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THERMAL PERFORMANCE ANALYSIS OF A SOLAR-ASSISTED HELICAL STEEL PILE THERMO-ACTIVE FOUNDATION WITH A DOUBLE U-LOOP HEAT EXCHANGER IN A COLD CLIMATE

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ABSTRACT

With the growing need to increase the share of renewable energy in the energy mix and limit the emission of greenhouse gases, ground source heat pumps (GSHPs) are emerging as an alternative option for space heating. They are the most efficient technology for space heating and cooling available today. However, with the high upfront costs, mainly due to the drilling and installation of ground heat exchangers (GHEs), some homeowners have decided against the use of these systems despite the low operating costs. To offset some of the drilling cost, a new option is to install the heat exchanger within the steel structural piles used in the new construction of homes, thus creating a "thermo-active foundation" for the building. The pile can also be used as an in-ground helical steel GHE in the case of existing buildings. With this relatively new area of research, only a limited number of configurations of the heat exchanger have been explored. This work explores the potential of a double u-loop GHE configuration coupled with a solar thermal collector to further enhance efficiency and long-term performance in a cold climate. The solution was implemented using a finite volume-based computational fluid dynamics tool considering realistic building loads, soil thermal properties, and climatic data for Minnesota's cold climate. The developed model was thoroughly validated with other experimental work using actual temperature and weather data, ensuring its accuracy. Additionally, the influence of several parameters on the heat pump's coefficient of performance was investigated. Results show that coupling the foundation heat exchanger with solar thermal increases the annual COP from 3.60 to 3.90 and 4.15 for the base case, 2 m^2 solar 1 case, and 3.8 m^2 solar 2 case respectively.

KEY WORDS: Cold Climate, Double U-loop Heat Exchanger, Ground Source Heat Pump Helical Steel Pile, Solar Thermal Collector

NOMENCLATURE

ρ	Density	(kg/m^2)	Z	Depth	(m)
c_p	Specific heat capacity	(kJ/kg·K)	t	Time	(s)
k	Thermal conductivity	(W/m·K)	Q	Heat exchange	(J)
α_s	Thermal diffusivity	(m^2/day)	I_T	Solar irradiance	(W/m^2)
T	Temperature	(K)	**Subscripts**		
T_s	Undisturbed soil temperature	(K)	f	fluid	

$T_{s,avg}$	Annual average soil temperature	(K)	i	inlet
$T_{s,amplitude,n}$	n^{th} order surface amplitude	(K)	o	outlet
PL	Phase angle	(-)	s	soil
t_p	Period of temperature cycle	(days)	G	ground
a, b, c	Constant in COP equations	(-)		

1. INTRODUCTION

Globally, buildings account for a third of the total energy consumption. Of that, 38% is used to meet the building's heating and cooling loads, equivalent to about 12% of the total energy consumption [1]. This is greater in cold climates. For example, in Norway, a cold climatic country, residential buildings account for around 70% of the total energy used due to space heating and domestic hot water [2]. The most common way to meet the building's heating and cooling loads is to use natural gas furnaces, electric furnaces, and central air conditioning systems. To reduce the carbon footprint of buildings, switching from these large carbon-producing forms of heating and cooling to highly efficient and sustainable renewable energy heating and cooling systems is desired. Ground source heat pumps (GSPHs) are highly efficient energy systems for heating and cooling buildings. Unlike air source heat pumps, which use ambient air as the source and sink of heat, GSHPs use the ground as a heat source during the longer cold weather season, and during the warm weather season, the ground will act as a heat sink. The stable ground temperature is the main reason GSHPs are more efficient than ASHPs.

GSHPs have two main components: the heat pump and the piping configuration inserted in the ground - called the ground heat exchanger (GHE). The GHE is the heart of a GSHP system, facilitating the transfer of thermal energy between the heat pump and the ground. There are different types of heat exchangers, and the most popular are horizontal and vertical systems. Horizontal systems are generally the most cost-effective for residential construction when sufficient land area is available [3]. The problem with horizontal systems is that they require a greater area than the conventional vertical systems. Conventional vertical systems are typically at depths between 60 m and 160 m [4]. Owing to the required borehole depth, the associated drilling cost leads to higher initial system costs [5]. Coupling heat exchangers in building foundations is emerging as an effective way of reducing the capital costs associated with GSHPs. "Thermo-active foundations" are a combination of a ground heat exchanger as well as the building's structural foundation [6] are becoming more popular. This is because foundation piles, such as steel piles, are already commonly installed as structural support for buildings in unconsolidated soil formations. The advantage of these thermo-active heat exchangers compared to conventional heat exchangers is the potentially low installation cost and heat storage capacity compared to the surrounding soil [7]. As such, several researchers are exploring the potential of thermo-active foundations for space heating and cooling.

