Ground-Source Direct Radiant Cooling System Using Existing Overhead and Underground Water Storage Tanks for South Asian Weather

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***Abstract*—Current research endeavors increasingly emphasize achieving net-zero operations by curbing building energy consumption. Among these efforts, radiant cooling systems offer the potential to reduce energy use, yet their implementation often entails complexity and high costs. This study evaluates the performance of a low-cost ground-source direct radiant floor cooling system for a four-story residential home in South Asia (Karachi, Pakistan). The system uses cold water from an existing underground concrete storage tank commonly found in South Asia, which is supplied to the radiant floor system covering half of the top floor. The water then flows up to an existing overhead storage tank where it can be used to supply building water needs. The system supplements existing air-air split air-conditioning units without requiring extensive modification. The study used an infrared thermometer to measure the temperature of the water in the underground tank and the air temperature of the floor between 1 PM to 5 PM for several days. A mathematical model based on Engineering Equation Solver (EES) was used to estimate the cooling capacity of the system and potential energy savings. The results show that when the air temperature is maintained at 25 ℃, the radiant system can provide an average cooling capacity of 12 Wm^2. Compared to the existing air-conditioning system, the system would save 319 W of electricity, with less than 3 W of additional pumping power due to pressure loss in the radiant pipes. Developing detailed models of building load and underwater tank heat transfer and validating the results using an experimental setup may lead to further insights.**

***Keywords*—hydronic cooling, radiant cooling, building energy efficiency, ground-source direct cooling**

I. Introduction

The need for efficient and sustainable cooling and heating systems has increased with the rise in global temperatures. Radiant systems have been gaining popularity as a sustainable and energy-efficient solution for cooling and heating buildings [1]. Radiant floor heating systems have gained widespread adoption in various regions across the globe. For instance, nearly all residential buildings in Korea and 85% of rural houses in northern China feature this heating technology [2, 3]. In Denmark, Germany and Austria between 30% to 50% of residential buildings that are newly constructed incorporate radiant floor heating systems [2]. Furthermore, when integrated with a ventilation system capable of managing latent load, the radiant cooling system has demonstrated successful implementation in hot and humid climates, such as China, India, Thailand, Singapore, and other similar regions [2, 4]. However, very few new buildings in Pakistan have radiant systems [5]. One of the possible barriers to wide-scale adoption is the excessive cost of additional equipment such as pumps, chillers, and heat exchangers.

To reduce costs associated with radiant cooling systems, various researchers have focused on incorporating renewable energy sources and exploring alternative technologies [6]. Feng *et al.* [7] and Hassan and Abdelaziz [8] suggest incorporating a small chiller powered by solar panels to cool the water circulating through the channels, resulting in quicker and more efficient cooling. Additionally, Luo *et al.* [9] proposed a Thermoelectric Radiant Panel (TERP) system, which utilized Thermoelectric Modules (TEM) instead of conventional water pipes as a heat source. This method provides a more reliable and simplified system control, along with reduced initial and operational expenses, making it an attractive option for cost-conscious projects. However, most of these systems are even more complex than simple radiant cooling systems which discourages adoption.

Mokhtari and Ghasempour [10] proposed an active radiative cooling system for a single-family home located in the hot, semi-arid climate of Iran. The system uses a hydronic cooling system inside the house and rooftop panels to provide cooling. Results of the feasibility study showed that the storage tank and number of panels directly affect the system's performance, while there is an optimum value for the water flow rate. Larger storage tanks and higher water flow rates resulted in a larger cooling energy. However, the system components are not readily available off the shelf, and further studies and tests are required to establish the system's performance and long-term reliability. Other researchers, such as Srivastava *et al.* [11] have explored incorporating more typical and reliable equipment, such as cooling towers, as sources for Radiant Cooling systems. A study conducted by Mokhtari [12] introduces the concept of the Cooling Station, a passive urban cooling system providing radiant cooling to people through rooftop radiative cooling panels. The simulations conducted by Timothy [13] show that hydronic radiant cooling with an evaporative supply water source and dedicated outside air system can achieve significant energy savings (54% to 71%) compared to conventional VAV systems.

