 x 24 in Wooden Sandbox
- Polyethylene Tubing - ⅜ in ID, ½ in OD
- 8 Type T Thermocouples [-454°F to 700°F]
- National Instruments 9219 and 9210 Data Acquisition Modules
- Brass Panel-Mount On/Off Nickel-Plated Ball Valve - Fitting for ½ in Tube OD

In this system, water is held in an elevated 50-gallon barrel where it remains at constant temperature with the aid of an attached 1300W Immersion water heater. The water is then directed down to a 3/4HP transfer pump via HDPE piping. This pump feeds into the buried HDPE ground loop. A precision flowmeter mounted to the side of the sandbox allows for flow control at the outlet of the ground loop. For added flow control, a manual 3-way valve creates a bypass, splitting this output flow into two streams which are then separately directed into the elevated tank. This cycle continues as long as the pump is running.

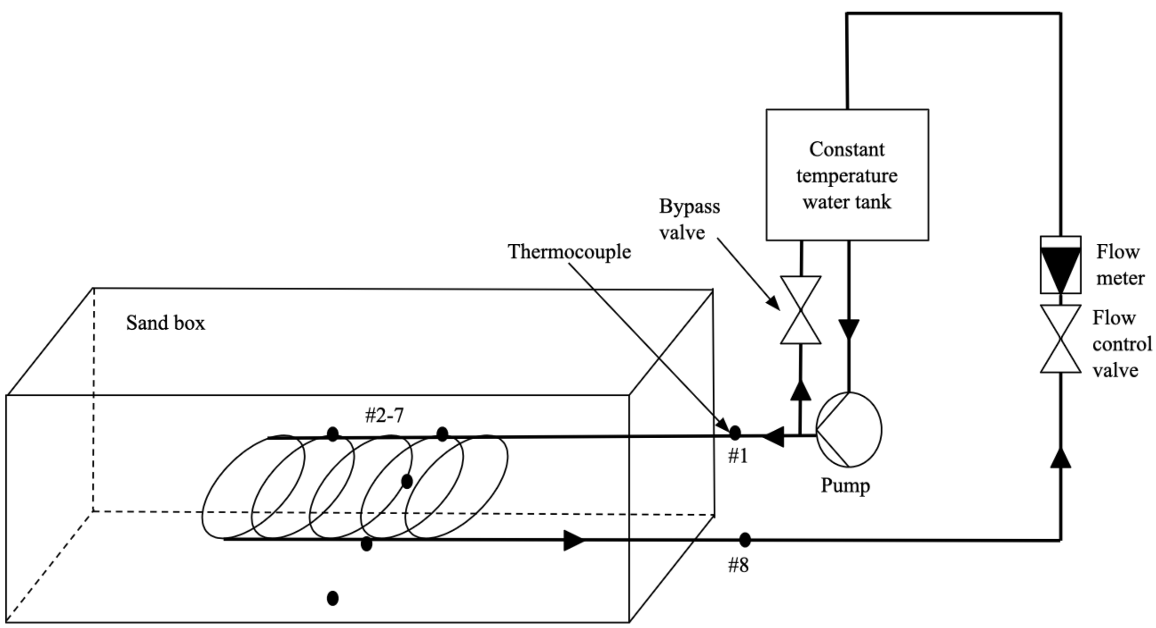

Figure 20: Schematic of system layout with key components

THERMOCOUPLE PLACEMENT

Eight T-type thermocouples were placed strategically throughout the system for data collection. Four in-line water thermocouples were then attached to the ½ in piping using inline tees with adjustable thermocouple/RTD compression fittings. Three ground thermocouples were buried in the sand such that the tips of the thermocouples were in-plane with the ground loop. (Exact positioning of each of these thermocouples is illustrated in Figure 21.) The remaining thermocouple was affixed to the inside of the tank, partially submerged in water, to monitor initial temperature during heating.