A shallow 20 m single u-loop heat exchanger in a thermo-active energy pile has been investigated in a cold climate [8]. Results show that without any enhancement, the pile struggles to meet the building's energy loads. Nicholson et al. [6] modeled a helical pile with two offset plastic pipes disconnected in a large volume of water in a steel casing. This steel pile, which acts to provide both structural support for the building and works as a heat exchanger coupled with a heat pump, has a helical tip. It was modelled using computational fluid dynamics and finite element analysis, showing the percent difference in the outlet temperature with and without the helix tip is only 0.07% [6].

Coupling solar thermal energy with ground source heat pump systems has shown potential for performance improvement. Cimmino and Eslami-Nejad [9] developed an analytical model to simulate shallow solar-assisted double u-loop bore fields connected in series. It was found that a shallow system with a solar collector of 10 m^2 can decrease the length of the borehole by up to 29% [9]. This significant decrease in length compared to a conventional deep system can decrease the initial cost of the installation. In a study by Eslami-Nejad [10], a conventional deep system double u-loop heat exchanger with a ground heat pump was compared to the performance of a novel deep system solar-assisted heat exchanger of two independent loops, one loop to the heat pump and the other to the thermal solar collector. With the use of the two independent loops, the total length of the borehole was shortened by 17.6% compared to conventional double u-loops [10]. This shows the potential of solar thermal coupling in enhancing the performance of ground source heat pump systems.

The reviewed literature demonstrates the potential of solar-assisted heat pumps. However, there are limited studies on the solar enhancement of double u-loops in thermo-active foundations, especially in cold climates. Moreover, limited computational fluid dynamic studies which facilitate the coupling of building energy loads with the ground heat exchangers have been considered.

In this study, to further enhance the "thermo-active pile," instead of using a single u-loop heat exchanging pipe, two u-loop piles are selected. This study also investigates thermo-active double u-loop heat exchangers with the addition of a solar thermal collector. This study aims to numerically study the thermal performance of the solar-assisted thermo-active foundation pile double u-loop heat exchanger in Duluth, MN. This location was selected because of Duluth's high heating loads and lower cooling loads in the long winters and mild summers. In this study, actual weather conditions, soil properties, and realistic building loads specific to a cold climate are used as inputs and parametric studies are conducted to calculate the coefficient of performance (COP), a measure of a heat pump's performance and determine the maximum heating load a single "thermo-active pile" can support with a solar thermal collector.

2. Physical Configuration

The helical steel pile considered in this study, includes a hollow steel pipe with a welded screw tip used for driving the pile. Inside the pile, a double u-loop heat exchanger is used for the transfer of heat between the heat pump and the ground. The void inside the pile is backfilled with grout. The schematic including the heat pump, the pile and how it is coupled to a solar collector is shown in Figure 1.

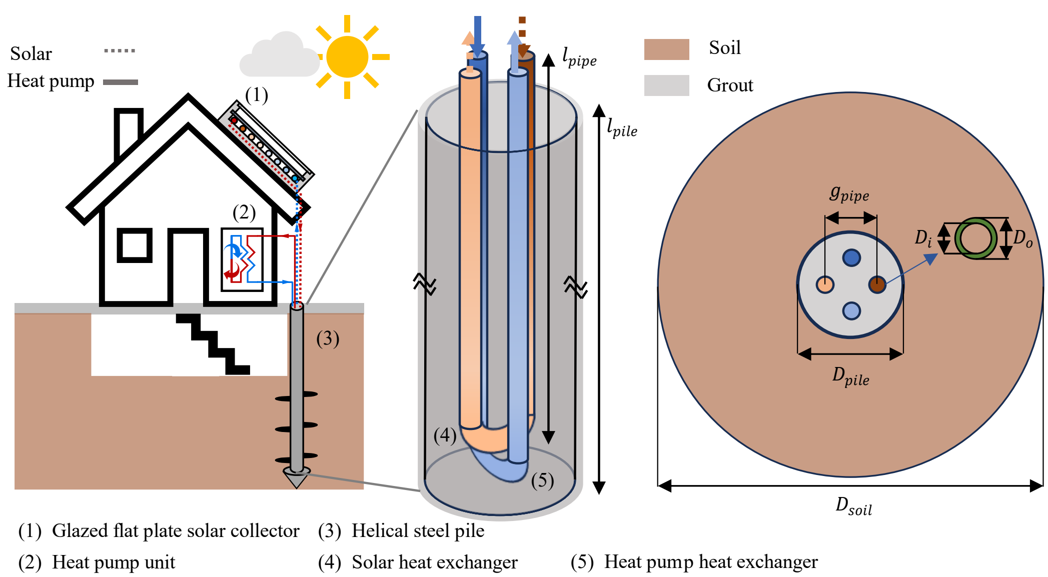

Figure 1: The schematic model of the proposed solar assisted GSHP system, the heat exchangers, and physical dimension parameters

