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THE POTENTIAL AND LIMITATIONS OF USING GEOTHERMAL-SOURCED CHILLER PLANTS TO ELIMINATE COOLING TOWERS

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ABSTRACT

The purpose of this research is to evaluate the feasibility and performance of a vertical ground source refrigeration system for cooling a typical 7300 m2 office building after replacing cooling towers used for condenser cooling with three different designs of ground heat exchangers. To that end, a three-dimensional, transient, and conjugated finite volume model is developed and simulated to compare the thermo-hydraulic performance of the traditional single U-tube with that of double U-shaped, and spiral-shaped ground heat exchangers at different flow rates. Based on the results, spiral shaped ground heat exchangers outperform other designs, as seen by better heat exchange rates between the fluid and the soil, which translates to a greater temperature reduction of the cooling water. This improvement not only allows for using smaller number of boreholes which saves the construction costs compared to other designs, but it also improves the coefficient of performance of the system by significantly lowering the cooling water temperature flowing back to the condenser when compared to the conventional cooling tower. This approach also eliminates cooling tower water consumption (saves about 14,500 L/day), tower noise, annual maintenance expenses, and costs for periodical cooling tower replacement. The presented findings make a significant contribution to society by offering innovative and sustainable solutions for cost reduction, environmental conservation, and energy efficiency.

Keywords: Geothermal energy, Cooling towers, Ground heat exchanger, Potentials, Limitations, water saving.

NOMENCLATURE

BL	blowdown losses
C	cycle of concentration
COP	coefficient of performance
C_p	specific heat
DL	drift losses
E	thermal effectiveness
EL	evaporation losses
EWT	entrance water temperature
GHX	ground heat exchanger
GSR	ground source refrigeration
GPM	gallon per minute
HVAC	Heating, Ventilation, and Air Conditioning
k	thermal conductivity
LWT	leaving water temperature
L	liter
$\dot{m}$	mass flowrate
PEX	Crosslinked Polyethylene
P	pressure
Q	rate of heat transfer
T	temperature
TR	ton refrigeration
$\dot{V}$	volume flowrate
Greek symbols	
ρ	density
η	efficiency
Subscripts	
$cond$	condenser
$evap$	evaporator
e	electric
HX	heat exchanger
in	inlet
out	outlet
ref	reference
w	water

1. INTRODUCTION

Over the last few decades, new phrases such as climate change and harsh weather have been coined to characterize the state of the Earth's climate, which shows an increase in average temperature over time. Climate change is natural when it occurs over a period of centuries, allowing living things to adapt, but what is happening now is different. According to United Nations statistics, global warming has led the temperature to rise by 1.1 degrees Celsius since the late nineteenth century, causing the Earth to record the hottest temperatures between 2011 and 2020 [1]. This situation causes Harsh Weather as the climate shifts from mild to intense heat or freezing cold. As a result of this shift, Heating, Ventilation, and Air Conditioning (HVAC) systems have become almost essential for achieving comfort and coexisting with these challenging conditions. One of the key components of HVAC systems is the cooling tower which is a water-to-air heat exchanger used to provide cooling for condensers that have been heated during the refrigeration process and to discharge waste heat into the atmosphere. In water-cooled chillers, the heat of the condenser is removed through the water and then this water is sent to the cooling tower to be cooled and reused again. Despite the cooling tower's critical position in the HVAC system, it has several drawbacks that compel the need to find alternate solutions and other effective technologies to replace it. Some disadvantages of utilizing a cooling tower include its complexity, which raises maintenance and operating costs, power consumption because it uses mechanical pumps and fans to circulate water and air, significant water loss, sensitivity to climate change, noise, and dangerous environmental impact [2].

Because of its great energy efficiency and low environmental effect, the use of the ground as a heat sink has aroused the interest of the research community for cooling applications [3]. Shallow geothermal energy systems (<400 m depth) depend on the stable temperature (10-50 ℃) of soil mass, which normally exists 10-15 m below the earth surface [4]. Instead of using conventional air conditioning and heating systems, shallow geothermal energy systems can be a cost-effective solution for heating and cooling applications [5]. According to Michopoulos et al. [6], a ground source heat pump (GSHP) uses 25.7 % less energy than a traditional heating and cooling system and emits 22.7 % and 99.6 % less CO_2 and NO_x, respectively. However, high drilling and initial costs is the key challenge for wide spreading of these systems [7]. Therefore, the borehole number and depth have been shown to be the most influential parameters on the total cost of such systems [8].

Numerous recent studies have been conducted to develop and build innovative ground heat exchangers (GHXs) to reduce borehole number and depth, and hence drilling and set-up costs. Different designs were employed to dissipate or absorb heat from the ground including both horizontal [9] and vertical [10] designs. Vertical GHXs are buried at depths ranging from 15 to 150 m, whereas horizontal GHXs are buried in trenches ranging from 1 to 2 m deep [11]. Vertical ground source systems are probably the most popular and commonly utilized geothermal systems since they require less land footprint and have higher operational energy efficiency than other geothermal systems [7]. To reduce installation expenses, GHXs can be used on constructing pile foundations [12] or can be installed in the base soil layer or the concrete layer of a building foundation [13].

The current work introduces and assesses a new approach for cooling condensers in commercial applications by substituting cooling towers in air-conditioning systems with a geothermal source. As a result, the feasibility and performance of installing and operating a vertical ground source refrigeration (GSR) system for cooling a typical 7300 m^2 office building are explored. To that end, a 3D, transient, and conjugated finite volume model is developed and simulated to compare the thermo-hydraulic performance of the traditional single U-tube with that of double U-shaped, and spiral-shaped GHXs at different flow rates. The performance of the inspected system in terms of inlet and outlet cooling water temperatures, heat rejection, compressor power, and coefficient of performance (COP) are shown to highlight the proposed design's improvement in the condenser energy management compared to conventional cooling towers. The findings of this study show that using the ground as a heat sink to dissipate heat from condenser are a promising method for eliminating cumbersome cooling towers and enhancing HVAC systems performance. This approach also eliminates cooling tower water consumption, tower noise, annual HVAC system maintenance expenses, and capital costs for periodical cooling tower replacement. Therefore, the total cost and environmental impact of HVAC systems is expected to decrease.

2. PROPOSED SYSTEM ANALYSIS AND ITS SIMULATION MODEL.

The current work introduces and assesses a new approach for cooling condensers in commercial applications by substituting cooling towers in air-conditioning systems with a geothermal source. As a result, the feasibility and performance of installing and operating a vertical ground source refrigeration (GSR) system for cooling a typical 7300 m^2 office building are explored. A schematic diagram of the proposed GSR system considered in the current analysis is presented in Fig. 1. As shown in the figure the system consists of three main circuits. The first is a closed-loop pipe implanted inside the office building that transports heat via a heat-carrying fluid such as water with anti-freezing and transfers it to the refrigeration cycle. The second is the refrigeration cycle used to transfer the absorbed heat from the building to be dissipated to the ground instead of using a cooling tower. It mainly consists of an evaporator, a compressor, a condenser, and an expansion valve as can be seen in the figure. The third is a closed-loop ground heat exchanger (GHE) located inside the ground (the current study's focus). The stable temperature field of the earth serves as a heat sink, allowing the heat from the building to be rejected to the ground via the GHE circuit which allows for replacing the cumbersome cooling towers.