Moreover, Olesen [14, 15] has discussed the advantages of hydronic radiant floor heating which include the efficient use of space and that cleaning is not required. Also, the system does not produce noise, cause drafts or use ducts. The system has uniform temperature distribution and is a low-temperature heating system.

In this paper, a simple and cost-effective hydronic cooling scheme was simulated for a residential complex located in Karachi, Pakistan. The proposed setup utilizes an underground water storage tank instead of a conventional heat pump to provide a source of cold water. Temperature readings of the underground tank and the air in the top floor of the building under consideration were taken with the aid of an infrared thermometer. A mathematical model was developed using Engineering Equation Solver (EES) to compute the cooling capacity of the system and its potential energy-conservation benefits. The subsequent section elaborates the methodology used for the study.

II. METHODOLOGY

A. *Modeling Parameters*

Table 1. Design parameters

Parameter	Value
Radiant floor cooling system design parameters	
Pipe material	PEX
Thermal conductivity of pipe material (kt) [18]	0.38 W m^{-1} K^{-1} (0.22 Btu/h.ft ℉)
Pipe nominal diameter [19]	1 in.
Pipe outer diameter (Do) [19]	0.0286 m (1.124 in.)
Pipe inner diameter (Di) [19]	0.0222 m (0.875 in.)
Thermal conductivity of concrete (kp) [20]	1.9 W m^{-1} K^{-1} (1.1 Btu ft^{-1} h^{-1} $℉^{-1}$)
Embedded depth (x_p) [16]	0.05 m (0.164 ft)
Pipe spacing (M) [21]	0.2 m (0.656 ft)
Floor cover material	Marble
Floor cover specific resistance, r_c [18]	0.0317 m^2 K W^{-1} (0.18 ft^2 h ℉ Btu^{-1})
Pipe configuration	Series, 45 sections of 9.1 m (30 ft)
Pipe length	420 m (1379 ft)
Flow rate of water [22]	9.39×10^{-5} m^3s^{-1} (89.2 gpm)
Building Parameters	
Total number of floors	Three floors plus ground floor
Building size	18.2 m by 18.2 m (60 ft by 60 ft)
Radiant floor dimension (3rd floor)	9.1 m by 9.1 m (30 ft by 30 ft)
Underground tank dimensions (length by width by depth)	12.2 m by 9.1 m by 2.4 m (40 ft by 30 ft by 8 ft)
Underground tank capacity	272 m^3 (9600 ft^3 or 71,876 gallons)
HVAC system	
Type	Single air-to-air split air-conditioner
Rated capacity	3.52 kW (1 ton or 12000 Btu/h)
Seasonal cooling COP (SEER)	3.2 (10.9 Btu/kWh)

For the study, the top floor of a four-story residential building (ground plus three floors) was chosen. The specific floor being studied currently uses air-to-air split air-conditioners only during extreme heat, and when not in use (as was the case in March when temperatures were being recorded), ventilation is achieved through windows, doors, and ceiling fans. Table 1 provides additional information on the building and the underground water storage tank. The tank is lined with waterproof concrete, has a large capacity, and exchanges heat freely with the ground, making it a reservoir of cold water that is close to the ground temperature. Various water-based hydronic system configurations are available [16, 17]. The simulation for this study utilized a basic radiant floor cooling system that covered half of the entire floor, consisting of PEX pipes embedded in concrete with a marble top cover and insulation on the bottom (Fig. 1). The design specifications for the system, which were derived from previous studies, are listed in Table 1.