The depth of the steel pile, l_{pile} is 20 m which is shallower than conventional boreholes. The steel pile, with a diameter, D_{pile}, 0.1397 m also acts as the structural foundation for new buildings. The pile is surrounded by the soil domain, with a diameter, D_{soil} of 10 m, which was obtained after a domain dependence test to ensure that far field boundary conditions are accurately modelled. The hollow pile is filled with two different materials. The first is two u-loop pipes of thickness, $(D_o - D_i)/2$ of 1.85 mm where D_i is 25.4 mm and made of high-density polyethylene (HDPE). The legs of the u-loop pipes are at a distance g_{pipe} of 70 mm from each other. The solar loop and the heat pump loop overlap perpendicularly as shown in Figure 1. Then, a backfill material in this case grout, that is used to fill the void space between the steel pile and the two u-loops. The working fluid flows within each of the two u-loops. The steel helical pile has a screw on the bottom, which is used for insertion into the ground. This screw was not modeled in the geometry as it has no significant impact on the heat transfer performance; not modeling the screw also decreases computational time significantly [6], [8]. Independent working fluids in each u-loops exchange heat with the heat pump, thermal solar collector, and the ground. Because of the possibility of freezing, the working fluid of the base case was selected as a 25% mixture of propylene glycol and 75% water.

Table 1 Material Properties

Material	Density (kg/m^2)	Specific Heat (J/kg·K)	Thermal Conductivity (W/m·K)	Viscosity (N·s/m^2)	References
Soil	1198.16	1588.31	1.67	-	[11]
Grout	2410	705	1.4	-	[12]
Steel Pile	7860	473	54	-	[13]
HDPE Piping	920	2174	0.4	-	[12]
Antifreeze fluid	1024.42	3903.3	0.46	0.0075	[14]
Water	998.2	4182	0.6	0.001003	[12]

3. Numerical Modeling

This study numerically solves the fluid flow in the pipe along with the conjugate heat transfer between the heat exchanger and the ground using the finite volume method (FVM) software ANSYS Fluent [14]. The software solves the conservation of mass and momentum within the fluid domains, and the conservation of energy within fluid and solid domains. Flow inside the u-loop pipes may be laminar or turbulent depending on the case and u-loops within the case. As such, the continuity, momentum, and energy equations for laminar flow and the modified Reynolds Averaged Navier Stokes equations for turbulent are solved.

3.1 Governing Equations

To solve the heat transfer problem in the soil domain, steel pile, backfill material, high-density polyethylene pipe, the conservation of energy in the solid domain is given as:

$$\rho_s C_{p,s} \frac{\partial T_s}{\partial t} = \nabla \cdot (k_s \nabla T_s) \quad (1)$$

In Equation (1), ρ_s is the density of the solid, $C_{p,s}$ is the specific heat of the solid, T_s temperature of the solid, t is time, and k_s is the thermal conductivity of the solid. For the laminar fluid domains, the continuity, momentum, energy, are solved:

Continuity:

$$0 = \rho_f \nabla \cdot \boldsymbol{u} \quad (2)$$

Momentum:

$$\frac{\partial \boldsymbol{u}}{\partial t} + \boldsymbol{u}.\nabla\boldsymbol{u} = \frac{-\nabla P}{\rho_f} + \upsilon\nabla^2\boldsymbol{u} \tag{3}$$

Energy:

$$\rho_f C_{p,f}\frac{\partial T_f}{\partial t} + \rho_f C_{p,f}\boldsymbol{u}\cdot\nabla T_f = -\frac{\partial(\rho_f)}{\partial t} + \nabla\cdot\left(k_f\nabla T_f\right) \tag{4}$$

The previous equations are written for a flow in a laminar regime. Equations (2-4) are used in the no solar double u-loop base case and for the u-loop connected to the heat pump in the enhancement cases. In these equations $\boldsymbol{u}$ is the velocity vector, P is the pressure, T_f is the temperature of the fluid, k_f is the thermal conductivity of the fluid, ρ_f is the density of the fluid, and finally $C_{p,f}$ is the specific heat of the fluid. For the validation cases and the solar u-loop in the enhancement cases, the flow regime is turbulent. Because of the turbulent flow, modified time-averaged versions of the previous equation require less computational to solve the closure problem arising from averaging the Navier-Stokes equations [15]. The realizable k – ε turbulent flow model was selected, where k is the turbulent kinetic energy and ε is the turbulent dissipation rate [15]. The detailed description of the k – ε model is given in [15]. The turbulent viscosity is calculated using this equation:

$$\mu_t = \rho_f C_\mu \frac{k^2}{\varepsilon} \tag{5}$$

Where μ_t is the turbulent viscosity and C_μ is constant [15].