The vertical borehole ground loop heat exchanger is the most popular and widely used GHE configuration. Except for the fact that they are costly to install, they require a limited installation area, provide a long-term economic benefit, and

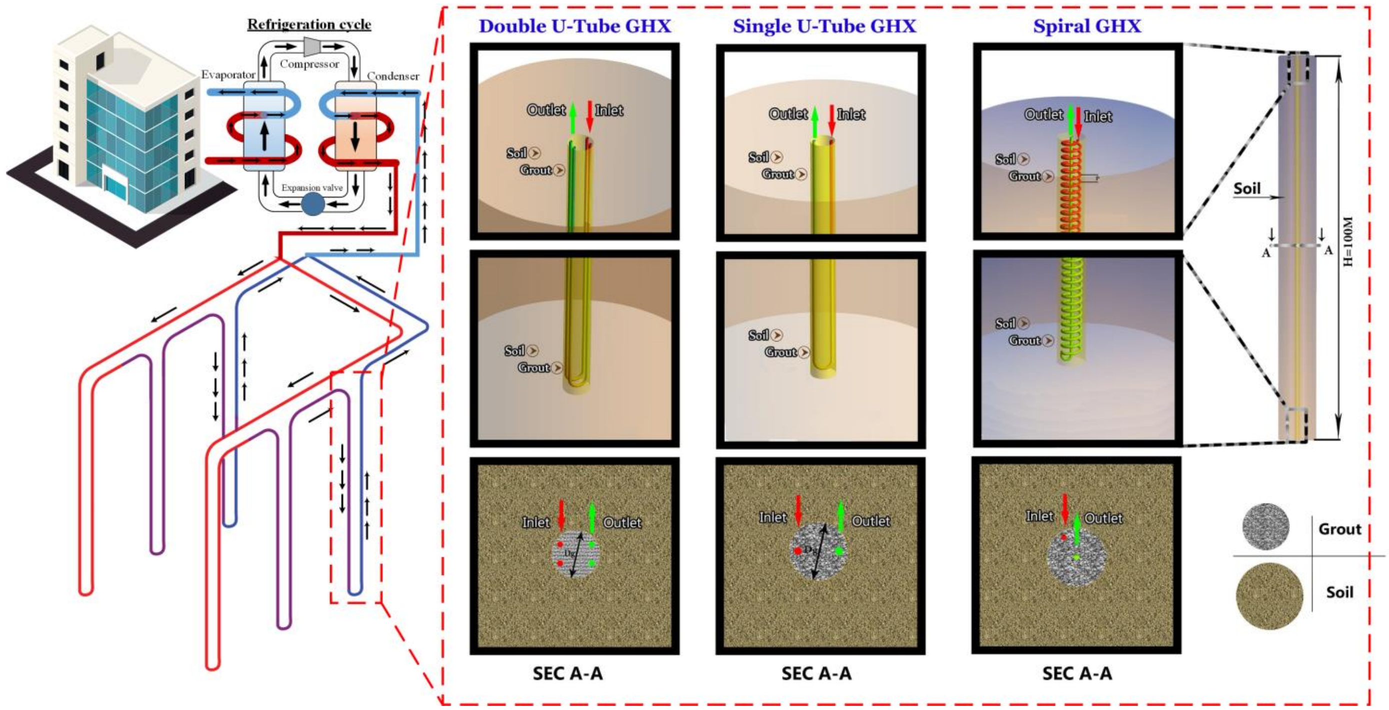

FIGURE 1: A schematic diagram of the proposed GSR system considered in the current analysis

yield improved thermal efficiency. Furthermore, they consist primarily of a vertical borehole filled with grout material and a closed-loop pipe system through the borehole with various U-bends at the bottom or with a spiral shape that permits downward and upward flow of heat-carrying fluid via the GHE. In the current study, three different designs of the GHE, including single U-shaped, double U-shaped, and spiral-shaped GHXs, are compared and analyzed at three different flowrates as shown in Fig. 1. All of the GHXs examined in this study were made from Crosslinked Polyethylene (PEX) tubes with inner and outer diameters of 118 mm and 156 mm, respectively [14]. PEX Tubes were chosen because they are suited for hot water applications up to 82.2 °C and can withstand 2.76 MPa hydrostatic design stress and 0.69 MPa pressure. The borehole was 100 m deep and filled with silica sand grouting material. The soil domain that surrounds the borehole has a radius of 10 m and is composed of sandy clay. Table 1 provides the dimensions of the proposed GHEs, including all parts implemented in the current study. Moreover, the thermophysical parameters of the materials employed in this study are listed in Table 2.

TABLE 1 Dimensions of the proposed GHEs.

Soil Diameter	10 m
Borehole depth	100 m
Spiral Turns spacing	0.2 m
U-tube leg spacing	0.5 m
Grout diameter	1.0 m
Pipe internal diameter	0.118 m
Pipe wall thickness	0.019 m

TABLE 2 Thermophysical properties of employed materials.

Property/Material Name	Soil	Grout	Pipe
Density, ρ (kg/m^3)	1960	2210	920
Specific Heat, C_p (J/kg.K)	1200	750	2300
Thermal Conductivity, k (W/m. K)	2.1	1.4	0.35

2.1 Theoretical analysis

Mathematical model: The purpose of this research is to evaluate the feasibility and performance of a vertical GSR system for cooling a typical 7300 m^2 office building after replacing cooling towers used for condenser cooling with three different designs of GHXs. To that end, a three-dimensional, transient, and conjugated finite volume model is developed and simulated to compare the thermo-hydraulic performance of the traditional single U-tube with that of double U-shaped, and spiral-shaped GHXs at different flow rates. To ensure the model's promptness and ease of use in generating results, the following assumptions are considered in the adopted model in a way that does not conflict with the quality and accuracy of the result: (1) The fluid in pipes is water which is considered as incompressible fluid. (2) In all domains, soil properties are regarded as homogeneous. (3) The thermophysical properties of all materials used are temperature independent. (4) The top surfaces of the module including grout and soil domain are considered as adiabatic surface. (5) The outer side wall and bottom surface of the soil domain are isothermal. (6) The water flow heat transfer either convection or advection are neglected. The work conducted by Serageldin et al. [15] was used to validate the current developed model by applying the same system geometry, boundary conditions, and material properties were used. Thus, the governing equations for the introduced system can be expressed as those in [15].

