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Experiments Using Capillary Mat as Ground Heat Exchanger for Ground Source Heat Pump Heating Application

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Abstract

The cooling and heating of spaces are among the largest sources for household's energy demand. Ground Source Heat Pump (GSHP) is a promising technology to reduce the energy for cooling and heating purposes. However, the major obstacle hindering the utilization of this technology is the high initial cost, especially for the installation of ground coupled heat exchanger. The horizontal closed-loop system offers lower installation cost, as it requires no vertical borehole construction. Instead, the heat exchangers can be installed in shallow trenches that may be excavated, by small excavator or even by human labor. This paper presents the comparison of two different heat exchangers, namely, the capillary mat and the widely used slinky pipe. Both heat exchangers are connected to a heat pump, where continuous heating tests were carried out for 165 hours (~7 days) for each configuration. The purpose of this research is to show the performance of capillary mat in comparison to slinky pipe. Despite during the entire test for capillary mat required 6% higher electricity consumption, compared to slinky heat exchanger, the results still suggest the potential use of capillary mat as alternative to slinky heat exchanger. Additionally, the results also highlight the high hydraulic resistance of installed capillary mat heat exchangers may become the major disadvantage of the capillary mat.

Keywords

Horizontal Heat Exchanger, Ground Source Heat Pump, Capillary Mat

1. Introduction

The Ground Source Heat Pump (GSHP) has been widely used as an alternative way to reduce the electricity consumption for space cooling and heating [1] [2]. Unlike the conventional Air Source Heat Pump (ASHP), which uses ambient air, GSHP utilizes stable ground temperature, as heat source or heat sink. Ground temperature has a relatively stable temperature due to its high heat capacity, in contrast to fluctuating air temperature. In area where ASHPs are extensively used, such as big cities, extensive use of ASHP for cooling in summer season could accelerate the heat island effect [3] [4] [5] [6]. This is a phenomenon where the air temperature is significantly higher than its surrounding area because of higher heat in-flux and lower heat dissipation. Kakegawa *et al.*, in 2002, studied that the replacement of ASHP with GSHP in Wast-Shinjuku area, Tokyo, could reduce 54% of CO_2 emission on the area [3]. Kardinal Yusuf *et al.* investigated the relationship between industrial urban and commercial land use with the increasing urban heat island in Singapore [5]. Arifwidodo *et al.* investigated the household energy consumption and its impacts on the urban heat island in Bangkok area [6]. Further, they concluded that the combination of urban heat island mitigation, adaptation planning and energy-efficient housing design would contribute to better solutions.

The GSHP system can be expected to be an alternative to solve this problem. However, the growth of GSHP utilization is relatively slow as a result of high initial cost. Most of the initial costs arise from the installation of ground heat exchanger [7] [8] [9] [10]. In the closed-loop GSHP system, there are mainly two classifications of ground heat exchanger configuration, namely, vertical and horizontal configurations. Vertical configuration allows higher heat rejection and/or extraction rate by the groundwater advection. However, the drilling of borehole is the highest cost for most cases. On the other hand, GSHP with horizontal ground exchanger is cheaper, as the heat exchanger can be installed in a shallow trench. The trench is typically 1 - 2 meters depth, which only requires small excavators, or even direct excavation by human labor [11]. However, soils in shallow depths are affected by the atmospheric temperature, soil thermal properties, depth and climatic conditions [12] [13] [14]. Thus, a careful assessment and planning must be carried out prior to the installation.

The use of high-density polyethylene (HDPE) pipe-based horizontal heat exchanger, configured in both slinky and helical configurations has been investigated in various researches. Wu *et al.* performed experimental and numerical study on slinky ground heat exchangers [15]. From the validated numerical results, they extended the study by performing sensitivity analysis on other parameters. Widiatmojo *et al.* investigated the use of GSHP coupled with horizontal ground heat exchangers in Thailand. The results revealed that in the hot-humid cooling dominated region, the use of GSHP can still provide a significant electricity reduction [11]. Xiong *et al.* developed validation model of Slinky ground heat exchanger by using analytical ring source method [16].