3.2 Initial and Boundary Conditions

Specifying the correct boundary conditions is key to accurate modeling of GSHPs. The boundary conditions corresponding to the realistic climatic conditions of Duluth, MN were considered to simulate the performance of the GSHP in such a location. The following double harmonic equation given by Xing [16] was used to model the time-varying temperature distribution of the soil.

$$T_s(z,t) = T_{s,avg} - \sum_{n=1}^{2} exp\left(-z\sqrt{\frac{n\pi}{\alpha_s\times t_p}}\right)T_{s,amp,n}\cos\left[\frac{2\pi n}{t_p}(t - PL_n) - z\sqrt{\frac{n\pi}{\alpha_s\times t_p}}\right] \tag{6}$$

where $T_s(z,t)$ is the undisturbed ground temperature of the depth, z (m), and t is the time of year starting from January 1st, in days. t_p is the period of soil temperature per cycle (365 days) and PL_n is the phase angle. $T_{s,avg}$ is the average annual soil temperature, in K, and $T_{s,amp,n}$ is the nth order surface amplitude, which is half of the difference between the maximum and minimum monthly temperatures in a year. The initial ground temperature, the temperature of the top wall of the soil domain and the pile, as well as the bottom wall of the soil domain, are governed by this equation. For the side wall boundary condition, a far-field adiabatic boundary condition was used as a 10 m diameter soil domain was found to be sufficiently large to ensure no thermal interference between the neighboring piles and no interference with the far field [13].

Based on the operation mode of the heat pump, which depends on the demand for heating or cooling, the interaction of the GHE with the heat pump changes. To consider this, Equation (7) is used during the heating mode, Equation (8) is used for the cooling mode, and Equation (9) is used to determine the COP given by the heat pump [17].

$$Q_{GHE}^{heating} = Q_{Building}\left(1 - \frac{1}{COP_{heating}}\right) \tag{7}$$

$$Q_{GHE}^{cooling} = Q_{Building}\left(1 + \frac{1}{COP_{cooling}}\right) \quad (8)$$

$$COP = aT_{G,o}^2 + bT_{G,o} + c \quad (9)$$

As the COP relates to the different heating and cooling loads, the coefficients change from a = -0.003, b = 0.056, c = 5.784 for cooling to a = -0.001, b = 0.133, c = 3.257 for heating. Where $T_{G,o}$ is the outlet temperature of the GHX.

Equations (7) and (8) allow us to determine the inlet temperature of the GHE based on the outlet temperature of the previous time step as:

$$T_{G,i}{}^{j+1} = T_{G,o}{}^{j} + \frac{Q_{GHE}}{\dot{m}_g c_{p,f}} \quad (10)$$

The inlet temperature of the independent solar u-loop is based on the thermal solar efficiency equation as:

$$\eta = \frac{\dot{m}_s C_{p,w}(T_{o,s} - T_{i,s})}{A_c I_T} = a_0 - a_1\frac{(T_{i,s} - T_{amb})}{I_T} - a_2\frac{(T_{i,s} - T_{amb})^2}{I_T} \quad (11)$$

Rearranging equation (11), the inlet temperature of the solar loop is calculated based on the outlet temperature of the previous time step:

$$T_{i,s}{}^{i+1} = T_{o,s}{}^{i} + \frac{I_T{}^i A_c}{\dot{m}_s C_{p,w}}\left[a_0 - a_1\frac{(T_{o,s}{}^i - T_{amb}{}^i)}{I_T{}^i} - a_2\frac{(T_{o,s}{}^i - T_{amb}{}^i)^2}{I_T{}^i}\right] \quad (12)$$

Where the first term in Equation (11) is the maximum efficiency, the second term is the loss factor, and the third term is the quadratic loss term. $T_{i,s}$ is the inlet temperature of the solar collector, which is the outlet temperature of the u-loop, T_{amb} is the ambient air temperature, and I_T is the solar irradiation. The inlet temperature of the u-loops is dependent on the solar collector equation (12) and the heating or cooling load of the heat pump equations (7) and (8). These equations are dependent on the local weather data, which is used as an input and contributes to the boundary conditions used, Figure 2(a) shows the solar radiation at an angle of 67.6 degrees for Duluth, MN. Figure 2(b) shows the building loads used in equation (10) [18]. Equations (7) – (12) are implemented using user defined functions to couple the building energy model with the computational fluid dynamics model of the helical steel pile.