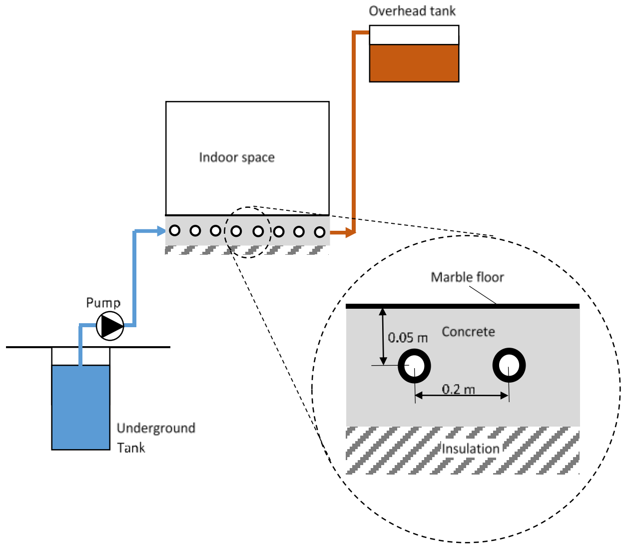

Fig. 1. Schematic of the ground-source direct cooling radiant floor system. The composition of the hydronic floor is also shown.

B. *Heat Transfer Mathematical Model*

The heat transfer calculation from the radiant cooling panel can be divided into two steps as shown in Fig. 1. The first part is the heat travelling from the surface of the panel to the air inside the room air through convection and radiation (Eq. (1)). Various correlations suggested by different standards and researchers to calculate the convection heat transfer has been provided by Feng *et al.* [7]. For the present study, we used the algorithm developed by Walton [23] (Eq. (2)). This algorithm with the "TARP" option has been integrated in Energy Plus and has been used by previous researchers [7]. Finally, the heat that is transferred at the surface cooling systems via radiation can be approximately using Eq. (3) provided in the ASHRAE Handbook, HVAC Systems and Equipment [18].

$$q"_s = q"_{conv} + q"_{rad} \quad (1)$$

$$q"_{conv} = 0.7589\,(T_a - T_s)^{1/3} \quad (2)$$

$$q"_{rad} = 5\times10^{-8}\cdot[(AUST + 273.15)^4 - (T_s + 273.15)^4] \quad (3)$$

where, q''_s, q''_{conv} and q''_{rad}, are the heat fluxes from the panel surface (total), from the panel surface via convection and from the panel surface via radiation respectively. T_s and T_a are panel surface and air temperature, respectively. AUST is the area-weighted temperature of all unheated or uncooled indoor surfaces of walls, ceiling, floor, window, doors, etc. (excluding active cooling surfaces) (AUST was assumed to be equal to T_a which is consistent with section 6.4 of ASHRAE Handbook [18]).

After being absorbed by the radiant surfaces, the heat is transferred between radiant surfaces and the hydronic loop by conduction. This heat is equal to the heat gained by the water in the hydronic loop (Eq. (4)). The surface temperature is approximately constant, however, the temperature of the water rises as it flows through the hydronic loop. This necessitates the use of NTU and effectiveness (ε) relations for a single-stream heat exchanger (Eqs. (5) and (6)) which have also been used by other researchers for hydronic cooling applications [24].

$$q = q''_s A_s = \dot{m}_w c_w (T_{out} - T_{in}) \quad (4)$$

$$\varepsilon = \frac{q}{q_{max}} = \frac{q}{\dot{m}_w c_w (T_s - T_{in})} \quad (5)$$

$$NTU = \frac{U_{sw} A_{sw}}{\dot{m}_w c_w} \quad (6)$$

$$\varepsilon = 1 - e^{-NTU} \quad (7)$$

where, $\dot{m}_w$ and c_w is the mass flowrate and specific heat capacity of water. T_{in} and T_{out} is the temperature of the water entering and leaving the hydronic loop. U_s and as is the overall coefficient of heat transfer and the area for heat transfer from the surface to the water. During the transfer of from the panel surface to the water, the mass of the panel/slab does not produce any delay (i.e., heat transfer takes place under steady state) except for in the case of TABS with night-time precooling. Various methods to characterize the steady-state resistances have been summarized by Feng *et al.* [7]. For the present study, equations provided in the ASHRAE Handbook [18] (Equation were used to calculate the characteristic (combined) panel resistance, r_u (resistance per unit area or specific resistance in m^2KW^{-1} or ft^2h ℉.Btu^{-1}) which is used to find U_s (Eq. (8)).