Unlike the slinky or helical ground heat exchangers which are very popular, the use of capillary mat as ground heat exchanger is still limited. Most of the researches focus on the application of capillary mat for radiant heat exchanger inside buildings [17] [18] [19] [20]. Zhou and He evaluated the thermal performance of radiant floor system with various heat storage materials and heat exchanger pipes [17]. They showed that the capillary mat provides more uniform temperature and achieves it in a significant shorter time, in comparison to polyethylene coils. Further, their results suggested that the combination of capillary mat and phase change material (PCM) provides advantages over another system. Xia and Zhang showed potential cost reduction by proposing a new double-layer radiant floor system with organic phase change material for heat storage during summer and heat source in winter [18]. Capillary mat heat exchangers are popular among other heat exchangers, such as, copper pipe, for its easy installation, lower cost [19]. Carbonell *et al.*, develop a numerical model for ice storage by using capillary mat heat exchangers. Their numerical model was validated with the experimental results for both cooling and heating stages. Further, they also pointed out the robustness and reliability of the capillary mats for the given low temperature test condition, although longer test periods need to be carried out to evaluate its long-term reliability [20]. Zhao *et al.* presented the comparison between capillary mat radiant and floor radiant heating system by ASHP system in residential building. The result showed that capillary mat radiant consumed 45% less electricity than floor radiant heating system [21]. The use of capillary mat incorporated into ceramic panel and passive cooling system in Mediterranean housing was studied by Echarri [22]. To the best of author's knowledge, technical applications of capillary mat as ground heat exchanger are very limited, for instance in China [23] and Europe [24].

Compared to the HDPE pipe-based ground heat exchangers, capillary mat allows simple installation and transportation as it requires no on-site loop preparations. In addition, small capillary tubes provide more heat transfer area available per unit footprint of installation area, which is important for installation in space-limited area.

This study presents the experimental results of continuous space heating by using GSHP, coupled with two different types of ground heat exchangers, namely slinky heat exchangers and capillary mat heat exchanger. The aim of this study is to evaluate the performance of capillary mat as shallow ground heat exchangers for a GSHP system, in comparison to a widely used HDPE-based slinky ground heat exchangers.

2. Test Field and Heat Exchanger Configurations

The experimental tests were conducted at the Renewable Research Center, National Institute of Advanced Industrial Science and Technology, in Koriyama, Fukushima Prefecture. The research center is located approximately within a 60 km radius from the crippled Fukushima Daiichi Nuclear Power Plant. The climate at the research center is classified as humid continental (Koppen Cfa). In 2018, the monthly average temperature was highest in July (26.3˚C) and lowest in January (0.2˚C). Meanwhile, the annual precipitation in 2018 was 837.5 mm/year, with precipitation being highest in September (146 mm/month). The soil within the top layer of 2.5 m was mainly silty sand with the existence of gravel (average diameter of ~1 cm) observed in some locations. **Figure 1** shows the annual temperature variations, starting on April 1st, recorded in the test field at 0.1 m, 0.5 m, 1 m, 1.5 m and 10 meter depths [25].

The experimental field was designed to have several heat exchangers for comparative analysis, as illustrated in **Figure 2**. Vertical heat exchangers comprised of HDPE pipes, installed in two 40 m (double U-tube) boreholes and a 100 m (single U-tube) borehole. Horizontal (shallow) heat exchangers were slinky-coil HDPE pipes and capillary mat heat exchangers. Both types of horizontal heat exchangers were installed in two orientations, horizontal and vertical, at a depth of 1.4 m. However, only horizontally configured capillary mats and slinky heat exchangers are to be discussed in this paper.

Two trenches, each of which occupies 67.6 m^2 (13 m × 5.2 m) area, were excavated for both slinky and capillary heat exchanger configurations. The trenches are wider than the total area required for both types. These extra spaces are designed to allow piping connection as well as proper spacing between heat exchangers.

The installation of slinky heat exchanger involves pre-shaping HDPE pipe into the slinky configuration with necessary diameter and pitch size. This step, however, can be omitted for the case of capillary mat.

All heat exchangers are connected to header valves allowing different combination of heat exchangers to be selected. The distance from the main header valves to both heat exchangers are vary between 18 m to 45.5 m, depending on its position. The lateral distance between main header valves to the compressor unit inside the experimental room is 17 m. However, due to the compliant with building standard for earthquake measures, the pipe connection must be installed through the building supporting system, 3 m below the ground level, in series of bends and curves. The heat exchangers and the heat pump are connected by the thermally-insulated HDPE pipes having 3.2 cm outer diameter and 2.58 cm inner diameter. The heat pump having 4 kW and 5 kW of cooling and heating capacity, respectively, was installed in an 8.6 m × 6.4 m × 2.8 m experimental room. The 38% propylene glycol-water mixture is used as ground loop heat exchanger fluid. Several sensors were installed to record temperatures, flow rate, and humidity for the thermal performance analysis of different heat exchangers. The sensors are listed in **Table 1**. All sensors are connected to the GL-820 data logger and the data are recorded in every 1-minute interval. The weather data are obtained from the weather station installed approximately 100 m away from the test field. **Figure 3** presents the experimental room and the installation area of Ground Heat Exchangers (GHE).