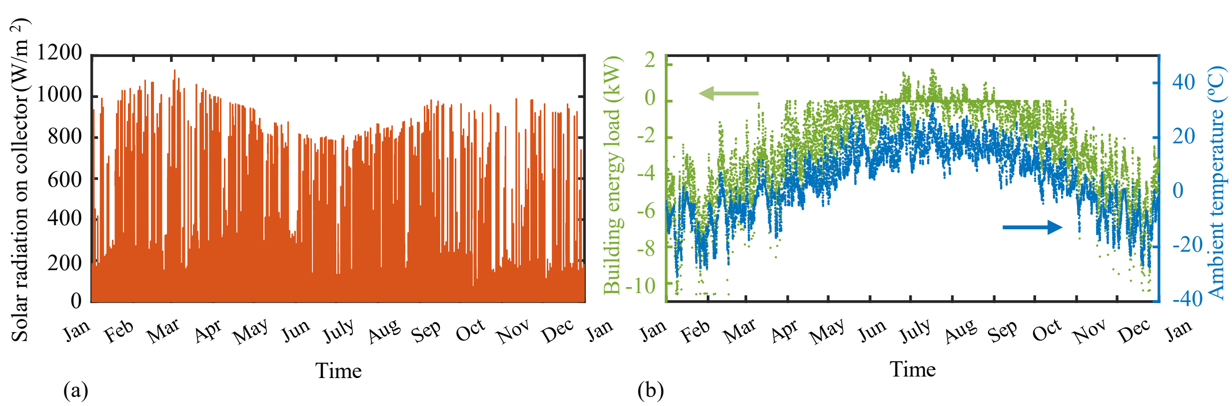

Figure 2: Solar irradiation, building load and ambient air temperature for one year in Duluth, MN

3.3 Solution Procedure

The numerical solution is implemented in ANSYS, the computational domain is modeled in Design Modeler, the meshing is completed in ANSYS Fluent meshing using polyhedral mesh elements to produce a high quality and the smallest mesh count thus creating a more desirable result and faster computational times [13],[19]. The selected mesh of 3.4 million cells is shown in Figure 3.

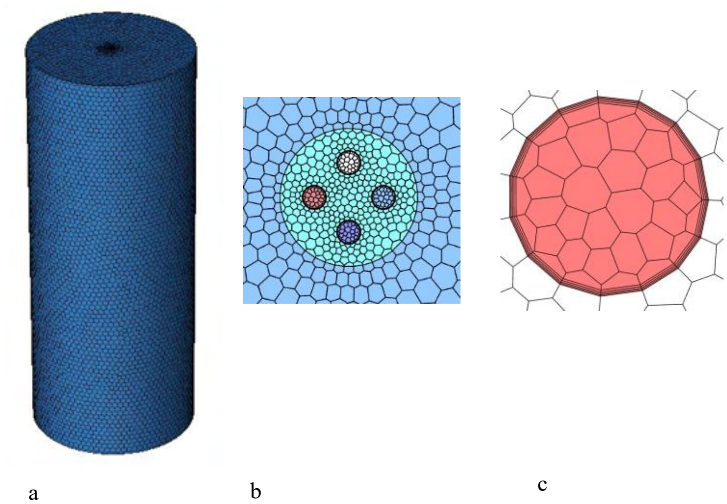

Figure 3: Computational mesh shown for (a) isometric view (b) top view of pile and two u-loops and (c) top view of one inlet.

The solution is obtained in ANSYS Fluent [19], with the Coupled algorithm for pressure velocity coupling, second-order upwind schemes were used for integrating the equations together with the boundary conditions. The solution is considered converged when the residuals for continuity are lower than 10^{-4}, lower than 10^{-5} for momentum and turbulence parameters, and lower than 10^{-7} for energy. Moreover, a detailed mesh dependency study was undertaken to ensure that the solution is independent of the grid element size. As shown in Figure 4, the Darcy friction factor and the temperature difference between the inlet and the outlet of u-loop 1 are depicted converging as the number of mesh cells increase.

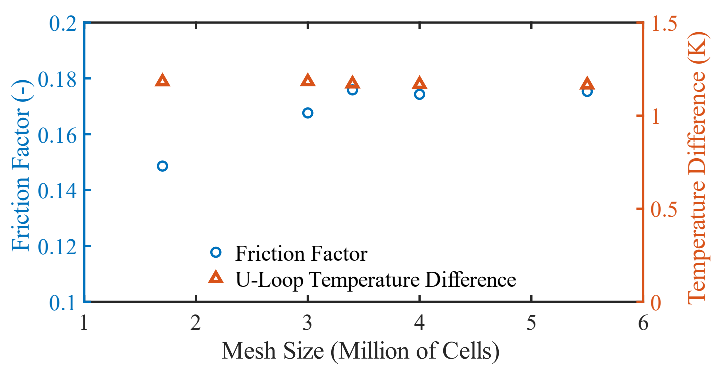

Figure 4: The Darcy friction factor and temperature difference of u-loop 1 shown for the mesh independence test.

4. Model Validation

The developed model was validated in a two-step process to ensure accurate results were obtained and to independently validate the two main components of the solar-assisted GSHP system. The two components validated are the GHE and the solar thermal collector.

The present study results were compared with experimental results by Shah et al. [12] under the same conditions. The first step was the independent validation of the model for double u-loop GHE. The ground heat exchanger, as described previously, consists of two inlets and two outlets. As shown in Figure 5, the present study results and the experimental temperatures are comparable in both the trend and values, showing that the model can predict the thermal performance of the GHEs in a satisfactory way.