$$\text{Total resistance} = \frac{1}{U_s A_s} = \frac{r_u}{A_s} \quad (8)$$

$$r_u = r_t M + r_s M + r_p + r_c \quad (9)$$

$$r_p = \frac{x_p - D_o/2}{k_p} \quad (10)$$

$$r_t = \frac{1}{2\pi k_t} ln\left(\frac{D_o}{D_i}\right) \quad (11)$$

where r_t is the thermal resistance of the wall of the tube per unit pipe spacing in a hydronic system ($m^2KW^{-1}ft^{-1}$ or ft^2h ℉.$Btu^{-1}ft^{-1}$), r_s is the thermal resistance between the pipes and panel body per unit spacing between neighboring pipes ($m^2KW^{-1}ft^{-1}$ or ft^2h ℉.$Btu^{-1}ft^{-1}$) ($r_s = 0$ for embedded pipes), r_p is thermal resistance of panel body (m^2KW^{-1} or ft^2h ℉.Btu^{-1}), r_c is the active panel surface covers thermal resistance (m^2KW^{-1} or ft^2h ℉.Btu^{-1}) (For marble floor, r_c = 0.0317 m^2KW^{-1} (0.18 ft^2h ℉.Btu^{-1}) [18].

Eqs. (1)–(11) were used to develop the mathermatical model in Engineering Equation Solver (EES). EES pipeflow function [25] was used to calculate the additional pumping power due to the pressure loss in the radiant pipes. The pump efficiency was assumed to be 70% [26]. The underground tank water temperature and the air temperature in the top floor were measured using an Berrcom JXB-178 infrared thermometer (Accuracy ±0.3 ℃ or ±0.6 ℉). Multiple readings between 1 PM to 5 PM (building was found to be hottest during this period) were taken for 10 days from March 5th to March 20th. No data was recorded on six days (6th, 9th, 13th, 14th, 16th, and 19th March) due to lack of access to the site. The daily average temperatures were used to estimate the cooling capacity and energy savings under the scenario under which the floor is maintained at 25 ℃ by the Air-conditioners. The measured temperatures and modeling results are presented in the subsequent section.

III. RESULTS

Fig. 2 shows the daily average temperature of the top floor indoor air, underground tank and outside air measured from 5th March to 20th March. The temperature of 25 ℃ is also shown (dashed line) in the figure. The top floor indoor air temperature is remarkably close to the outdoor air temperature. This is to be expected since the floor under consideration currently has air-to-air split air-conditioners which are only used under extreme heat. The units were not in operation and ventilation from windows and doors along with ceiling fans were being used. Overall, the average top floor indoor air temperature and outdoor air temperature is 32.6 ℃ and 31.7 ℃. In contrast, the underground tank is consistently colder than the outdoor air (average ΔT = 10.2 ℃, minimum ΔT = 9 ℃). Furthermore, except for one day (10 March), the underground tank is always colder than 25 ℃.

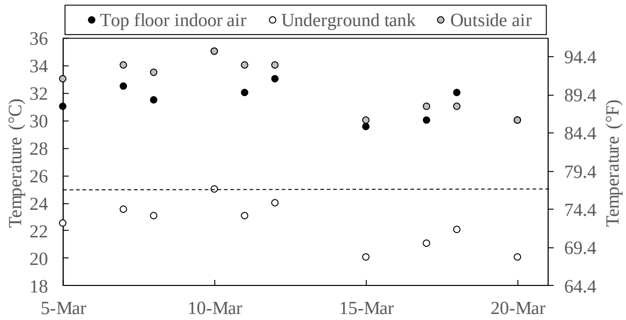

Fig. 2. The daily average temperature of the top floor indoor air, underground tank and outside air measured from 5th March to 20th March. The temperature of 25 ℃ is also shown (dashed line).

Fig. 3 shows the heat transfer rate (W) and the heat transfer per unit floor area (Wm^{-2}) from 5th March to 20th March. It can be seen that that aside from 10th March, the radiant system contributes to the cooling capacity. On average, the radiant cooling system provides 1020 W cooling to the conditioned space while the maximum cooling rate is 1962W. The average cooling provided per unit radiant floor area is 12.2 Wm^{-2} while the maximum flux is 23.5 Wm^{-2}.