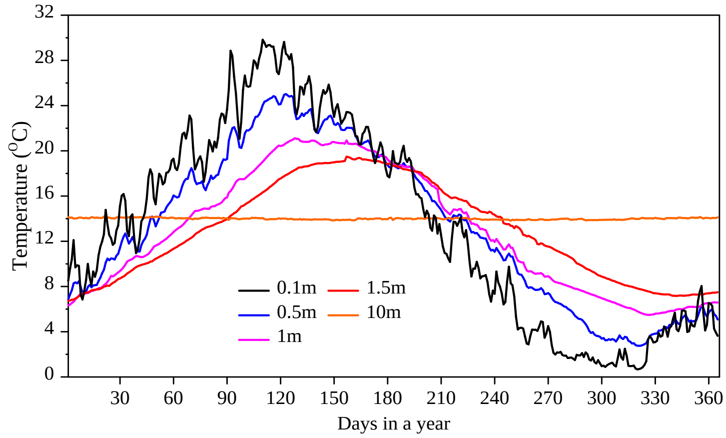


Figure 1. Annual soil temperature recorded at the test site, from April 1, 2018 to March 31, 2019.

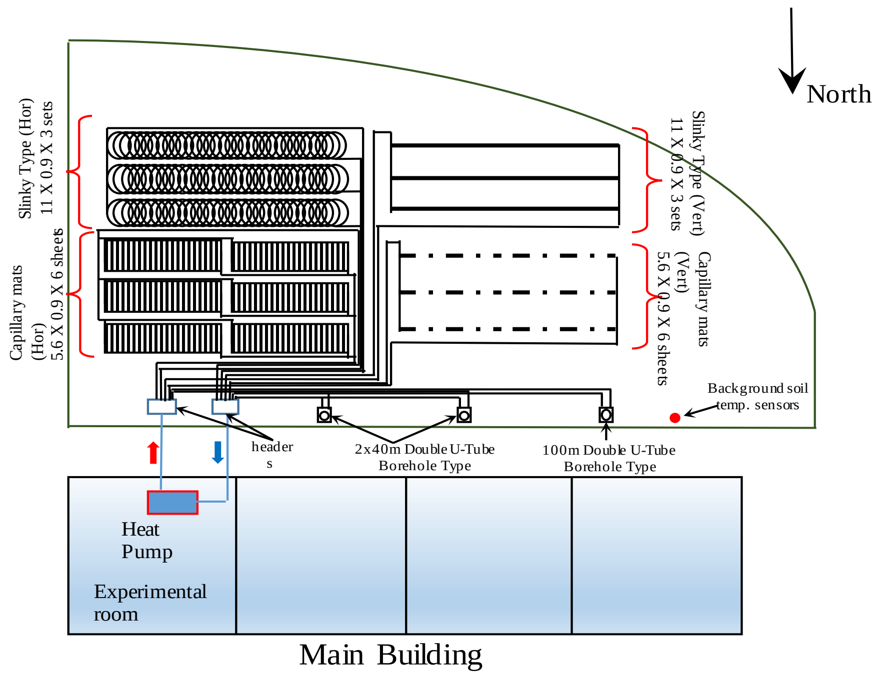


Figure 2. Schematic showing the configuration of heat exchangers in the test field.

Figure 3. Experimental room with the main GSHP system (a) and installation area of GHE's (b).

Table 1. List of sensors.

Location/sensor	Sensor Type	Accuracy
Fluid HP outlet	PT100 class A	±0.15˚C
Fluid HP inlet	PT100 class A	±0.15˚C
Flow meter	Electromagnetic flowmeter	±0.5 L/min
Room temperature	PT100 class B	±0.3˚C
Room Rel. humidity	Polymer resistance type	±3%
Outside temperature	PT100 class B	±0.3˚C
Outside Rel. humidity	Polymer resistance type	±3%
FCU fan outlet tem	PT100 class A	±0.15˚C

2.1. Capillary Mat Heat Exchangers

The capillary mat heat exchanger was made from polyethylene material having dimension of 5.6 m (in the flow direction) × 0.9 m × 0.0064 m. The exchanger comprised 117 small capillary tubes arranged in parallel, with each tube having an outer diameter 0.0064 m and inner diameter 0.0048 m. A single heat exchanger is equivalent to a 655.2 m similar small capillary tube providing a heat exchange area of 13.178 m^2 over an installation area of 5.04 m^2. In total, the equivalent heat transfer area for six capillary mats is 79.07 m^2. Figure 4 shows the capillary mats being prepared for installation as ground heat exchanger.