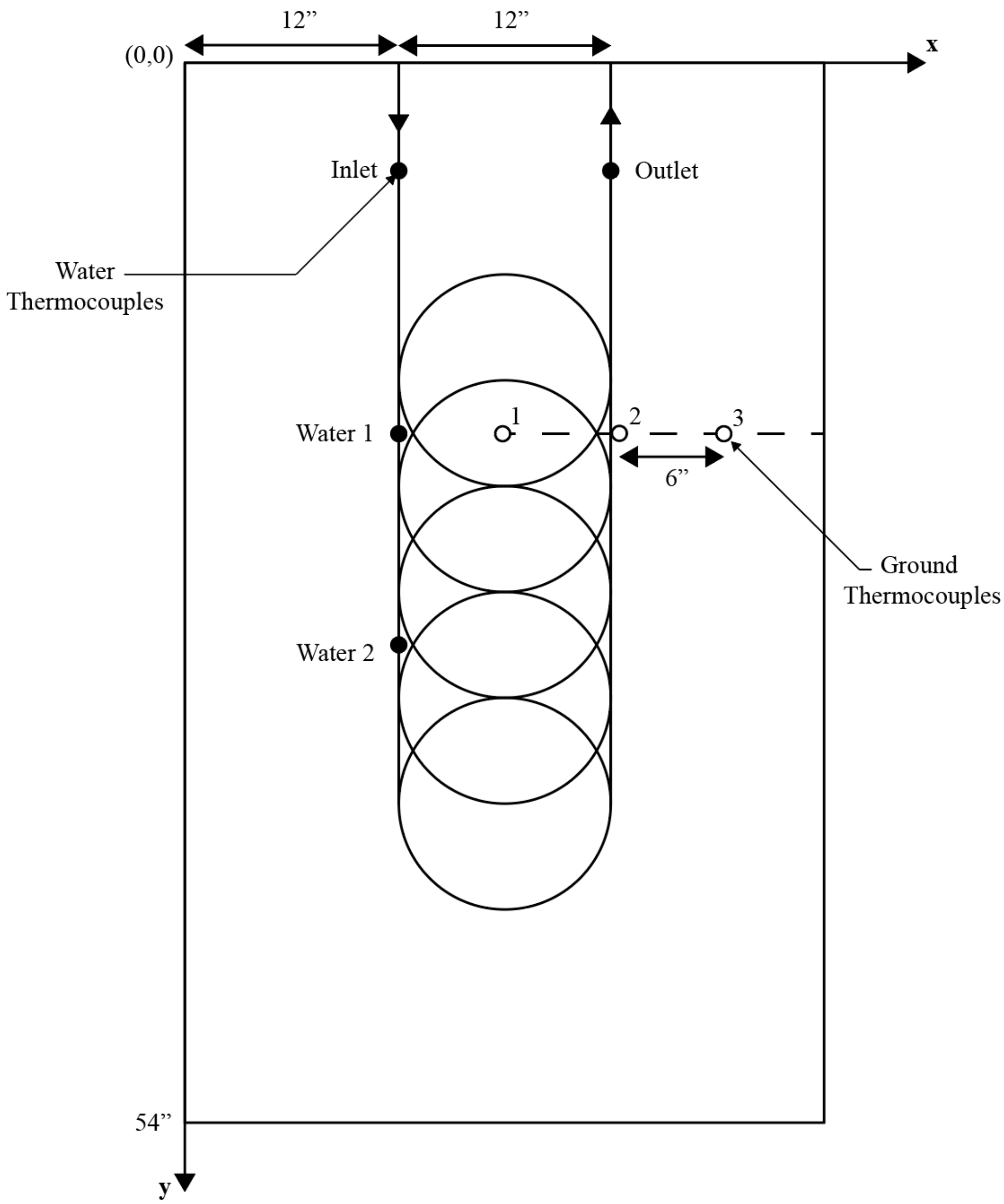

Figure 21: Ground loop schematic with thermocouple positioning

DATA LOGGING

As described in our system layout, our design required temperature to be continuously monitored at eight different points of interest. We chose to accomplish this using thermocouples and a data logger. T-Type thermocouples were selected for this application, with a temperature range of -454°F to 700°F and accuracy of +/- 0.75% these instruments balanced reasonable cost with easily repeatable measurements. The data logging hardware and software selected was built around the National Instruments (NI) LabVIEW platform. Our system used two NI modules each with the capability to measure from four thermocouples. The NI9219, C Series Universal Analog Input Module, is designed for multipurpose testing. It has the capability to measure signals from many different powered sensors such as strain gauges and load cells; however, pins 4(+) and 5(-) can be used for thermocouple measurement. The NI9210, C Series Temperature Input Module is designed specifically for thermocouple measurements with four positive negative terminals for thermocouple connections. Each of these C series modules were plugged into a powered CompactDAQ to USB interface and thermocouple wiring was carried out as shown in the diagram below.

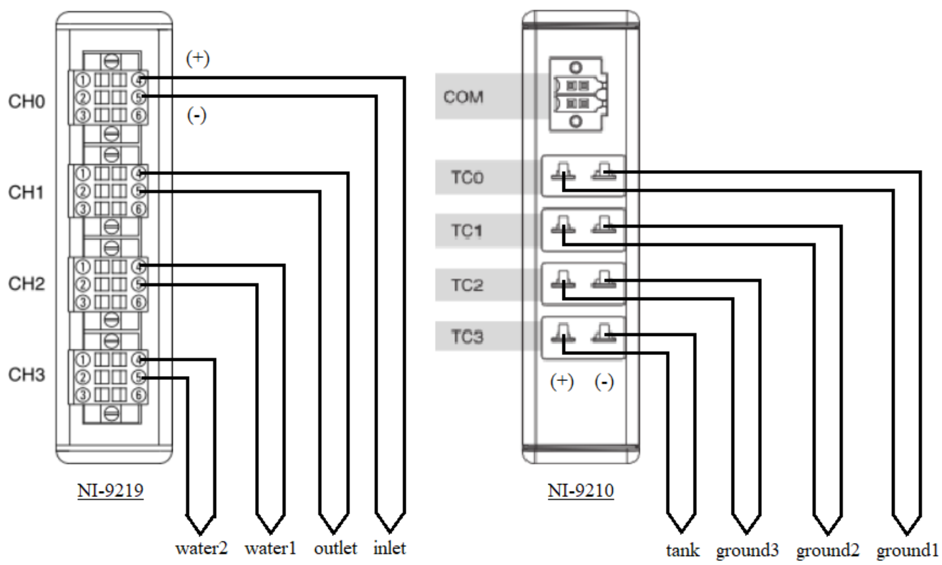

Figure 22: Wiring diagram for data logging modules

Once the thermocouples had been properly connected, they were initialized in LabVIEW software with the correct T-Type parameter and Fahrenheit units. Before being incorporated into the system, all eight thermocouples were submerged in a uniform temperature water bath to verify their accuracy and ensure they were within a degree of each other. After setting up all the physical hardware, the LabVIEW software was configured to record each point of measurement every two seconds and log the data to a timestamped excel file. LabVIEW block diagram and front panel configuration are shown below.