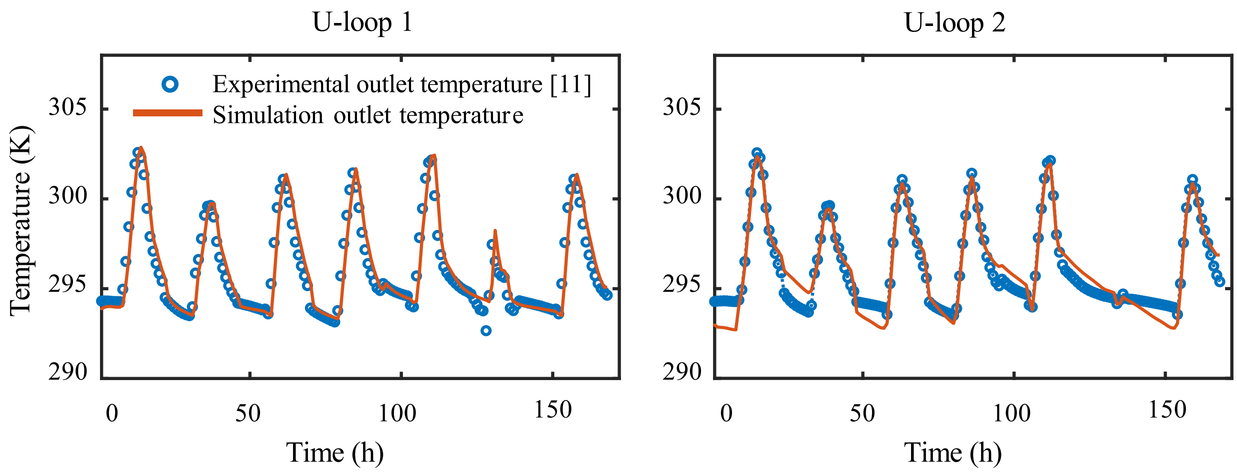

Figure 5: Experimental and simulated hourly outlet temperature of u-loop 1 and u-loop 2.

In the next step, the model of the solar thermal collector was coupled with the solar u-loop. As shown in Figure 6. Figure 6 shows that there is a matching trend between the average simulated outlet temperature and the average experimental outlet temperature. The simulated slope of the increase and decrease of temperature matches the experimental results and is reasonable, considering the fluctuations of the solar irradiation; this trend is what verifies that our methodology for modeling the solar-assisted GSHP can reasonably predict the performance of the whole system. The significant deviations seen at some time steps are likely due to the limited information on the solar thermal collector and its efficiency, experimental uncertainties, as well as heat losses in other fluidic components.

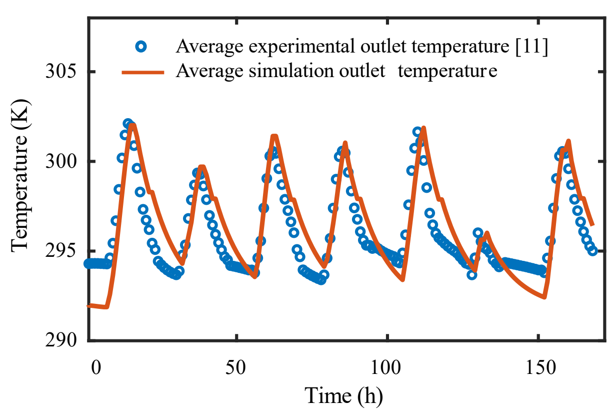

Figure 6: The experimental and the simulated average outlet temperature of the two u-loops with the use of the solar collector.

5. Results and Discussion

With the verified model, the performance of the helical steel pile coupled with a heat pump and realistic building energy loads was determined. The building heating and cooling loads for Duluth were calculated for a detached single-family house of 2000 ft^2 conditioned area using BEopt™, an hourly building energy modeling software that uses EnergyPlus™ as the calculating engine [20]. Obviously, due to the short length of the pile (20 m), a single pile is not capable of meeting the whole load, and it should be distributed between several thermo-active foundations connected in parallel [9]. As the base case, the design load of Duluth was normalized to 0.4 tons (1406 W) per energy pile as the initial case to examine how a double u-loop can handle the load. The outlets of the two u-loops are merged before entering the heat pump while the outlet of the heat pump is divided between the two u-loop inlets. Thus, the average outlet temperature of the u-loops is assigned as the inlet for the heat pump. The average outlet temperatures of the base case are shown in Figure 7.