Boundary conditions: The thermal boundary conditions for all surfaces of the proposed system are carefully defined to solve the heat transfer governing equations. At time t = 0, the whole domain is initially set to 17.7 °C equal to the soil temperature. Thermally coupled boundary condition with no temperature jump is applied at all solid-solid interfaces. Besides, no-slip boundary condition is applied at liquid-solid interfaces existing between cooling water and the pipe walls. The adiabatic boundary condition is applied at the top surface of the module including grout and soil domain as previously clarified. Furthermore, the soil domain's outer and bottom walls are kept at a constant temperature of 17.7 °C. Under laminar and turbulent flow conditions, three different flow rates were selected for study. Laminar flow occurred at a flow rate of 12.7 L/min, while turbulent flow occurred at rates of 17.0 L/min and 34.7 L/min. Finally, inlet water temperature was kept at 34.6 °C (307.75 K) for all cases to mimic the temperature of cooling water leaves the condenser. Meanwhile, the outlet pressure is defined to be equal to the atmospheric pressure. Table 3 summarizes all the boundary conditions adopted in the current analysis.

TABLE 3 Boundary Conditions.

	Flowrate L/min	Velocity m/s	Pressure $Pa_{(g)}$	Temperature/ Heat flux
Inlet	12.79	0.0195	-	34.6 °C
	17.05	0.0260	-	34.6 °C
	34.75	0.0560	-	34.6 °C
Outlet	-	-	0	-
Soil Top surface	-	-	-	Adiabatic
Grout top surface				Adiabatic
Soil outer wall	-	-	-	17.7 °C
Soil Lower surface	-	-	-	17.7 °C

Data reduction: Even though all refrigeration units operate for less than 24 hr, interspersed with rest periods when refrigeration is not required, all parameters were examined in 24 hr, 48 hr, and 72 hr continuous operation because the heat exchange rate becomes nearly steady after 72 hr, allowing for relative evaluation of the GHXs. The temporal thermo-hydraulic performance of single U-shaped, double U-shaped, and spiral-shaped GHXs was numerically compared at different operating scenarios. The cooling water temperature difference (ΔT), the thermal effectiveness (E), temperature contours, temperature profile, pressure drop (ΔP), heat transfer rate (Q) and COP improvement were determined using the following equations.

The heat exchange rate (Q_{HX}) is determined using the fluid inlet and outlet temperatures, as well as the mass flow rate and fluid heat capacity, using the following equation.

$$Q_{HX} = \dot{m}_w \, C_{p,w} \, (T_{w,in} - T_{w,out}) \qquad (1)$$

where $\dot{m}_w$ and $C_{p,w}$ are the water mass flowrate (kg/s) and specific heat of water (J/kg. K), respectively. $(T_{w,in} - T_{w,out})$ is the inlet-outlet water temperature difference.

The effective coefficient of thermal energy (E) is the ratio of the actual heat transfer to the maximal amount of heat transfer that can occur. Furthermore, it is a dimensionless number that ranges between zero and one, and it demonstrates the GHX's efficacy in delivering the maximum outlet temperature. It is calculated using the following equation [16]:

$$\boldsymbol{E} = \frac{T_{W,out} - T_{W,in}}{T_{ground} - T_{W,in}} \qquad (2)$$

where $T_{W,in}$ is the fluid inlet temperature (34.6 °C), and T_{ground} is the undisturbed ground temperature (17.7 °C).

The electric pumping power required for moving the fluid inside the heat exchanger to overcome the friction in pipes can be calculated through the following equation:

$$\boldsymbol{P_e} = \Delta P \cdot \dot{V} / \eta_{pump} \qquad (3)$$

where ΔP and $\dot{V}$ are the pressure drop across the heat exchanger in (Pa) and the water volume flowrate in (m^3/s), respectively. η_{pump} is the overall pump efficiency, including the hydraulic, mechanical, and motor efficiencies which is assumed to be equal to 0.42 [17].

Case study: According to [18], the total heat of a general office ranges between 18.6-27.9 m^2/Ton of Refrigeration (TR), which means that every 27.9 m^2 of gross office area can be air-conditioned with a cooling load of 1 TR [18]. As a result, for the current analysis, a medium-scale office building with a gross area of 7300 m^2 can have a cooling load of around 262 TR. According to Carrier Global Corporation Products, the AquaForce® Water-Cooled Liquid Screw Chillers [19] with a model nomenclature of 30XW275 can handle this load. The specifications of which are provided in the table 4 below.

Table 4 Water cooled 30XW275 Chiller specifications.

30xw Unit capacity	262.9 TR / 924.4 kW
Input Power	167.9 kW
Evaporator Flow	2382 L/min
Condenser Flow	3042 L/min
Full Load Efficiency	0.6384 (kW/Ton)

As shown in the table, the condenser requires a water flowrate of 3042 L/min to allow the refrigerant vapor gets condensed and converted to a liquid state by releasing the latent heat to the flowing water. Furthermore, the manufacturer specifies a minimum allowed temperature difference of 5.2 °C of the water passing through the condenser with an entrance water temperature (EWT) of 29.4 °C and a leaving condenser water temperature (LWT) of 34.6 °C for stable and efficient operation of the refrigeration cycle [5]. In conventional systems, water travels directly to the cooling tower after leaving the condenser at 34.6 °C to reject heat to the atmosphere. Using Equation (1), the overall heat rejection rate for the proposed case is calculated to be 1100 kW (313 TR).

The cooling tower loses water in three ways: evaporation, which is the primary technique used to reject heat, blow down, which is intentional water replacement to keep the solid and contaminant concentrations under acceptable levels, and drift, which is unintentional water droplet loss carried by the induced air through the cooling tower. The water loss by the cooling tower can be calculated using the following equations [18]:

$$Evaporation\ losses, \boldsymbol{EL}\ (gpm) = GPM_{cond} * \Delta T_w * 0.0008 \quad (4)$$

$$Drift\ losses, \boldsymbol{DL}\ (gpm) = GPM_{cond} * 0.0002 \quad (5)$$

$$Blowdown\ losses, \boldsymbol{BL}\ (gpm) = \frac{EL - [(C-1) * DL]}{(C-1)} \quad (6)$$

where GPM_{cond} is the condenser water flow rate in gpm, ΔT_w is the difference between LWT and EWT in °F, and *C* is the cycle of concentration (3-5 cycles), will be considered 4 cycles in this analysis.

After applying the above equations to the proposed case, the Evaporative losses (*EL*) equal 5.98 gpm, Drift losses (*DL*) equals 0.16 gpm and Blowdown losses (*BL*) equals 1.83 gpm. The make-up water of the cooling tower is the sum of the three losses which is 7.97 gpm. The cooling tower water demand will be over 14,500 L/day for 8 hours of operation per day, which is a massive waste of water that might be increased if the system running hours are expanded. Therefore, replacing the cooling towers with GHXs is innovative solution to eliminate cooling towers' water consumption, tower noise, annual system maintenance expenses, and capital costs for periodical cooling tower replacement.