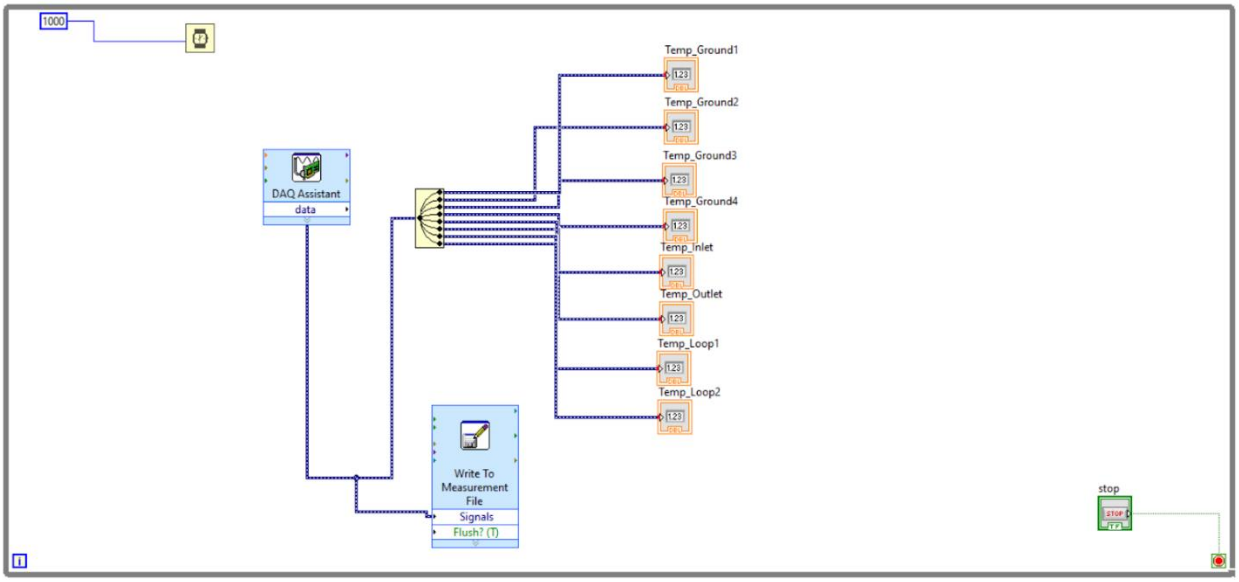

Figure 23: LabVIEW block diagram

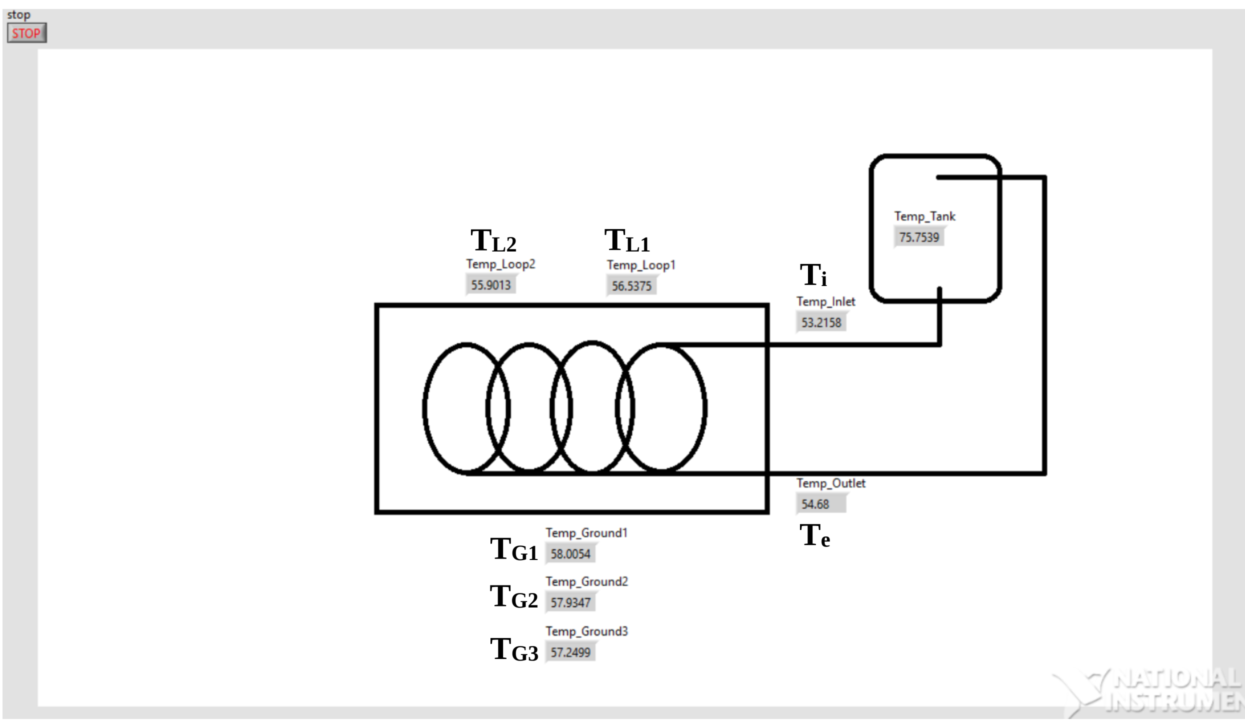

Figure 24: Front-end interface for LabVIEW data logging

PROCEDURE

A very specific order of operations was followed for each of our experiments to maintain consistency in our results given the large number of variables we needed to control per trial. First and foremost, we powered on the pump and adjusted the bypass valve and flow meter to the desired flow, powering off the pump afterwards. While waiting for ground temperature to settle at a consistent level given ambient conditions, we turned the 1300 Watt immersion heater in our 50 gallon water tank to the desired temperature for the given trial. As we waited for the water to heat, the temperature was monitored from one of our thermocouples to confirm our auto shut-off setting on the heater. Once the tank auto shut-off was triggered, data collection was initiated in LabVIEW and the pump powered back on. Throughout the duration of the experiment, temperatures were monitored from the LabVIEW front panel readout and the flow meter to confirm our desired experiment settings. The tests were run and data collected for one hour and six minutes or until our temperature exceeded the 95°F rating of our pump. The collected data was exported via timestamped excel files as described in our data logging and brought into MATLAB for graphical analysis.

RESULTS AND DISCUSSION

LOOP EVALUATION

To evaluate the effectiveness of the ground loop design and validate our ANSYS simulation findings, the team wanted to determine the heat transfer rate of the loop under different operating conditions such as with different working fluid inlet temperatures and flow rates. As defined in equation 11, the heat transfer rate of the loop is calculated using the following equation:

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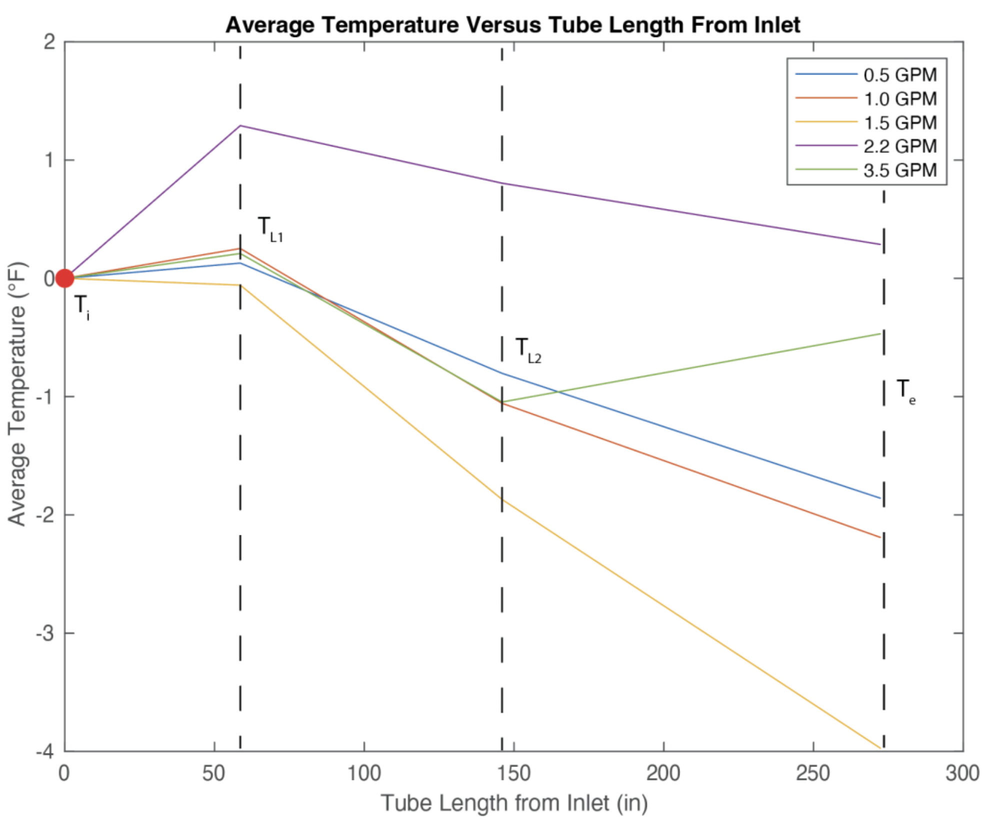