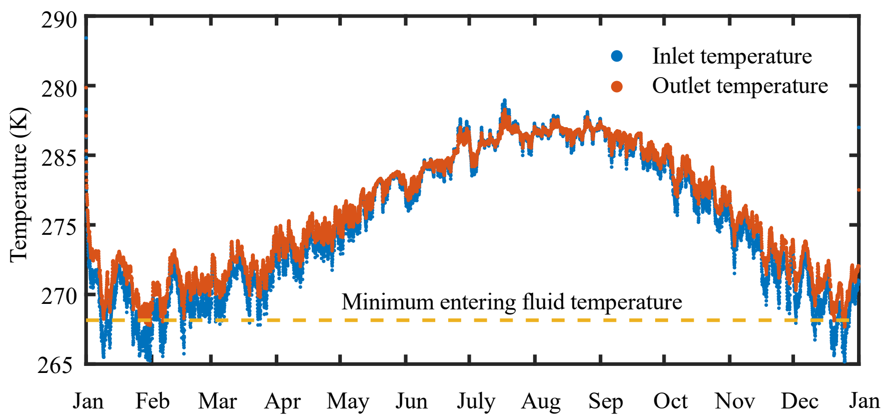

Figure 7: Inlet and outlet temperature of the double u-loops with a normalized 0.4-ton building load and no solar.

According to ASHRAE design guidelines [21], a flow rate of 3 gallons/minute per ton of heating is recommended. For the pile, at this flow rate, the velocity through the heat pump is 0.149 m/s for the base case. One of the limitations of the system is the heat pump's working fluid temperature range. The selected heat pump model is not able to operate with entering fluid temperatures below 268.15 K, which is shown with a dashed line in Figure 7. For these circumstances, the heat pump shuts down, and an auxiliary heating source is needed. Figure 7 shows that the pile with a 0.4-ton load is not able to handle the entire load for the full year as the outlet temperature of the u-loop is below the working temperature range of the heat pump for a few hours since the inlet temperature into the heat pump is significantly below this temperature limit. To improve performance and ensure that the pile can handle this load while meeting the heat pump's temperature requirements, a thermal solar collector is added to the system.

The enhanced cases consist of the solar-assisted GSHP system consisting of two independent u-loops, one going to the thermal solar collector and the one to the heat pump with a load of 0.4 tons, as depicted in Figure 1. The solar-assisted u-loop, named u-loop 1, has a working fluid of water. This is done for four reasons: (1) the selected solar collector is rated using water as the working fluid in the development of the solar collector's efficiency equation (2) the higher temperatures in the solar u-loop are expected to be above freezing temperature (3) water has a higher thermal conductivity and significantly lower viscosity which helps with higher heat transfer and lower pumping work (4) using antifreeze increases the capital cost. As recommended by the solar collector manufacturer the mass flowrate of water was 0.4 kg/s which was used in the model. The 2.00 m^2 area solar collector named Solar 1 case, the 3.80 m^2 are solar collector named Solar 2, and the base case with No Solar are depicted in Figure 8. As shown, the addition of solar energy increases the heat pump entering water temperature. The coefficients for the solar collector for Solar 1 are a_0 = 0.676, a_1 = 3.965, a_2 = 0.0114 and for Solar 2 a_0 = 0.718, a_1 = 2.290, a_2 = 0.044.

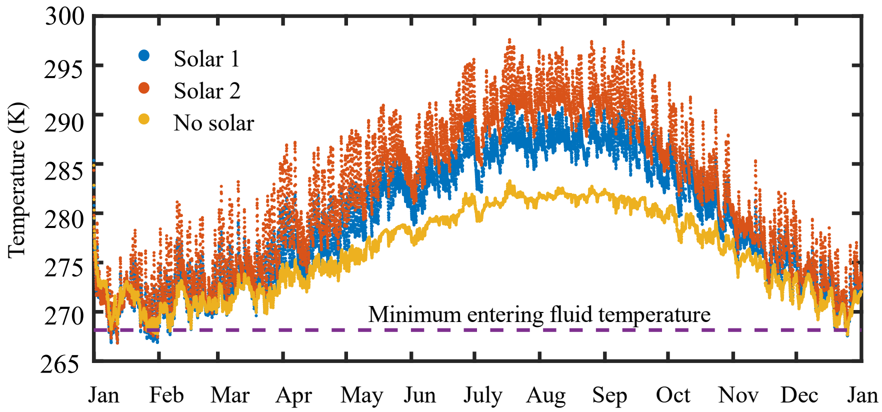

Figure 8: Outlet temperature of u-loop 2 going to the heat pump for the three cases.

The water temperature of u-loop 1, the loop that goes to the solar collector, for both Solar 1 case and the Solar 2 case are shown in Figure 9. In both cases the water temperature falls below water's freezing point of 273.15 K. This occurs 971 hours of the year for Solar 1 case and 670 hours for the Solar 2 case. With the frequent water freezing temperature of Duluth, it causes the temperature of the fluid to quickly fall below the freezing temperature when no solar irradiance is present. This is a problem to be investigated further as the increase in the solar collectors does lower the number of hours freezing thus the study of sizing is suggested to expand to larger area collectors. Since the temperatures are above the heat pump's entering water temperature, the water freezing also suggests that future simulations should incorporate an antifreeze mixture.