The proposed system is also predicted to achieve a temperature reduction equal to or greater than 5.2, which will not only alleviate the adverse effects of cooling towers but will also increase the system coefficient of performance (COP). As a result, one more parameter will be utilized to evaluate the effect of replacing the cooling tower with the GHX, which is known as COP improvement (ΔCOP) and can be calculated using the equation below.

$$\Delta COP = \frac{COP_{carnot,GHX} - COP_{carnot,cooling\ tower}}{COP_{carnot,cooling\ tower}} \times 100 \quad (7)$$

where $COP_{carnot,GHX}$ and $COP_{carnot,cooling\ tower}$ are the Carnot coefficients of performance of the chiller using the GHX, and that using the cooling tower, respectively. The Carnot coefficient of performance law is [20]:

$$COP_{Carnot,Ref} = \frac{T_{Evap,ref}}{T_{Cond,ref} - T_{Evap,ref}} \quad (8)$$

$$T_{Evap,ref} = T_{chilled,out} - 1.67 \quad (9)$$

$$T_{Cond,ref} = LWT + 2.22 \quad (10)$$

where $T_{Evap,ref}$ and $T_{Evap,ref}$ are the saturation temperature of the refrigerant in the evaporator and condenser in (K), respectively. $T_{chilled,out}$ is the chilled water temperature leaving the evaporator (equal to 6.2 ℃ based on the manufacture datasheet [19]). LWT is temperature of water leaving the condenser, which equals to cooling water temperature from the cooling tower or, GHX plus 5.2 ℃. In the current analysis, for the chiller using cooling tower, the LWT = 307.75 K and $T_{chilled,out}$ = 279.35 K. Thus, the $COP_{carnot,cooling\ tower}$ is calculated using Equation (8) to be 8.75.

2.3 Computational procedure and validation

The full transient, and conjugated finite volume model for implemented in the current investigation was developed and simulated using ANSYS software based on the finite volume technique. To numerically solve the model, the following four stages were implemented. First, using the ANSYS design modular tool, a 3D geometry of the computational domain consisting of the conjugated heat and fluid flow through the heat exchanger and the surrounding grouting and soil materials. Second, using the ANSYS meshing tool, a high-quality grid was produced inside each block. The grid size is chosen after careful analysis of the data to achieve grid independence and satisfy both solution accuracy and convergence at a low run time. Third, for the optimal number of grids, the governing partial differential equations combined with boundary conditions were numerically solved using the ANSYS Fluent 20.1 simulation tool. The model solution is continued until reaching a certain residual of 10^{-9} in the energy equation, 10^{-3} for continuity equation and 10^{-5} for other parameters. Finally, after the convergence, the temperature distribution all over the computational domain, the cooling water temperature difference (ΔT), the thermal effectiveness (E), temperature contours, temperature profile, pressure drop (ΔP), and heat transfer rate (Q) were determined and analyzed.

The SIMPLE segregated method was utilized in this study to solve the pressure–velocity coupling within the fluid domain. The pressure, momentum, energy, turbulent kinetic energy (K-equation), and dissipation rate (∈-equation) equations were discretized using a second-order technique. To model the transient behavior of the flowing fluid, a first-order implicit scheme was adopted.

Model validation: Initial simulations were carried out to validate the proposed model by comparing the predicted results to available data in the literature using the same system geometry, boundary conditions, and material properties. Before proceeding, such comparisons thoroughly assess the model's accuracy and reliability. Tables 5 and 6 compare the transient simulation results (outlet water temperature (T_{out}) and the heat exchange rate (Q_{net})) of the typical single U-tube GHX with the corresponding numerical results of Serageldin et al. [15]. A 72-hour transient simulation was performed for a single U-tube GHX with a depth of 20 m and inner pipe diameter of 0.026 m at different mass flowrates of 2 L/min and 8 L/min, corresponding to laminar and turbulent flow regimes. Comparisons of the output fluid temperature shows good agreement, with a maximum error value of 0.29 % and 0.08 % for laminar and turbulent flow situations, respectively. Furthermore, the highest error percentage of the heat exchange rate was 3.51 % for laminar flow and 2.44 % for turbulent flow conditions.

TABLE 5 Comparison of current predicted fluid outlet temperature with the numerical results indicated by [15] for single U-tube GHX.

	Outlet temperature, T_{out} (℃)					
	2 L/min			8 L/min		
	Current	Ref. [15]	Error	Current	Ref. [15]	Error
24 hr	24.1	24.15	0.2%	26.16	26.14	0.08%
48 hr	24.28	24.35	0.29%	26.23	26.21	0.08%
72 hr	24.38	24.45	0.29%	26.26	26.25	0.04%

TABLE 6 Comparison of current predicted heat exchange rate with the numerical results indicated by [15] for single U-tube GHX.

	Heat exchange rate, Q_{net} (W)					
	2 L/min			8 L/min		
	Current	Ref. [15]	Error	Current	Ref. [15]	Error
24 hr	407.79	396.5	2.8%	468.06	478.8	2.24%
48 hr	381.31	369.1	3.3%	427.74	438.4	2.43%
72 hr	367.11	354.6	3.51%	407.4	417.6	2.44%

3. RESULTS AND DISCUSSION

The current study introduces and evaluates the feasibility of a new approach for cooling condensers by replacing cooling towers in air-conditioning systems with a geothermal source. The performance of three different ground heat exchanger (GHX) designs, including single U-shaped, double U-shaped, and spiral-shaped GHXs, was compared and analyzed at three different flowrates. These comparisons propose the best design that reduces conventional construction costs, simplifies installation, enhances heat transfer, and reduces thermal resistance to efficiently dissipate heat from condenser to the ground. The inspected system's performance in terms of inlet and outlet cooling water temperatures, thermal effectiveness, heat rejection, pumping power, and total number of bore holes required are shown to highlight the proposed design's improvement in condenser energy management over conventional cooling towers.

3.1 GHXs performance comparisons.

The temporal thermo-hydraulic performance of single U-shaped, double U-shaped, and spiral-shaped GHXs was numerically compared at different operating scenarios. The cooling water temperature difference (ΔT), the thermal effectiveness (E), temperature contours, temperature profile, pressure drop (ΔP), and heat transfer rate (Q) were determined and analyzed.