Figure 25: Average Fluid Temperature Along Ground Loop Length

Figure 25 shows the change in temperature of the working fluid at the four thermocouple locations along the ground loop. As heat is transferred from the working fluid to the heat sink, the temperature of the water decreases; therefore, a line with a steeper gradient indicates a greater rate of heat transfer. From Figure 25, it is evident that the greatest heat transfer occurred between thermocouples ˮƳؤه and ˮƳؤ. and that lower flow rates such as 0.5, 1.0, and 1.5 GPM optimized this temperature drop. Interestingly, higher flow rates such as the 2.5 and 3.5 GPM trials have significantly smaller drops, with the 3.5 GPM trial seemingly re-heating the working fluid in the ˮƳؤ. to ˮƳت segment. One potential explanation for the heating affect in the 3.5 GPM trial could be that the cooled water must pass by earlier loop segments that contain warmer water to reach the outlet.

Figure 25 also highlights that there is not a significant change in temperature between thermocouple ˮƳت and thermocouple ˮƳؤه except in the 2.2 GPM trial. In this trial, thermocouple ˮƳؤه was on average 1°F greater than ˮƳت indicating a slight heating affect from the heat sink. The team believes that this is because the inlet thermocouple, ˮƳت was exposed to ambient air while ˮƳؤه was embedded in sand and therefore more insulated. The combination of ˮƳؤه reading greater than ˮƳت and the small temperature drop across the loop at this flow rate meant that the 2.2 GPM trial resulted in a negative heat transfer value when using equation 11. To use the 2.2 GPM trial in our analysis, the team decided to calculate an adjusted heat transfer value for this trial using the difference between ˮƳؤه and ˮƳت:

Ữة ت ه خ ځ خ ت ά ت Ő ˮƳؤه ˮƳت (13)

Finally, Figure 25 shows that the 1.5 GPM trials maximized the temperature drop across the ground loop achieving an average drop of nearly -4°F. This temperature drop is roughly double the drop experienced at the 0.5 and 1.0 GPM. While the significant temperature drop of the 1.5 GPM could indicate that this flow rate optimizes ground loop performance, after plotting the impact of inlet temperature versus heat transfer (see Fig. 26 below), the team decided to exclude the 1.5 GPM trial.

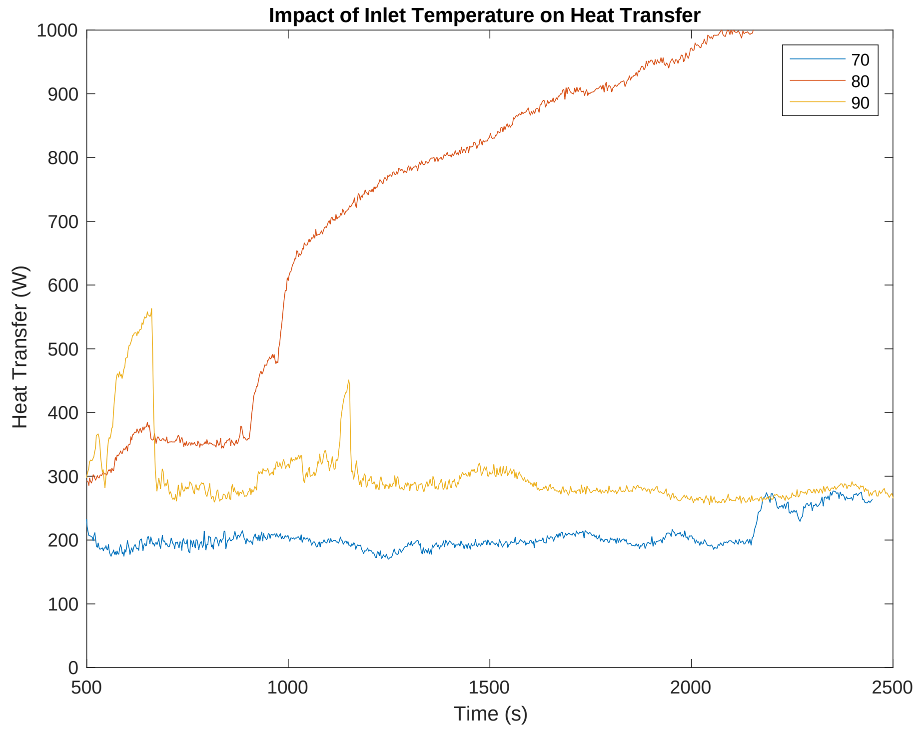

Figure 26: The Impact of Inlet Temperature on Heat Transfer at 1.5 GPM

From Fourier's law it is known that the rate of heat transfer through a material is proportional to the negative temperature gradient. Therefore, the team expected that increasing the tank temperature (and therefore inlet temperature) would yield a greater rate of heat transfer as the working fluid and the heat sink would experience a greater temperature difference and thus steeper temperature gradient. However, as Figure 26 shows, the test data does not reflect this relationship.

While the heat transfer of the 70°F trial consistently remains below the 90°F trial as expected, the heat transfer of the 80°F trial significantly exceeds both the 70°F and 90°F trials. Additionally, the 80°F trial exhibits a positive slope indicating that heat transfer is increasing over time, whereas the two other trials exhibit constant heat transfer behavior. The increasing heat transfer is likely not a result of the 80°F inlet temperature but indicates that another system component could be malfunctioning. For example, the sharp increase in heat transfer at the 900 second mark could be the result of the flow meter inaccurately reading 1.5 GPM when it was a higher value. The odd spike and increasing behavior of the 1.5 GPM cannot be explained by any physical phenomenon other than a component malfunction, therefore, the trial has been excluded from further analysis.

THE IMPACT OF FLOW RATE ON HEAT TRANSFER

Residential GSHPs can operate over a wide range of working fluid flow rates with some pumps using flowrates as low as 2.25 GPM and some as high as 23.5 GPM. Generally, it is recommended that a GSHP is operated with a 3 GPM flow rate per ton of cooling capacity. From the Trace3D+ analysis outlined earlier; the team determined the reCOVER house represented a 1 ton cooling load and therefore the ideal ground loop flow rate would be 3 GPM. The team decided to evaluate a range of flow rates around the 3 GPM mark to determine the impact this would have on the system performance.