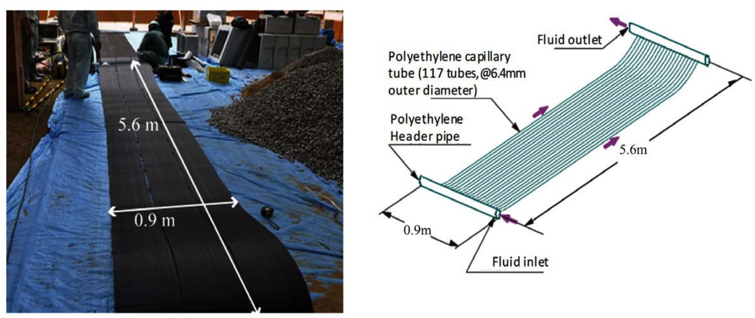
Six horizontal capillary mat heat exchangers were installed at a depth of 1.4 m in three rows and two columns, as shown in Figure 5. All mats were configured in parallel to the main pipeline. The main reason for adopting this configuration was that the pressure drop of a single mat is relatively high owing to the small-diameter capillary tubes. Considering the performance curve of the circulation pump given by the manufacturer, any series connection can result in a high pressure drop, exceeding the capacity of the pump.

2.2. Slinky Heat Exchangers

The HDPE pipe having inner outer diameter, d_0 = 0.032 m and inner diameter, d_i = 0.0254 m, were configured in 0.9 m diameter and 0.3 m pitch slinky loops configuration. With this configuration, a 120 m length HDPE pipe is converted into 11 m slinky. The slinky heat exchangers were installed in three parallels connections, each of which having an actual length of 120 m. The spacing between slinky was set 0.6 m. Figure 6 shows the schematic arrangement of slinky heat exchanger. The total available heat transfer area is equivalent to 36.173 m^2, which is 54.25% less than those for capillary mat configuration.

3. Continuous Heating Test

The continuous heating test for each heat exchanger type was carried out continuously for 7 days for each type. The test for capillary mat configuration was accomplished on January 15-22, 2019 and February 1-8, 2019, for capillary mat and slinky configuration, respectively. Figure 7 shows the ambient (outdoor) temperature and background soil temperatures at 1 m and 1.5 m depths. The soil temperatures during the slinky test were lower than during the test for capillary mat, by 1˚C - 1.2˚C. The data shows that the soil temperatures during slinky test, at 1 m and 1.5 m depths were 6.6˚C and 8.5˚C, respectively, in contrast to 7.5˚C and 9.7˚C, respectively, for capillary mat configuration. Meanwhile, at the end of the test, the soil temperatures at 1 m and 1.5 m depths were 6.1˚C and 8.1˚C, for the case of slinky test and 7.2˚C and 9.1˚C for capillary mat. The temperature profile shown in Figure 1 also showed that the soil heat flux at the heat exchangers depth was moving upward to the surface during the tests periods, as indicated by positive temperature gradient in upwards direction. It must be noted that the background soil temperatures were not measured in the ground heat exchanger location. Thus, it may not accurately represent the actual ground temperature at heat exchanger location. Nevertheless, the temperature variations in ground heat exchanger locations are likely to have similar variation with the background data. For both cases, the heating temperature setting was set to 21˚C, with 50% fan speed and fixed 45-degree nozzle position. Figure 8 shows the wind velocity, direct solar radiation, air pressure and relative humidity data during the tests. No rainfall was recorded throughout the test periods. To minimize the heat loss from the experimental room, the central AC system and ventilation were turned off. Also, window's blinds were closed as well as the activities inside the experimental room were kept as minimum as possible throughout the test periods. Thus, the thermal load inside the building was only affected mainly by the outdoor temperature fluctuations.



(a) (b)

Figure 4. Capillary mat heat exchanger being prepared for installation (a) and its schematic description (b).

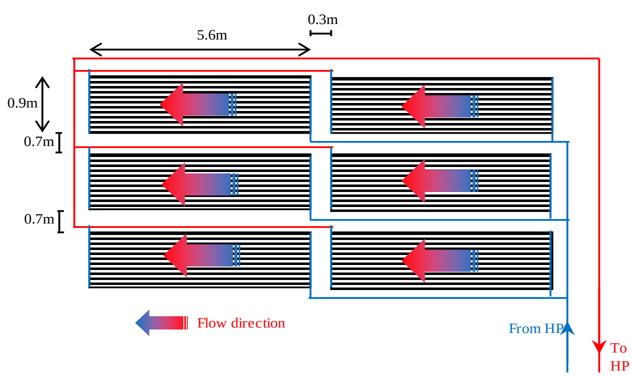


Figure 5. Schematic arrangement of the capillary mat heat exchangers.