Fluid temperature reduction Figure 2 depicts the temperature reduction of cooling water achieved while flowing through single U-shaped, double U-shaped, and spiral shaped GHXs at mass flow rates of 12.7 L/min, 17.0 L/min, and 34.7 L/min, corresponding to both laminar and turbulent flow regimes. Furthermore, the figure compares thermal effectiveness for all analyzed cases, which reflects the ratio of real to theoretical heat transfer capacity and demonstrates heat transfer efficiency. Despite the fact that all refrigeration units operate for

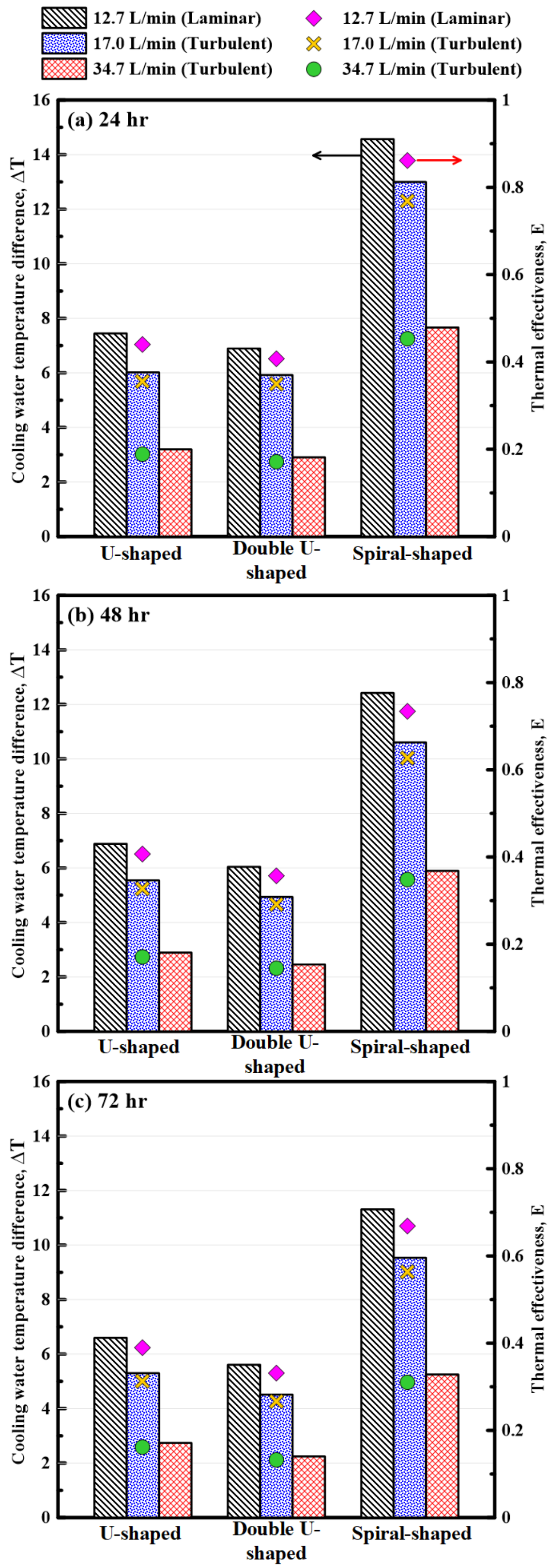

FIGURE 2: Comparisons of cooling water temperature difference and thermal effectiveness of the investigated GHXs.

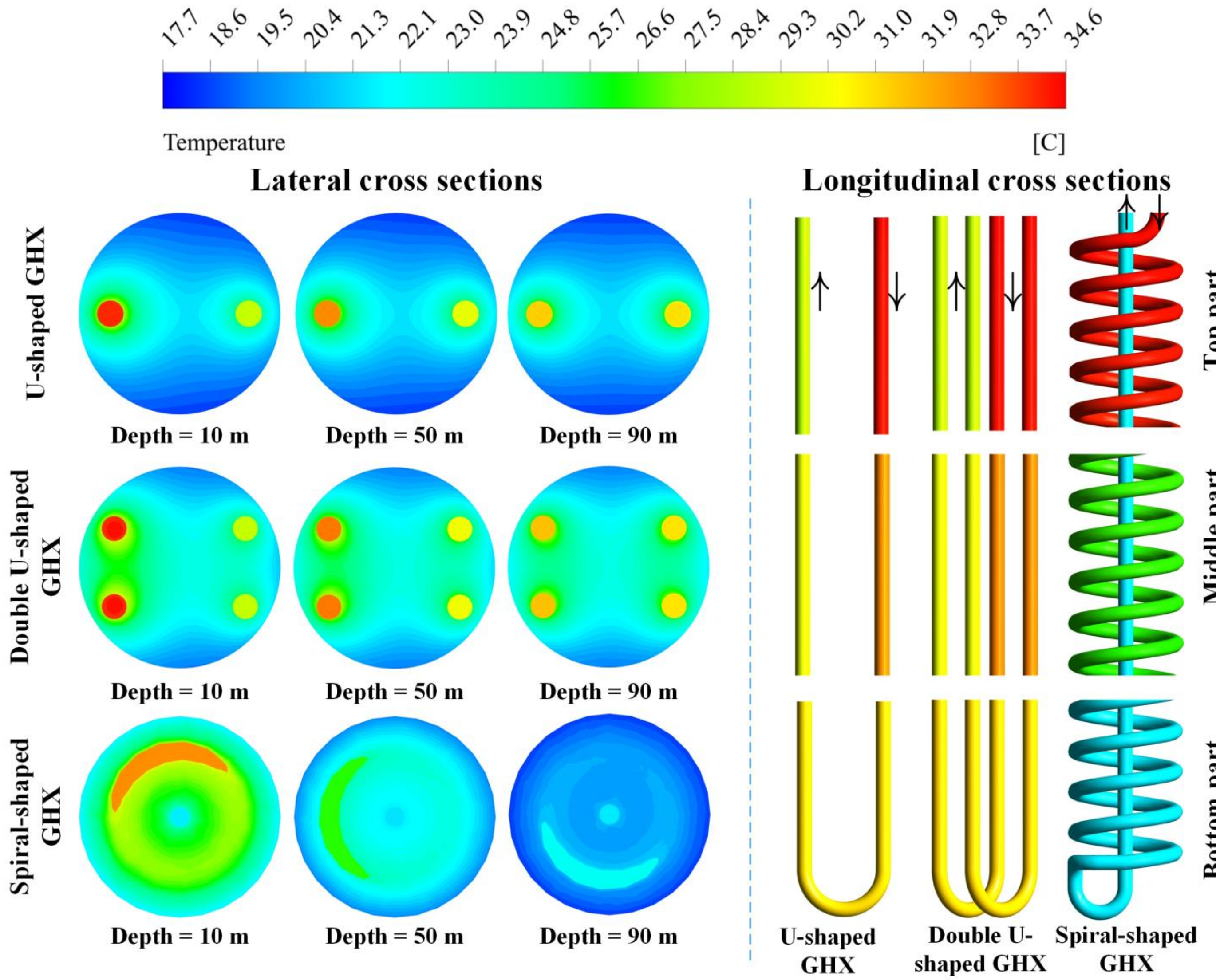

FIGURE 3: Temperature contours in the lateral cross sections at depths of 10 m, 50 m, and 90 m and longitudinal cross sections at top, middle and bottom parts of the investigated GHXs.

less than 24 hr, interspersed with rest periods when refrigeration is not required, all parameters were examined in 24 hr, 48 hr, and 72 hr continuous operation because the heat exchange rate becomes nearly steady after 72 hr, allowing for relative evaluation of the GHXs. As can be seen in the figure, water flowing through the spiral shaped GHX attained the highest temperature reduction compared to other designs operating at the same mass flow rate. This higher reduction indicates improved heat transfer from the cooling fluid to the surrounding soil as a result of the coil and turns in such design. For instant, after 24 hr of operation, employing the single U-shaped, double U-shaped, and spiral-shaped GHXs reduced the cooling water temperature with a 12.7 L/min mass flow rate by about 7.4 ℃, 6.9 ℃, and 14.57 ℃, respectively. Increasing the water mass flow rate from 12.7 L/min (laminar) to 34.7 L/min (turbulent) decreased the attained temperature reduction to 3.2 ℃, 2.9 ℃, and 7.66 ℃, respectively, for single U-shaped, double U-shaped, and spiral-shaped GHXs after 24 hr.