Table X shows the impact of different test flow rates on the average temperature difference between the inlet and outlet thermocouples, ŸY ˝Y ˝Y (see Figure 24), and the average heat transfer rate of the loop. The table also includes the full-scale equivalent flow rate which is a translation from the test flow rate to a dynamically equivalent version for the full-scale design. Similarly, Table XI shows the impact of flow rate on the adjusted average temperature ȟŸY ة ﺗ ج ﺨ خ ﺧ ﺗ ˝Yؤه ˝Y, and the adjusted average heat transfer.

Test Flow Rate (GPM)	Full Scale Equivalent Flow Rate (GPM)	Reynolds Number	Average ΔT (°C/ft)	Average Heat Transfer (W)
0.5	0.83	4189	0.0455	143
1.0	1.66	8379	0.0536	629
2.2	3.65	18434	-0.0070	-109
3.5	5.80	29328	0.0115	203

Table X
The Impact of Ground Loop Flow Rates on the Average Heat Transfer

Test Flow Rate (GPM)	Full Scale Equivalent Flow Rate (GPM)	Reynolds Number	Adjusted Average ΔT (°C/ft)	Adjusted Average Heat Transfer (W)
0.5	0.83	4189	0.0487	155
1.0	1.66	8379	0.0598	441
2.2	3.65	18434	0.0246	284
3.5	5.80	29328	0.0166	421

Table XI
The Impact of Ground Loop Flow Rates on the Adjusted Average Heat Transfer

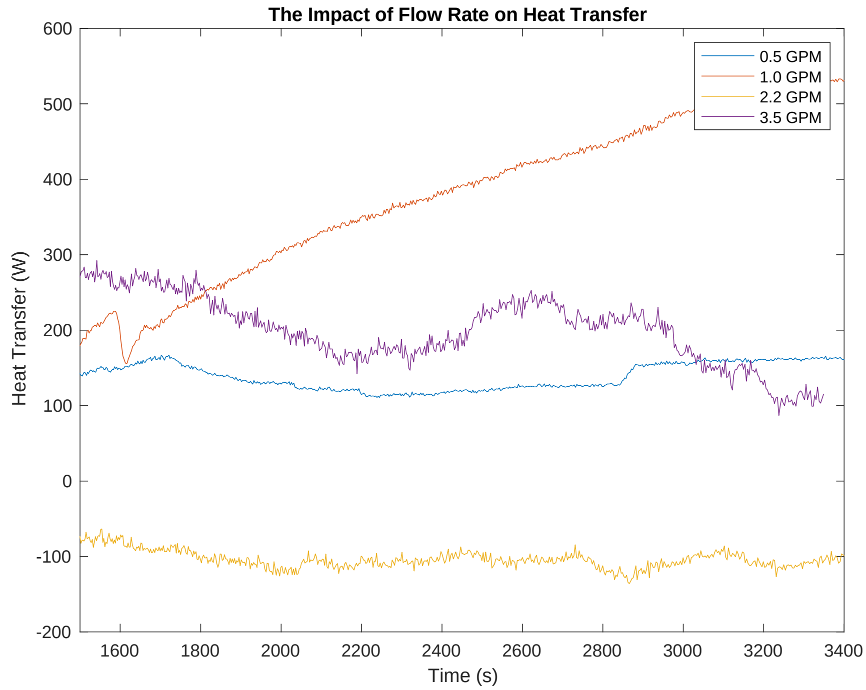

Figure 27: The Impact of Flow Rate on Ground Loop Heat Transfer

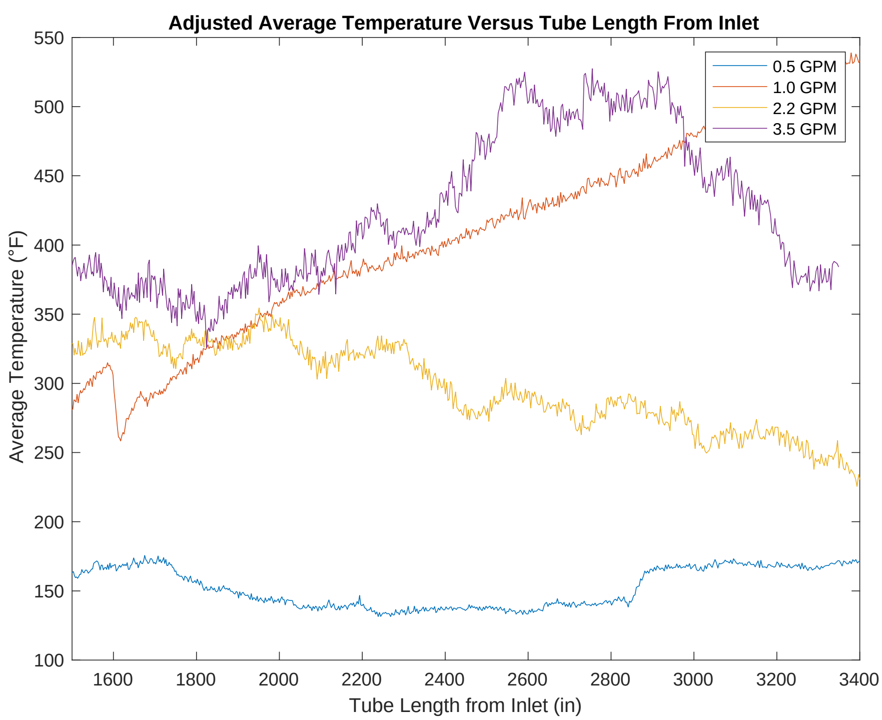

Figure 28: The Impact of Flow Rate on Adjusted Ground Loop Heat Transfer

From the ANSYS flow simulation using a 3 GPM flow rate, the horizontal ground loop design was predicted to achieve a 0.140 °C/ft temperature drop with an average heat transfer rate of 3980 W. While the test design resulted in a slightly smaller length of embedded tubing, the heat transfer achieved by the ground loop was lower than the ANSYS prediction even when accounting for this difference. From the ANSYS analysis, the team expected to achieve a heat transfer of 111 W/ft; however, the loop only managed to achieve a heat transfer of 27.7 W/ft when operated at the 1.0 GPM flow rate. Similarly, the ground loop section did not reach the ANSYS prediction for the temperature drop per foot and only achieved 0.0536 °C/ft. Overall the test section, when operated at the optimal flow rate of 1.0 GPM, attained 25% of the predicted heat transfer and 38% of the predicted temperature drop per unit length.