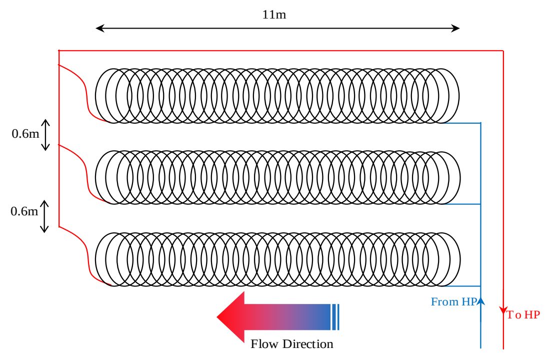


Figure 6. Schematic arrangement of the slinky heat exchangers.

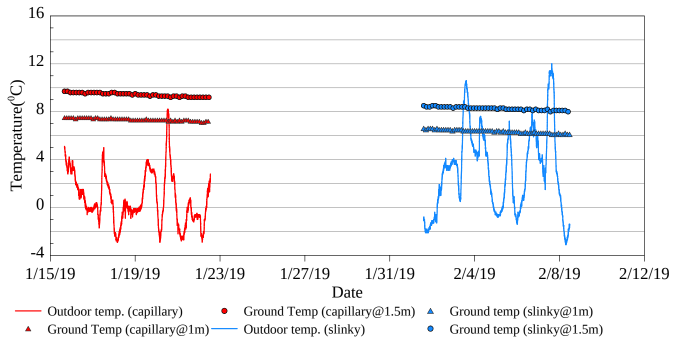


Figure 7. Outdoor temperature and ground temperatures at 1 m and 1.5 m depths, during experimental periods.

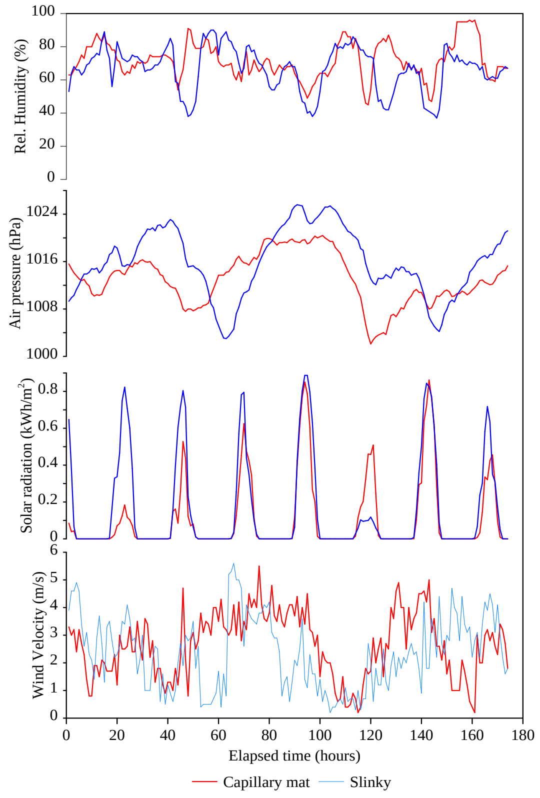


Figure 8. Outdoor temperature and ground temperatures at 1 m and 1.5 m depths, during experimental periods.

4. Analysis and Discussion

4.1. Quantitative Analysis

The Coefficient of Performance (CoP), representing ratio of heat supplied into the room to the work required is calculated as:

$$\text{CoP} = Q_H / W_T \tag{1}$$

here, Q_h (in W) is the rate of heat supplied to the building and W_T (in W) is the total electrical power consumption (see **Figure 9**), calculated as

$$W_T = W_C + W_F + W_P \tag{2}$$

where W_C, W_P, and W_P are the electrical power for the compressor, fan, and circulation pump, respectively.

The data logger records the temperature and the both inlet and outlet GHE fluid temperatures as well as the flow-rate. Thus, Equation (1) is re-written as:

$$\text{CoP} = \left(Q_G + W_C\right) / W_T \tag{3}$$

where, Q_G is the heat extraction rate from the ground, calculated as

$$Q_G = \left(T_{\text{in}} - T_{\text{out}}\right) \rho c V_m \tag{5}$$

T_{out} and T_{in} (in C) are, respectively, the GHE fluid temperatures at the heat-pump's outlet and inlet, while ρ (kg/m^3), c (J/kgC), and V_m (m^3/s) are, respectively, the density, specific heat capacity, and flowrate of the heat exchange fluid.

In order to quantify the variation of data during operational tests, standard deviation analysis is used

$$\text{CoP} = Q_H / W_T \tag{5}$$

here, x_i is the observed value, $\bar{x}$ is the average value, and N is the number of data points.