The single U-shaped and double U-shaped GHXs exhibit nearly identical behavior, with the first exhibiting slightly higher cooling water temperature difference. This is mostly due to the larger heat load in the double U-shaped GHX, which is caused by the use of two parallel U-tubes instead of one, which raises the grout temperature and reduces the temperature differential driving the heat transfer process. However, because of the twofold flow rate, such a design provides for higher heat removal rates per borehole, reducing the needed borehole number. It is also observed that the cooling water temperature difference is high in the beginning and gradually decreases with time. This is mostly due to the heat accumulation with time in the grout layer surrounding the GHX's pipes which reduces the temperature difference driving the heat transfer from the cooling fluid to the soil and consquently the fluid temperature difference. For the single U-shaped, double U-shaped, and spiral-shaped GHXs at 17 L/min water flow rate, prolonging the operating period from 24 hr to 72 hr reduces the reached cooling water temperature reduction from 6 °C, 5.9 °C, and 13 °C to 5.3 °C, 4.5 °C, and 9.5 °C, respectively.

Thermal effectiveness: Comparing the thermal effectiveness of the three GHXs at three flow rates, as shown in the same figure, revealed a similar trend for the transient variations of the temperature reduction of cooling water for all designs. As seen in the figure, rising flow rate reduces the GHX's thermal effectiveness since the outlet fluid temperature falls with increasing flow rate. This is due to the fluid not having enough time to exchange heat with the surrounding soil. When compared to other GHXs, the spiral shaped one had the highest thermal effectiveness value due to the better heat transfer of such design due to coils and turns. The spiral shaped GHX attained a thermal effectiveness of 0.86 after 24 hours at 12.7 L/min mass flow rate, which is 95.5 % and 111 % greater than that of the single U-shaped and double U-shaped GHXs, respectively. Increasing the water flow rate to 34.7 L/min decreased this difference to 71.5 % and 101.7 %.

Temperature contours: Temperature contours in the lateral cross sections at depths of 10 m, 50 m, and 90 m and longitudinal cross sections at top, middle and bottom parts of the investigated GHXs after 24 hr of operation at 17.0 L/min mass flowrate are presented in Fig. 3. For lateral cross sections, at a drilling depth of 10 m, utilizing the spiral shaped GHX followed by the double U-shaped GHX increases the rate of heat transfer from the cooling water to the soil, as evidenced by the higher temperature distribution through the grout compared to the single U-shaped GHX. Comparing the inlet fluid temperature at this depth shows that the spiral shaped GHX reduces the water temperature by about 1.2 ℃ compared to 0.4 ℃ and 0.37 ℃ for the single U-shaped and double U-shaped GHXs, respectively. As can be seen, the single U-shaped GHX achieves a little higher cooling water temperature reduction than the double U-shaped, but the heat transfer rate per borehole in the second is greater due to the twofold flow rate. Furthermore, at this depth (10 m), the temperature difference between the down hot water and the upper cooled water is high, allowing for some heat transmission between the two branches, particularly in the case of the spiral shaped GHX due to the close distance between the helical down leg and the straight-up leg tube.

Increasing the depth to 50 m and 90 m results in further decrease in the cooling water temperature, notably in the case of the spiral shaped GHX, which greatly reduces the grout average temperature as can be seen in the figure. A minor temperature difference between the cooling fluid in the spiral shaped GHX and the grout cab can be noticed at a depth of 90 m, in contrast to other designs. Such thermal behavior demonstrates the

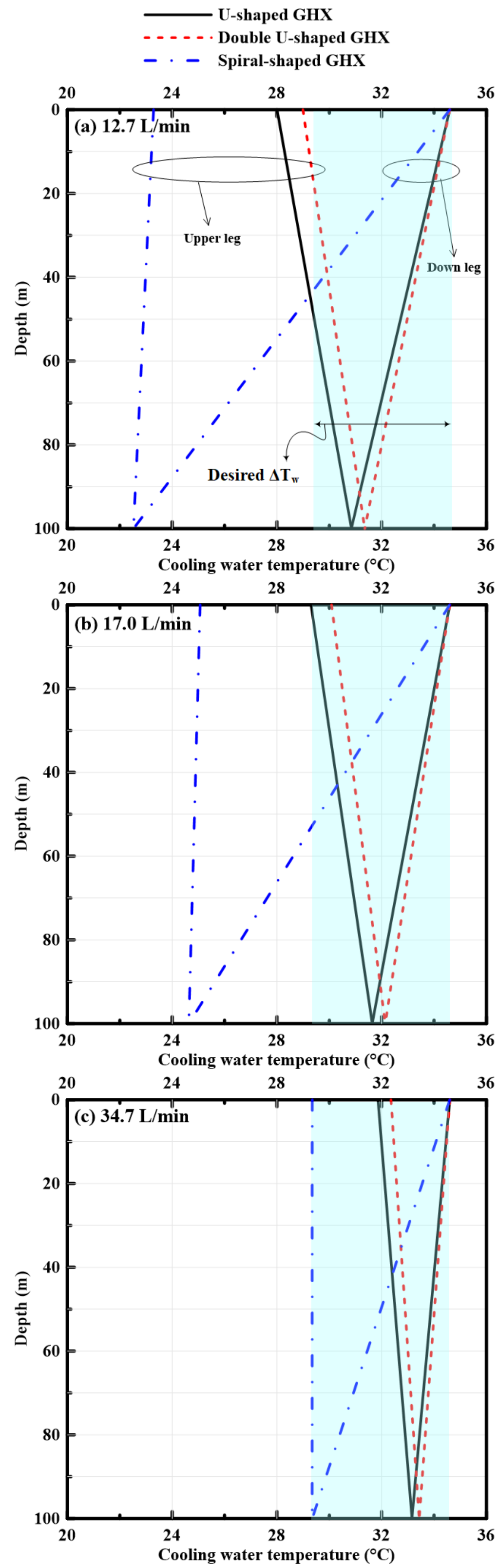

FIGURE 4: The fluid local temperature variation inside the investigated GHXs after 72 hr at mass flow rates of 12.7 L/min, 17.0 L/min, and 34.7 L/min.

efficiency of the spiral shaped GHX in reducing cooling water temperature, which minimizes the required depth or number of boreholes required to dissipate a specific heat load when compared to alternative designs. This is expected to significantly reduce the project's conventional construction costs when compared to other designs, even if the tube length is increased in the case of the spiral shaped GHX due to the substantially higher drilling costs when compared to the cost of pipes.