In addition to the data not supporting our ANSYS predictions, our results do not indicate a clear relationship between flow rate and heat transfer. The non-adjusted heat transfer values indicate that the ground loop achieves maximum heat transfer around the 1.0 GPM mark with flow rates below and above resulting in sub-optimal ground loop performance. One explanation for this behavior could be the tradeoff between longer residence time of lower flow rates and the greater mass transfer of higher flow rates. For example, the 0.5 GPM trial resulted in the greatest temperature drop as the working fluid had the longest residence time; however, the impact of the temperature drop was mitigated by the lower mass transfer. On the other hand, the 3.5 GPM trial had a temperature drop nearly four times smaller than the 0.5 GPM trial, but its heat transfer was slightly larger since the impact of the smaller temperature drop was moderated by the larger mass transfer. While Table X indicates that the 1.0 GPM flow rate optimizes the opposing effects of increasing mass transfer and decreasing the temperature drop, Figures 27 and 28 show that the 1.0 GPM trial also experienced increasing heat transfer over time, calling into question the validity of the trial. As mentioned previously, the non-constant heat transfer could indicate a component malfunction such as the flow meter gradually allowing the flow rate to increase without adjusting the reading. Another explanation for this increasing behavior could be that running the pump resulted in an input of work to the system, meaning that the longer the pump was operated, the greater the amount of heat that could be transferred. However, this explanation does not explain why the increasing behavior was not consistent across trials.

If both the 1.0 and 1.5 GPM trials are excluded and the adjusted heat transfer values are used, the relationship between flow rate and heat transfer appears to be directly proportional. From Table XI and Figure 28, heat transfer increases with flow rate and is optimized at 3.5 GPM (see Figure 29).

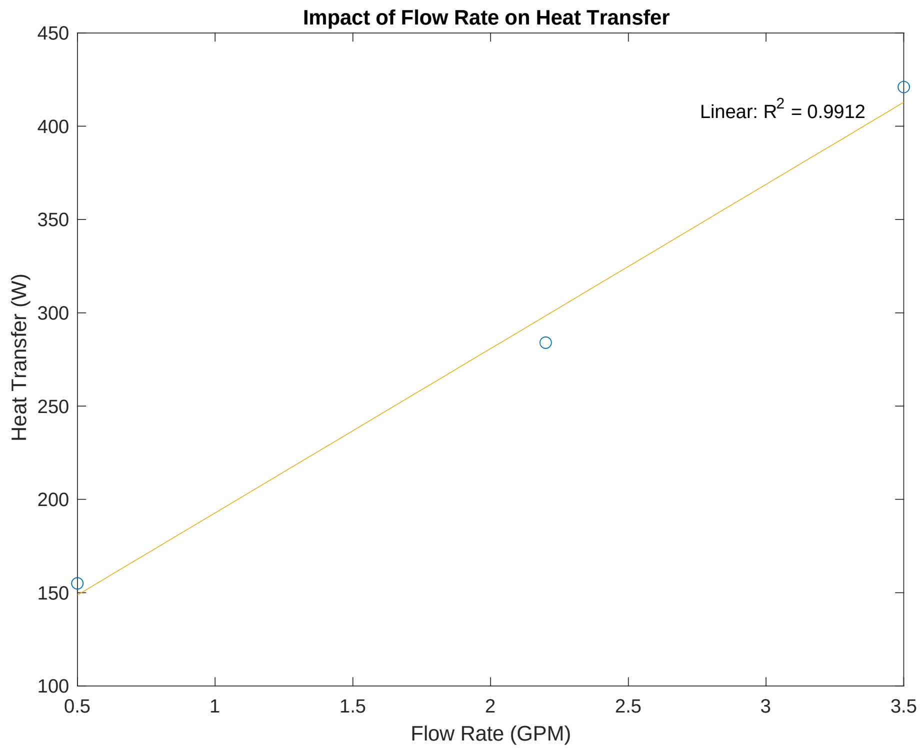

Figure 29: The Impact of Flow Rate on Heat Transfer Excluding the 1.0 and 1.5 GPM Trials

Excluding the 1.0 GPM trial and using the adjusted heat transfer values, the ground loop performance falls even further behind the ANSYS predicted with heat transfer and temperature drop only 16.7% and 11.8% of the predicted values, respectively. It should be noted that the ANSYS predictions were performed using a 3.0 GPM flow rate and the 3.5 GPM trial translates to 5.8 GPM, a flow rate nearly two times greater than that used to obtain these predicted values.

THE IMPACT OF FLOW RATE ON EXCESS GROUND TEMPERATURE

As GSHPs operate and use the earth to reject or absorb heat, they change the underground temperature of the neighboring soil which in turn changes dissolved oxygen content and can impact chemical reactions. While the temperature changes induced by ground loop operation are relatively small, since ground loops operate over long periods of time even these small changes can have long term impact on soil quality. Through the excess ground temperature measurements taken by three thermocouples embedded in the sand, the team wanted to determine the magnitude of the impact the embedded ground loop has on its surrounding medium.