Assuming, the temperature difference between indoor and outdoor is the only source of thermal load, building thermal load can be calculated by:

$$Q_L = hA\left(T_{\text{indoor}} - T_{\text{outdoor}}\right) \tag{6}$$

here, Q_L (Watt) is the thermal load, h(Wm^{-2}K^{-1}) is the overall heat transfer coefficient of building's wall, A (m^2) is the area of heat transfer. As the current experimental tests were carried out in a similar room, the unknown value h and A are the same. Thus, these values can be left unknown by introducing the normalized value, Q_L/hA (˚C), indicating that the thermal load for both slinky and capillary mat are only depend on the outside and indoor (room) temperature differences. In a precise analysis, the calculation of thermal load by conduction through the wall must also considers the total equivalent temperature different (TETD), in which heat conduction due to direct solar radiation is included. However, in this study, the experimental room is positioned in the North part of building (1st floor), receives almost no direct sunlight during winter. Furthermore, the window shades were closed throughout the experiments, so that the effects of solar radiation are negligible.

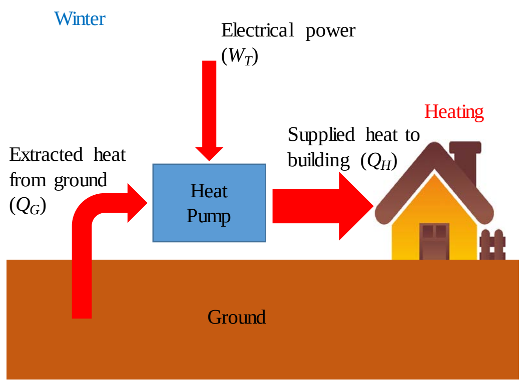


Figure 9. Schematic of energy flow during space heating using GSHP.

4.2. Temperatures and Ground Heat Extraction Rate

Figure 10 shows the heat pump's inlet and outlet temperatures for both heat exchangers, as well as the flow-rate of heat exchangers fluid. The test using slinky heat exchangers indicated lower heat-pump operation temperature compared to the test using capillary mats. This indicated the lower ground temperature during the test using slinky heat exchanger. The inlet and outlet temperature difference are higher for the capillary mats heat exchanger. This is attributed to the lower flow-rate of heat exchange fluid.

Figure 11 shows the outdoor temperatures, room temperatures and Fan Coil Unit (FCU) air temperatures during tests. The FCU air temperature was measured at the fan outlet. It is important to note that despite the ground temperatures at heat exchanger's depths, during the test using slinky heat exchanger are lower, the average outdoor temperatures indicate the opposite. Average outdoor temperature during slinky test is 3.04˚C ($\sigma = 3.49$) while for the capillary mats is 0.69 ($\sigma = 2.14$). The effect of higher outdoor temperature can be clearly seen at t = 40 - 80 hours and t = 140 - 160 hours, where the FCU temperatures for slinky case are lower than those for capillary mats. Despite similar temperature setting were used for both configurations, the room temperature is slightly higher for the slinky configuration with average of 20.07˚C ($\sigma = 0.48$), in contrast to 19.34˚C ($\sigma = 0.38$), for the capillary mat. The higher standard deviation for slinky test can be explained as a result of higher thermal load fluctuations due to greater outdoor temperature variations. To compensate the thermal load fluctuation, the heat-pump system adjusts its operational parameter in two ways, the variable inverter control and/or ground circulation flow-rate. The data from **Figure 10** and **Figure 11** suggest that heat-pump changes its heating load predominantly by adjusting the refrigerant compression ratio through variable inverter control, rather than flow-rate. This can be seen from a relatively constant flow-rate throughout the test.

The total heat extraction and calculated CoP are shown in **Figure 12**. The calculated CoP for slinky and capillary mat are 2.81 ($\sigma = 0.40$) and 2.78 ($\sigma = 0.28$), respectively. The heat extraction rate and heat exchange fluid temperatures are showing consistent outcomes with the FCU temperatures. In addition, the heat extraction rates were also decrease during these periods.

4.3. Electricity Consumption and Coefficient of Performances

Figure 13 presents the electrical consumptions for both heat exchanger configurations. The average electrical consumptions and the standard deviations for slinky and capillary mat, are respectively, 0.90 kW ($\sigma = 0.16$) and 0.96 kW ($\sigma = 0.16$). The total electricity consumptions are 149.67 kWh and 159.35 kWh, respectively. Slinky configuration consumed lower electrical energy due to lower heating load at t = 40 - 80 hours and t = 140 - 160 hours, which is consistent with the temperature and heat extraction rate, discussed in previous chapter (sub-chapter 4.2).