Local fluid temperature: Figure 4 illustrates the fluid local temperature variation inside the down-leg and up-leg tubes of the single U-shaped, double U-shaped, and spiral shaped GHXs after 72 hr at mass flow rates of 12.7 L/min, 17.0 L/min, and 34.7 L/min. The inlet water temperature is assumed to be constant at 34.6 °C (307.75 K) for all cases to mimic the temperature of cooling water leaves the condenser. Based on the figure, some observations can be made. First, water flowing through the down leg of the spiral shaped GHX achieves the greatest temperature reduction when compared to other designs, followed by the single U-shaped and then the double U-shaped GHXs. This demonstrates the spiral design's superior cooling capacity due to the longer path of the down-leg with coils and turns, which increases the heat transfer surface and, as a result, the heat removal rate. For instance, at a mass flow rate of 12.7 L/min (laminar), the water temperature drops by 3.76 °C (0.0376 °C/m), 3.25 °C (0.0325 °C/m), and 12 °C (0.12 °C/m) via the down-leg of the single U-shaped, double U-shaped, and spiral shaped GHXs, respectively.

Second, except for the spiral shaped GHX, the temperature of the water drops further during continuous flow through the up-leg tubes as can be seen in the figure. This is primarily due to the close proximity of the helical down leg and the straight-up leg tube, combined with a greater temperature difference between the down hot water and the upper cooled water, allowing for some heat transmission between the two branches, as previously clarified in the temperature contours. At 12.7 L/min flow rate, the temperature reduction per unit depth for the cooling water flowing through the up-leg tubes of the single U-shaped, and double U-shaped GHXs is 0.028 °C/m and 0.0236 °C/m, respectively. On the contrary, the water temperature rises by around 0.008 °C/m as it flows through the up-leg straight course of the spiral-shaped GHX. Third, raising the water mass flow rate to 17.0 L/min and 34.7 L/min resulted in similar trends for the change of local fluid temperature for all cases, except that it reduces the cooling water temperature reduction, as shown in Figs. 4a and b. This is due to the fluid not having enough time to exchange heat with the surrounding soil. For instance, at a mass flow rate of 34.7 L/min (turbulent), the water temperature drops by 1.44 °C, 1.165 °C, and 5.25 °C via the down-leg of the single U-shaped, double U-shaped, and spiral shaped GHXs, respectively.

Furthermore, as previously stated, managing the condenser cooling process is critical for the overall stability and efficiency of the HVAC system. This is accomplished not only by controlling the heat removal rate from the condenser via a proper

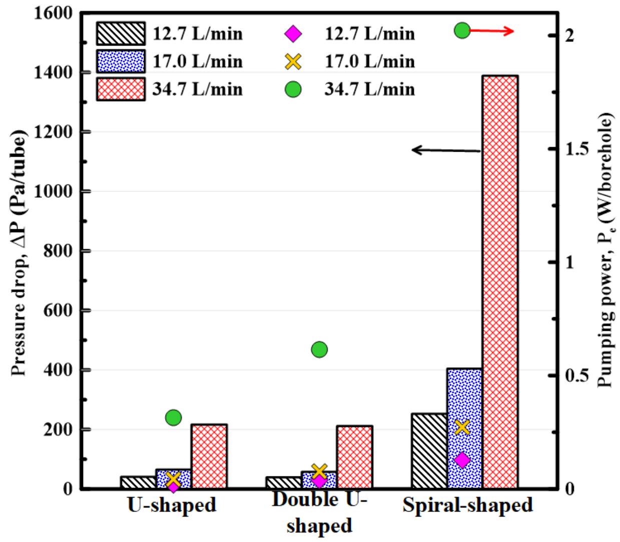

FIGURE 5: The pressure drop per pipe and corresponding pumping power per borehole for single U-shaped, double U-shaped, and spiral shaped GHXs at mass flow rates of 12.7 L/min, 17.0 L/min, and 34.7 L/min.

cooling fluid flow rate but also by adjusting the inlet and outlet temperatures of the cooling fluid passing through it. According to the manufacturer's recommendation [19], the inlet and outlet temperatures of the condenser cooling water should be kept at around 29.4 °C and 34.6 °C, respectively, with a temperature difference of no less than 5.2 °C. Thus, in the current analysis, the minimum acceptable temperature reduction of the condenser cooling water is chosen to be 5.2 °C, which is represented by the blue highlighted area in Fig. 4. As can be seen from the figure, at a lower cooling water mass flowrate of 12.7 L/min (laminar), all of the proposed GHX designs succeeded in achieving a temperature decrease of more than 5.2 °C. In the instance of spiral shaped GHX, 5.2 °C temperature reduction of cooling water is achieved while flowing downward in the helical down leg at a depth of 43 m. In such case, there are two possible scenarios. First, the depth of the borehole can be reduced to around 45 m which saves the installation costs. Second, the borehole depth is kept at 100 m to reduce the temperature of cooling water flowing back to the condenser to more than 5.2 °C (11.3 °C in this situation), considerably improving the HVAC system's COP and minimizing the number of boreholes required. Only a potentially lower construction cost or ease of installation can justify the preferred option. In most scenarios, reducing borehole depth provides greater economic feasibility in addition to easier installation and maintenance processes. In contrast, similar temperature reductions of the cooling water are achieved in the up-leg tube of the single U-shaped and double U-shaped GHXs at 50 m and 19.5 m before leaving the GHX. As a result, the ability to reduce the depth or number of boreholes is limited in both cases, resulting in greater initial expenses as compared to the spiral shaped design.

TABLE 7 Comparison of the cooling fluid temperature difference, ΔT, the total heat exchange rate, Q (kW), the total number of boreholes needed, and finally the COP improvement, ΔCOP (%) for all studied cases after 24 hr, 48 hr, and 72 hr.

		U-shaped GHX			Double U-shaped GHX			Spiral-shaped GHX		
		12.7 L/min	17.0 L/min	34.7 L/min	12.7 L/min	17.0 L/min	34.7 L/min	12.7 L/min	17.0 L/min	34.7 L/min
Fluid temperature difference, ΔT/borehole	24 hr	7.44	6.0	3.2 (N/A)	6.9	5.9	2.9 (N/A)	14.6	13.0	7.67
	48 hr	6.88	5.54	2.9 (N/A)	6.0	4.93 (N/A)	2.46 (N/A)	12.42	10.61	5.9
	72 hr	6.6	5.29	2.74 (N/A)	5.6	4.51 (N/A)	2.24 (N/A)	11.31	9.53	5.25
Heat exchange rate, Q (kW/borehole)	24 hr	6.63	7.14	8.17 (N/A)	12.27	14.048	14.84 (N/A)	12.96	15.42	19.59
	48 hr	6.13	6.58	7.4 (N/A)	10.75	11.7 (N/A)	12.56 (N/A)	11.05	12.59	15.06
	72 hr	5.87	6.29	7.0 (N/A)	9.98	10.7 (N/A)	11.47 (N/A)	10.07	11.31	13.42
Total borehole number	24 hr	165	153	134 (N/A)	89	77	73 (N/A)	84	70	55
	48 hr	178	166	148 (N/A)	101	93 (N/A)	86 (N/A)	99	86	72
	72 hr	186	174	156 (N/A)	109	102 (N/A)	95 (N/A)	108	96	81
COP improvement, ΔCOP (%)	24 hr	7.54	2.58	−5.98	5.57	−4.0	−11.98	41.69	24.97	11.78
	48 hr	5.54	1.03	−6.81	2.17	−6.82	−14.35	35.64	20.24	7.98
	72 hr	4.53	0.25	−7.23	−1.01	−9.47	−16.60	30.09	15.86	4.43