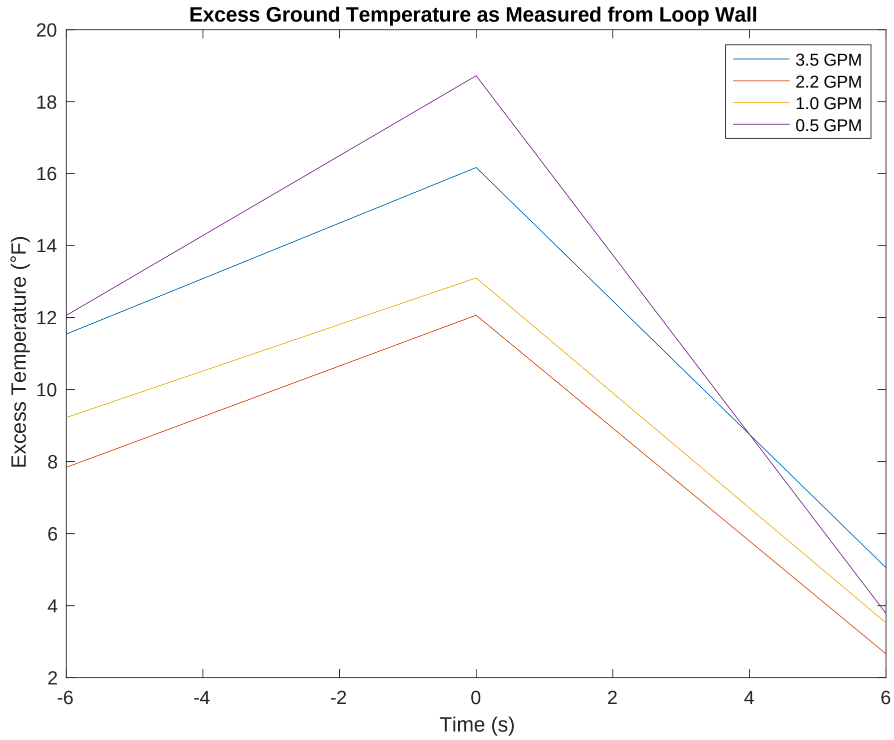

Figure 3030: Excess Ground Temperature as Measured from Loop Wall

The impact of ground loop operation on excess ground temperature in the center of the ground loop is significant. Figure 30 shows the temperature at the three ground loop locations for different flow rates at 2,500 seconds (6 in inside the loop, along the loop wall, and 6 inc outside the loop). From the figure, it is evident that all trials experienced a maximum temperature value at the loop wall, with all exceeding an excess temperature of 10°F. Additionally, Figure 30 highlights that the excess ground temperature located 6 in outside the loop is significantly smaller than that located 6 in inside the loop. The difference in excess temperature between the two ground locations indicates that the thermal interaction between the loop turns is significant. Most studies on GSHPs and soil properties indicate that no relevant changes of groundwater chemistry occur within a ±10.8°F excess ground temperature value (Casasso & Sethi, 2019). With excess temperatures at the center of the loop easily exceeding the bounds of the recommended 10.8°F interval, the ground loop design may have to be revisited to reduce the thermal interaction of the coils. In the next design iteration, this could involve increasing the center-to-center distance of the turns or by increasing the diameter of the loop turns.

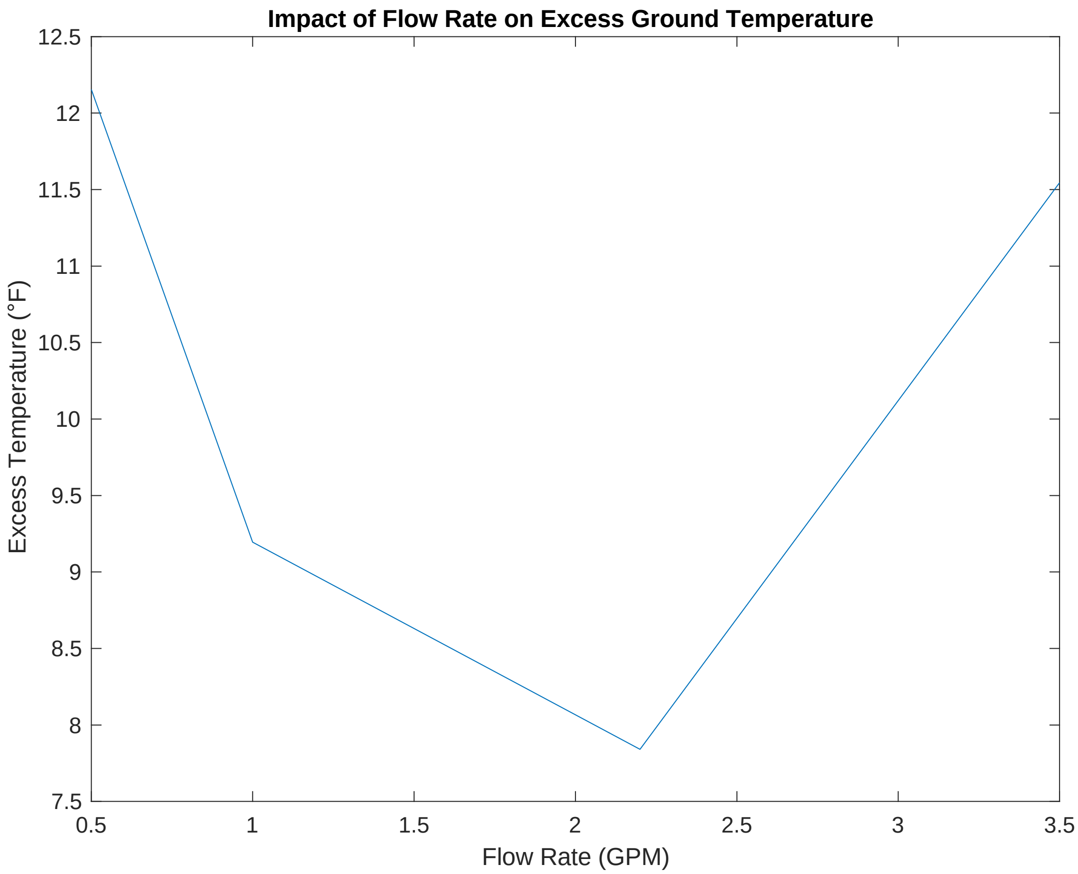

Figure 3131: The Impact of Flow Rate on Excess Ground Temperature Measured at 2500s and Ground 1 (Located 6 in Inside the Ground Loop)

Figure 31 depicts the relationship between the flow rate and the excess ground temperature taken at the ground 1 thermocouple, located 6 in inside the ground loop. As a ground loop transfers heat to the ground, the ground temperature will slowly rise therefore decreasing the difference between the working fluid and the ground and reducing system performance. Figure 31 indicates that heat sink degradation can be minimized by operating the ground loop at a flowrate between 1.0 and 2.2 GPM.

The Impact of Inlet Temperature on Excess Ground Temperature

The team also expected that as the inlet temperature increased the excess ground temperature at a given location from the loop wall would also increase. This hypothesis was proven true as evident in Figure 32, below as the 90°F inlet temperature exceeds both the 70°F and 80°F trials. Interestingly, the difference between excess temperatures of the 70°F and 80°F trials is not significant and could be a result of the inconsistent ground start temperature resulting from the Milton Airfield hanger not being temperature controlled.