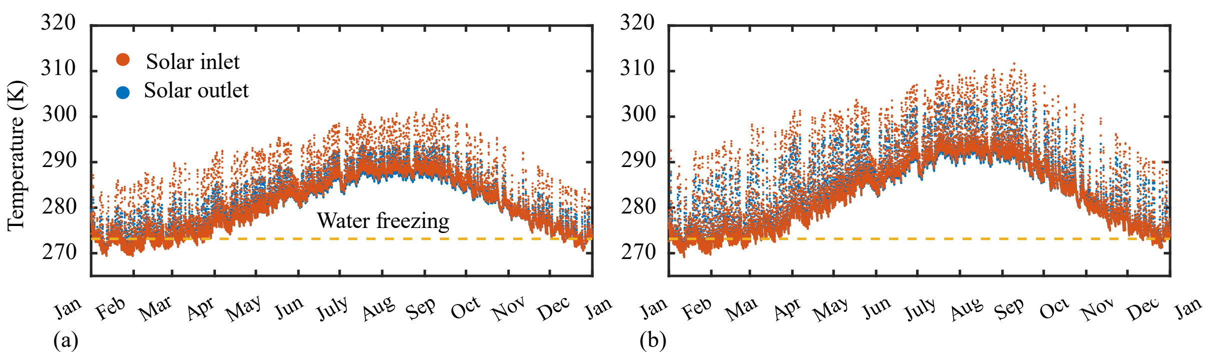

Figure 8: Temperature of inlet and outlet u-loop 1 of (a) the 2 m^2solar loop, and (b) 3.8 m^2 solar loop.

Comparing the COP is a way to determine the efficiency of the heat exchanger. Because of the significantly larger heating loads, the COP of the heating and cooling are evaluated separately, and then are averaged over the year. The outlet temperatures of the GHX are used to calculate the COP of the system using Equations (9).

The results from Table 2 show that the solar enhanced case noticeably increases the COP in the heating mode compared to the base case. The results also demonstrate that the larger area solar collector of the 3.8 m^2 in Solar 2 case has a greater COP in the heating mode than the 2.0 m^2 solar collector from Solar 1 Case. This trend is the opposite for the cooling mode as the base case has the greatest COP and then the greater the area of the solar collector the smaller the COP improvement. A possible explanation for this phenomenon is the increase in soil temperature due to the solar collector. However, because of the significantly larger heating loads of the system in cold climates, the larger heating COP has a greater impact on the annual average COP, thus it can be seen that the larger the solar collector the greater the total COP is for cold climates. For each case there are a few hours when the temperatures are below the recommended heat pump's entering water temperature causing shutdown, this occurs 0.37%, 1.16%, and 0.47% of the year, with respect to the increasing collector size. The implementation of solar collector charges the ground and increases the outlet temperature which in turn improves the $\boldsymbol{COP_{heating}}$ of the heat exchanger as the COP equation for the heating mode is an increasing function. This increases the $\boldsymbol{COP_{heating}}$ by 9.3% and 17.9% respectively. Contrarily, the COP equation for the cooling mode decreases with temperature, which explains why the $\boldsymbol{COP_{cooling}}$ degrades when solar collectors are in operation. This decreases the $\boldsymbol{COP_{cooling}}$ by 0.8% and 3.3% respectively. Overall, due to the heating dominated climate of Duluth, the annual COP improves when solar collector is incorporated.

Table 2 COP of the three cases

$\boldsymbol{COP}$**s**	**Base Case**	**Solar 1 Case**	**Solar 2 Case**
$\boldsymbol{COP_{heating}}$	3.43	3.75	4.03
$\boldsymbol{COP_{cooling}}$	6.04	5.99	5.84
$\boldsymbol{COP_{Ave}}$	3.60	3.90	4.15

7. Conclusion

In this study, a solar assisted GSHP model for a helical steel pile is first thoroughly validated in a two-step process, and then used to determine the long-term performance of a solar assisted thermo-active energy pile heat exchanger coupled with a GSHP for a typical 2,000 ft^2 home in Duluth, MN. Results show that the outlet temperature of the base case, solar 1 case, and solar 2 case remains above the heat pump limit with a load of 0.4 ton for most of the year. Our simulations show that the average heating COP of the system increases from 3.43 to 3.75 and 4.03 as the size of the solar collector area increases from 0 m^2 to 2.0 m^2 and 3.8 m^2, respectively. In addition, the cooling COP decrease from 6.04 to 5.99 to 5.84 as the size of the solar collector area increases from 0 m^2 to 2.0 m^2 and 3.8 m^2 respectively. However, because of the larger heating to cooling demand ratio in Minnesota, the total COP of the system is seen to increase from 3.6 to 3.9 to 4.15 as the solar area increases, this is an 8.3% and a 6.4% increase in COP from the base case. It also became clear that the working fluid of water for the solar collector needs to be a mixture of antifreeze as the temperatures reach below water's freezing temperature. These results reveal the advantages and shortcomings of the coupled system and can be used as a guideline for future studies on these novel systems.

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