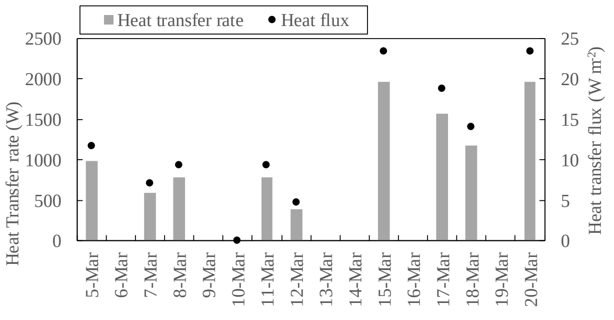

Fig. 3. The heat transfer rate (W) and the heat transfer flux (Wm^{-2}) from 5th March to 20th March.

Fig. 4 shows the electric power that could be saved if the cooling is provided from a radiant cooling system instead of the current air conditioning system (with a COP of 3.2). It can be seen that on average 319 W of electricity, with the maximum saving of 613 W. The additional pumping power due to the pressure loss in the radiant pipes was also calculated to be 2.79 W which is negligible compared to the electricity saved.

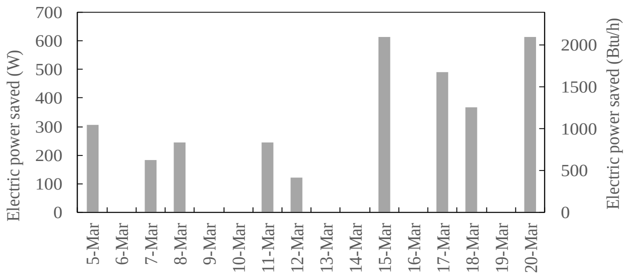

Fig. 4. The heat transfer rate (W) and the heat transfer flux (Wm^{-2}) from 5th March to 20th March.

It is crucial to comprehend the assumptions and circumstances that were considered while calculating the savings. It is essential to note that the savings are based on the maximum heat removal from the conditioned space when it is maintained at 25 ℃ using the radiant cooling system. If the conditioned space is hotter than 25 ℃ (as is the case in March), the heat removed by the radiant system will increase which would increase the electricity saved. Conversely, if the building envelope is improved and sources of heat gains are minimized, less heat will need to be removed from the system, decreasing the potential energy savings. Additionally, during periods when the underground tank is hotter than 25 ℃ or the indoor air is colder than 25 ℃, there will be zero savings.

IV. Conclusion

A low-cost ground-source direct radiant floor cooling system for a four-story residential home in Karachi was proposed. The cold water from an existing underground concrete storage tank is pumped to the radiant floor system and afterwards the water would flow up to an existing overhead storage tank where it can be used to supply building water needs. An infrared thermometer measures the temperature of the underground tank and the air temperature of the floor between 1 PM to 5 PM for several days. A mathematical model shows that when the air temperature is maintained at 25 ℃, the radiant system can provide an average cooling capacity of 12 Wm^{-2} which for the present system would correspond to 319 W of electricity saved. It is essential to note that the saving depends on the indoor air temperature being 25 ℃ and underground tank being at a temperature lower than 25 ℃. Other considerations such as building heat gains would also affect the estimated saving. Efforts are currently in progress to create mathematical models for (a) the building's load and (b) the heat transfer of the underwater tank. The validation of the findings through an experimental setup is planned for the future. Moreover, a system that utilizes the hot water from the overhead tank to provide heating will also be implemented for areas with substantial heating and cooling requirements.

Conflict of Interest

The authors declare no conflict of interest.

Author Contributions

Antash Najib was responsible for conceptualization, design of experiment, developing the simulation model, writing parts of the original draft, writing the review & editing and overall project supervision; M. Ahmed Memon was responsible for literature review, experimental setup and measurement, and writing the original draft; both authors had approved the final version.

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