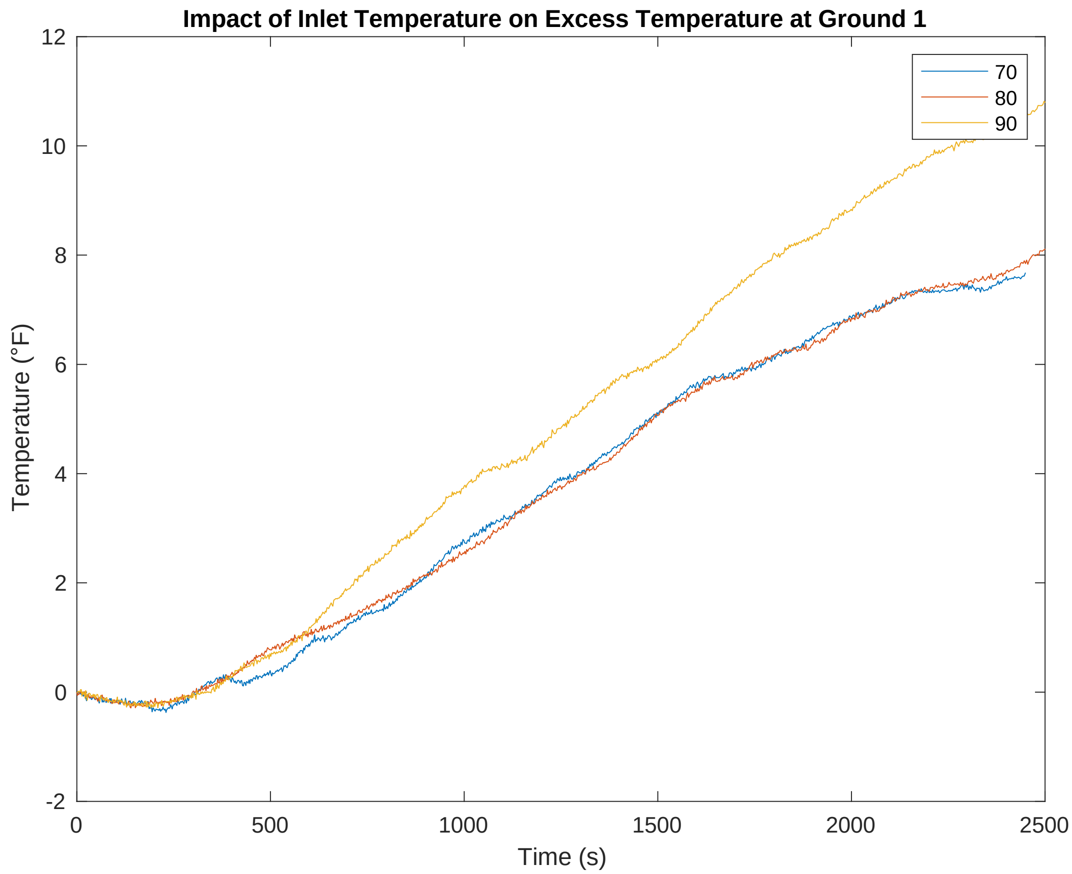

Figure 3232: The Impact of Inlet Temperature on Excess Temperature at Ground 1 (located 6 in inside the loop)

LIMITATIONS AND FUTURE WORK

LIMITATIONS OF DESIGN

Our data was limited through the lack of control over several important variables. These variables included the temperature of the tank, flow rate, and the ambient air temperature. While we were able to get meaningful data, aspects of the small-scale system could be improved to ensure greater data integrity. Greater control over variables such as working fluid flow rate and entering temperature would enable the data collected from the system to draw firmer conclusions on the impact of flow rate and temperature on the horizontal slinky's heat transfer capabilities.

The most glaring limitations was that the tank temperature slowly increased throughout each trial. The temperature of the water greatly increased after it went through our pump and this warmer water was circulated back into the tank. By the time the water had gone through the loop and was returned to the tank, it had not cooled back to the original temperature. With only a heating element in the tank we did not have any way to adjust for the increase. Eventually, the tank temperatures would approach 95°F, the maximum water temperature for the pump, and the trial would have to be ended. A more efficient pump may help to reduce some of this creep in temperature. A proportional integral derivative (PID) feedback system could help to maintain constant tank temperatures in the future. A feedback system would allow for the heating element to be modulated on and off to keep the temperature truly constant.

Flow rate was the one variable we had the hardest time controlling. Our constant flow rate pump would ramp up or down depending on how much we restricted flow to get the rate that we desired. Even when we had the bypass and precision flow meter installed, we were not confident that the flow meter was reading accurately for each trial. While we could get the flow rate to settle to a high or low flow, we could not set the flow rate at an exact value that we wanted to maintain. A higher quality pump may do a better job of pumping the water at a truly constant, measurable rate.

The final variable we could not control was the ambient air temperature. In a true geothermal system, the loop is placed deep enough so that the ground remains at a nearly constant temperature even if the outside temperature changes. For our small sandbox we saw that the temperature of water that had been sitting in the loop overnight was greatly affected by the ambient air temperature. This meant that on cold nights the loop water would get very cold. When we started up the pump the cold water would circulate into the warm water tank and cause the temperature to drop. This affected our startup temperature measurements and introduced a transient state that depended on the weather rather than the loop design itself. It would be helpful to insulate exposed piping and try to minimize the effect of ambient temperatures in the future.

More data points within the loop would have been helpful to identify how much heat transfer occurred. With only 3 data points within our loop, uncertainty analysis was not possible to perform. In the future, additional trials at each flow rate and tank temperatures would need to be done to ensure replicability of the data.

It is also important to note that we saw an excessive rise in ground temperature. Research shows that the ground temperature should not be increased by more than 10.8°F because of changes to ground chemistry. We measured nearly a 20°F increase in ground temperature at the center of our loop, but we believe that in the full-scale model there will be enough space and thermal mass that this will not be an issue.

Items to improve in future work:

1. Higher quality, more efficient pump
2. Constant Temperature tank through feedback control
3. More data points
4. Larger sandbox, or different ground material to minimize ambient air effects

FUTURE WORK

Moving forward, our design offers many opportunities for future work. Although we did not get the opportunity to build the full-scale ground loop, the tests we ran serve as a proof of concept and demonstrate that the loop design is effective. Even in our small-scale setup, there was measurable heat transfer throughout the loop. Our design compared several flow rates and entering water temperatures, but other aspects of the design, such as the loop geometry and working fluid type, can be evaluated to see how they affect the overall heat transfer. Since the reCOVER house is so unique, bespoke design could help to drive down the cost and size of installation. Results can be presented to UVA facilities to show that the proposed design will work and encourage them to approve the required permits.

Different ground loop configurations can easily be tested in the small-scale design setup. This will allow for the footprint of the ground loop at the reCOVER to be minimized. Industry relies on just two or three loop configurations due to their ease of installation and previous success, but perhaps another design is better suited for the conditions at Milton Airfield. This is a great opportunity to try new designs without the risk of it being for a client. Other than the geometry of the ground loop, piping materials can be easily compared. Using a high-density HDPE piping, as opposed to lower density extruded HDPE piping, may increase the heat transfer of the ground loop and allow for less piping to be placed in the ground. Even using other fluids or including glycol in the loop could provide valuable insights to the operation of the full-scale heat pump and convince UVA Facilities to allow for full scale construction.

After design has been completed, there are opportunities to test the performance of the heat pump within the reCOVER house. Flow control centers can be tested to optimize the use of the pump and only operate it when needed. This would be an interesting mechatronic design project to utilize a variable frequency drive (VFD) control and feedback loops. The transient portion of the loop provides the perfect opportunity to design these controls as the pump can be modulated on and off to prevent overshoot of desired temperatures. It could also be used to maximize the heat transfer within the loop depending on the amount of heat transfer that is needed.

On the scale of the entire house, the heat pump can be integrated with the solar panels that are currently being designed. This would allow for the pump to operate solely on power generated on site and not require any power connections from the utility company. Comfort controls of the house could be designed as well. This would include designing what air temperatures need to be supplied to the space, the amount of air supplied to the spaces, and controlling the quality of the air to the space.

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