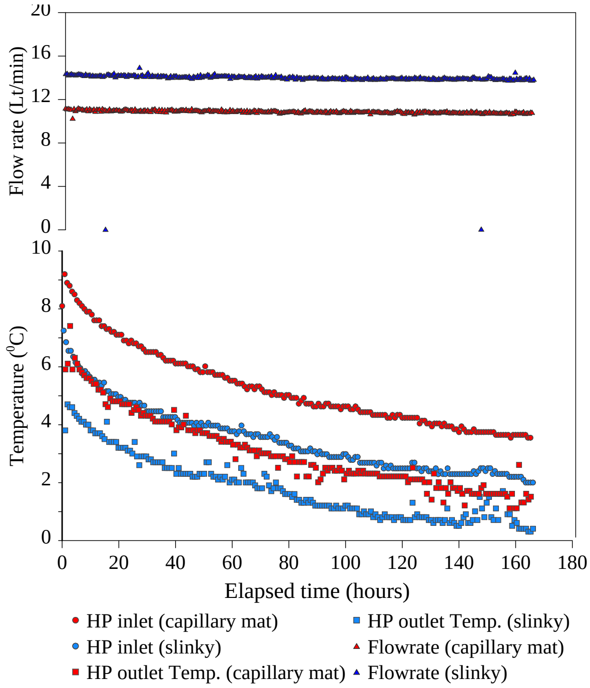


Figure 10. Heat-pump inlet and outlet temperatures and volumetric flow-rate of ground heat exchanger fluid.

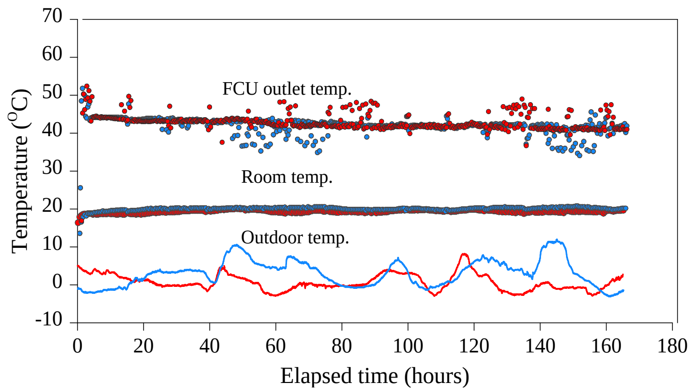


Figure 11. Outdoor temperatures, room temperatures and Fan Coil Unit (FCU) air temperatures for both slinky and capillary mats heat exchangers.

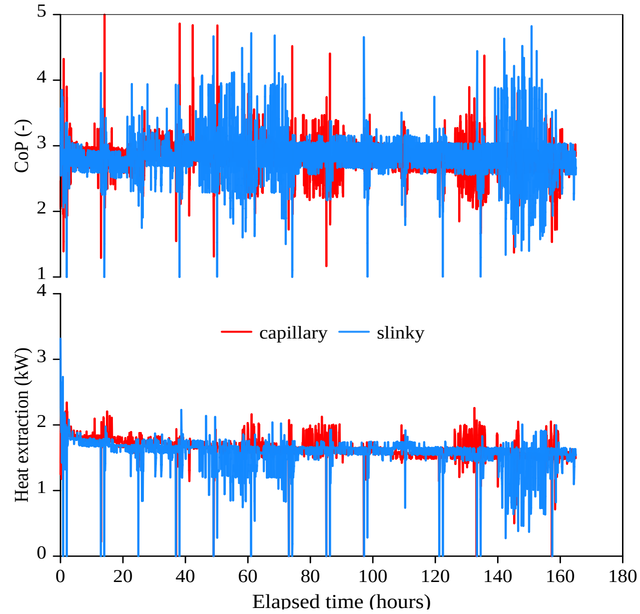


Figure 12. Heat extraction rates and coefficient of performances.

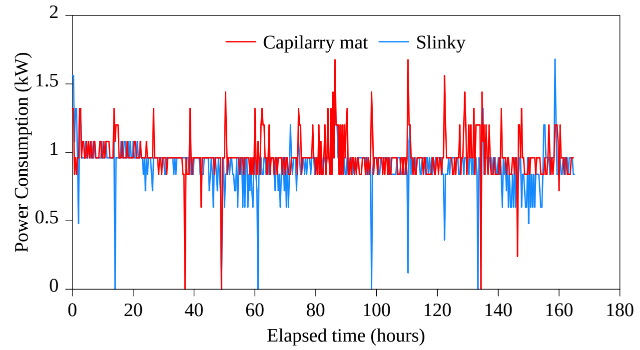


Figure 13. Electricity consumption for both capillary mat and slinky heat exchanger.