N/A: Not applicable (temperature difference below 5.2 ℃)

At cooling water mass flowrate of 17 L/min (turbulent), only the single U-shaped and spiral shaped GHXs can achieve a cooling water reduction equal to or more than 5.2 °C as shown in Fig. 4b. When the fluid flow rate is increased to 34.7 L/min, the single U-shaped and double U-shaped GHXs become impracticable in parallel arrangements because only the spiral shaped one can achieve a temperature drop of 5.2 °C of the cooling fluid, as shown in Fig. 4c. To address this issue, two or more GHXs of the single or double U-shaped designs can be connected in series to achieve the desired temperature reduction which raises the costs. These comparisons emphasize the significance of GHX design in increasing overall system efficiency and lowering construction costs.

Pressure drops and pumping power: Another important parameter to consider when comparing the proposed GHX designs is pressure drop and the consumed pumping power needed to circulate the cooling fluid. Thus, Fig. 5 depicts the pressure drop per pipe and corresponding pumping power per borehole for single U-shaped, double U-shaped, and spiral shaped GHXs at mass flow rates of 12.7 L/min, 17.0 L/min, and 34.7 L/min. For all flow rates, the pressure losses inside the spiral shaped GHX and acquired pumping power are always greater than the pressure drop inside the single U-shaped and double U-shaped GHXs, as shown in the figure. This is mainly due to the longer path of the helical down leg with coils and turns, which results in more friction with the tube surface and, accordingly, more flow resistance compared to straight pipes. At a mass flow rate of 34.7 L/min, the consumed pumping power per borehole to overcome friction losses and circulate the cooling fluid inside the single U-shaped, double U-shaped, and spiral shaped GHXs is 0.0315 W, 0.63 W, and 2.0 W, respectively.

3.2 Overall system performance and COP improvement.

The developed model is able to predict the thermal energy transfer by conduction to the surrounding soil as well as through the grout and tube wall material. The thermal energy was then absorbed from the moving fluid via convection from the tube's inner surface. The fluid outlet temperature and borehole outer wall temperature were calculated in conjunction with the heat flux at the borehole surface. To conclude, Table 7 presents and compares the cooling fluid temperature difference, ΔT, the total heat exchange rate, Q (kW) estimated using Equation (1), the total number of boreholes needed, and finally the COP improvement, ΔCOP (%) for all studied cases after 24 hr, 48 hr, and 72 hr continuous operation at mass flow rates of 12.7 L/min, 17.0 L/min, and 34.7 L/min. The total number of boreholes needed is calculated by dividing the overall heat rejection rate needed for the proposed case (1100 kW) to the heat exchange rate per borehole for each case.

Based on the results, spiral shaped GHXs outperform other designs, as seen by better heat exchange rates between the fluid and the soil, which translates to a greater temperature reduction of the cooling water. This improvement not only allows for using smaller borehole sizes or numbers which saves the construction costs compared to other designs, but it also improves the COP of the system by significantly lowering the cooling water temperature flowing back to the condenser when compared to the conventional cooling tower. The single U-shaped GHX, on the other hand, acquires a substantial number of boreholes to reject the same amount of heat as the spiral and double U-shaped designs for the same operating period and mass flow rate due to the lower heat removal rate per borehole. Even though the double U-shaped GHX achieves a larger heat removal rate than the single U-tube design due to the twofold flow rate translates to lower number of required boreholes, it frequently fails to achieve the target temperature reduction of the cooling water (5.2 °C), particularly at higher mass flowrates. This is mostly due to the larger heat load in the double U-shaped GHX, which is caused using two parallel U-tubes instead of one, which raises the grout temperature and reduces the temperature differential driving the heat transfer process. This difficulty can be overcome by connecting two boreholes in series to achieve the desired temperature drop, which may increase the number of boreholes required and, as a result, the entire project cost.

At the laminar flow rate of 12.7 L/min, after 24 hr of continuous operation, the heat removal rate per borehole attained by the spiral shaped GHX is 12.96 kW translates to a 14.6 °C reduction in the cooling water and significantly improves the Carnot coefficients of performance compared to the system using cooling tower by about 42 %. At the same operating conditions, the single and double U-shaped GHXs achieved a heat removal rate per borehole of 6.63 kW and 12.27 kW, a cooling water reduction of 7.44 °C and 6.9 °C, and a 7.54 % and 5.57 % improvement in the Carnot coefficients of performance when compared to the cooling tower system. This acquires 84, 89, and 165 boreholes from the spiral, double U-shaped, and single U-shaped GHXs, respectively, to disperse an overall heat rejection rate of 1100 kW. Increasing the flow rate to 34.7 L/min, reduces the required number of boreholes to 55, 73, and 134 for the spiral, double U-shaped, and single U-shaped GHXs, respectively. In contrast to previous designs, only the spiral shaped GHX can achieve a temperature drop greater than 5.2 °C at this flowrate.

Increasing the operation time to 72 hours reduces the heat removal rate from the cooling fluid as a result of heat accumulation in the grout and soil surrounding the heat exchanger tubes, which reduces the temperature difference driving the heat transfer process. This reduces the attained temperature reduction of the cooling water and increases the required number of boreholes. However, all refrigeration units operate for less than 24 hr, interspersed with rest periods when refrigeration is not required which can mitigate this negative effect.

4. CONCLUSION

The purpose of this research is to evaluate the feasibility and performance of a vertical GSR system for cooling a typical 7300 m^2 office building after replacing cooling towers used for condenser cooling with three different designs of GHXs. To that end, a three-dimensional, transient, and conjugated finite volume model is developed and simulated to compare the thermo-hydraulic performance of the traditional single U-tube with that of double U-shaped, and spiral-shaped GHXs at different flow rates. Based on the results, spiral shaped GHXs outperform other designs, as seen by better heat exchange rates between the fluid and the soil, which translates to a greater temperature reduction of the cooling water. This improvement not only allows for using smaller number of boreholes which saves the construction costs compared to other designs, but it also improves the COP of the system by significantly lowering the cooling water temperature flowing back to the condenser when compared to the conventional cooling tower. The single U-shaped GHX, on the other hand, acquires a substantial number of boreholes to reject the same amount of heat as the spiral and double U-shaped designs for the same operating period and mass flow rate due to the lower heat removal rate per borehole.

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