4.4. Flow Rate of Heat Exchange Fluid

According to **Figure 10**, the average flow-rates for capillary mats and slinky heat exchangers are respectively, 10.88 Lt/min and 13.98 Lt/min. The experimental data with slinky heat exchanger shows larger fluctuations ($\sigma = 0.85$) in contrast to the capillary mat ($\sigma = 0.53$). Another important factor is the distance between the heat pump and the ground heat exchangers. The distance from the main header valves to the heat exchangers vary between, approximately 18 m to 45.5 m, depending on its position. The lateral distance between main header valve and the compressor unit inside the experimental room is 17 m. The series of bends and curves in pipeline also increase the effective hydraulic resistance. On the other hand, both flow-rates decline steadily, with slightly higher linear decline rate for slinky heat exchanger. The main reason is likely the higher hydraulic resistance of capillary mat. This is attributed to the small diameter of capillary tubes and abrupt contraction between capillary mat's header and capillary tubes. Thus, higher hydraulic pressure is required to attain a given flow-rate.

4.5. Thermal Load

The calculated values of normalized thermal load for both cases are presented in Figure 14. It can be clearly seen that the thermal load during test using capillary mats is higher compared to the slinky heat exchangers, mainly as a result of different outdoor temperature during both tests. We found the average value $T_{(indoor)}$-$T_{(outdoor)}$ are 18.64˚C (σ = 2.15) and 17.07˚C (σ = 3.3) for capillary mat and slinky, respectively. The higher value of standard deviation for slinky indicates more thermal load variations. For the duration of the entire tests, thermal load for capillary mat is 9.3% higher than those for slinky.

5. Conclusions

In this study, experimental tests have been carried out for a heat-pump coupled with two different ground heat exchangers, namely capillary mats and slinky type heat exchangers. Both tests were carried out continuously for 165 hours (~7 days). Both tests have been carried out in the periods where outdoor and ground temperatures were moderately different. During the test with slinky configuration, the average outdoor temperature was 3.04˚C, in contrast to 0.69˚C for the capillary mat. Similarly, the average thermal load during slinky test was lower than the capillary mat. Due to this condition, the thermal load during the test with capillary mat was found to be 9.3% higher than the test using slinky.

The background soil temperatures, located nearby the ground heat exchangers, also indicate temperature decrease during both tests. The ground temperatures at heat exchanger location are presumably higher during the test with capillary mat, compared to slinky. This is primarily due to the test for capillary mat was carried out earlier than the slinky.

Data analysis also indicated that for the given conditions, the slinky test consumed 149.67 kWh electrical energy, which is 6% lower than those for capillary mat, which is strongly affected by the thermal load. The flow-rate of heat exchange fluid also suggests that the circulation pump works at high pressure, especially during the test for capillary mat. In addition, the far distance between heat pump and ground heat exchangers as well as bends and curves along pipelines increase the effective hydraulic resistance.

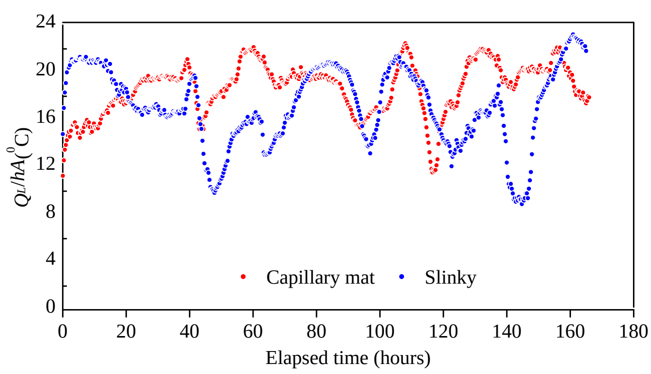


Figure 14. Calculated heat load for the both experiments using capillary mats and slinky heat exchangers.

The overall results point toward the technical applicability of capillary mat as alternative to HDPE-based shallow heat exchanger.

6. Future Study

Despite present study provides insight on the possibility of using capillary mat as ground heat exchangers, as an alternative to the well-known slinky heat exchanger, there are some points that must be considered for further analysis. First, even though the tests were carried out in the same winter season, the time gap between two tests resulting in different ground temperatures. Second, the heat-pump thermal outputs, which correspond to the thermal loads, are only about 32% of heat-pump total heating capacity. Consequently, the heat extractions are also relatively low, which may not reflect the ideal capacity of ground heat exchangers. Third, the thermal loads were not equal throughout the entire tests due to the different outdoor temperatures. These considerations making an ideal comparison for both heat exchanger configurations are impractical.

The future studies will consider Thermal Response Test (TRT), which enable the control of main test parameters, such as the heating rate and the flow-rate. This is expected to eliminate the second and third considerations mentioned above. On the other hand, numerical simulation plays an important role in analysis and design of shallow ground heat exchangers [26] [27] [28] [29]. The experimental results can be used to validate the numerical model, from which, other parametric analyses can be done. This is expected for eliminating the first mentioned consideration.

In addition, pressure-drop tests are also required to quantify the hydraulic resistance of the capillary mat heat exchanger.

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Conflicts of Interest

The authors declare no conflicts of interest regarding the publication of this paper.

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