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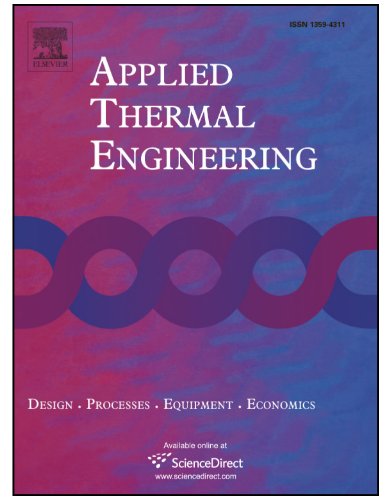

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A Novel Energy Pile: The Thermo-Syphon Helical Pile

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Abstract: This study focuses on assessing the heat transfer rate and structural stability of a novel self-operating energy pile based on the principle of a thermo-syphon. Specifically, this new energy pile, referred to as a "thermo-syphon helical pile" (THP), is formed by pressurizing a hollow helical pile with carbon dioxide (CO_2) to form a heat pipe, where spontaneous liquid-vapor phase change and natural convection inside the pile will facilitate self-operating heat transfer from the pile tip to the pile head. Based on the theories of heat transfer and fluid dynamics, a simplified analytical solution was developed to calculate the heat transfer rates within THPs with different geometries, which can be further converted into equivalent thermal conductivities. The results indicate that heat transfer within THPs is a function of the boundary temperature applied to the pile head, CO_2 pressure, working fluid properties, and THP geometry. The results also revealed that the equivalent thermal conductivity of the THP is 1,000 times higher than that of most metals due to the latent heat transfer of working fluid. An analysis of the structural stability of a THP under pressure indicates that bifurcation is not a problem if the ratio of the diameter to thickness of a THP is less than 90. While this analytical feasibility study demonstrates that THPs are a promising alternative for energy piles for both new and retrofitted buildings, future studies on soil-pile thermal and mechanical interaction under operational thermal gradients are needed to evaluate the range of heat transfer rates from the subsurface to a building when using a THP.

Key words: energy piles, helical piles, thermo-syphons, heat transfer, phase change, evaporation-condensation cycles

Nomenclature

A	area of liquid film
b_1	empirical constant, usually equal to 4.7
B	specific fluid flux (m^4/s^3)
C_p	specific heat capacity of CO_2 (kJ/$^oC\cdot$kg)
D	diameter of the THP (m)
E	modulus of elasticity (Pa)
L	length of the pile (m)
L′	distance from surface to constant temperature soil layer (m)
g	acceleration due to gravity (9.8 m/s^2)
$g_o^{'}$	effective acceleration of the fluid due to the density gradient (m/s^2)
h_e	heat transfer rate (kW)
h_v	latent heat (kJ/kg)
m_v	flow rate of CO_2 (kg/s)
ρ	density of CO_2 vapor (kg/m^3)
$\Delta\rho_o$	spatial variation of the density of CO_2 vapor (kg/m^3)
p	internal pressure (Pa)
p_p	dimensionless limit of bifurcation
Q	evaporative volume flux (m^3/s)
R	radius of the cylinder (m)
t	thickness of the pipe wall (m)
ΔT	temperature variation between evaporator and condenser sections (^oC)
ν	kinematic viscosity of the fluid (m^2/s)
w_m	vertical flow velocity (m/s)

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z vertical distance between THP head and the center of the liquid film (m)

1. Introduction

Due to the increase in energy consumption of buildings worldwide combined with regulatory restrictions on green-house gas emissions, energy piles have been increasingly used for the dual purpose of providing structural support for buildings and acting as subsidiary energy sources for building heating and cooling systems. The most common form of energy piles are cast-in-place reinforced concrete drilled shafts that incorporate ground source heat exchangers (GSHEs) to transfer heat between the subsurface and an overlying structure [1-8]. These energy piles must be used in tandem with a heat pump to circulate a working fluid through the closed-loop GSHEs and to exchange heat with the ground by controlling the temperature of the circulating fluid [9-11]. The heat transfer and associated thermo-mechanical interaction between energy piles and soil have been studied extensively in the past decade, confirming their feasibility for widespread implementation [2-4, 12-15]. In the past 20 years, the usage of energy piles has significantly increased in Europe and North America, especially in metropolitan areas. In addition to heating and cooling for buildings, they have been used to de-ice bridges in Northern Europe, Japan and the U.S. [11]. Although energy piles represent an appealing technology to harvest shallow geothermal energy, they require an external energy source to operate the circulation pump and heat pump [16]. The New England Geothermal Professional Association (NEGPA) estimated an operational cost (including annual maintenance costs) of $0.153/kWh for ground source heat pumps.

An alternative approach to improve the efficiency of the heat transfer is to utilize a two-phase thermo-syphon, where circulation of a pressurized working fluid is achieved by natural convection (i.e., a combination of gravity and buoyancy) without external energy for operation

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[4]. As no electrical and mechanical parts are needed to run the system, thermo-syphons are self-running and may be more reliable with lower maintenance costs when compared with existing energy piles [17-19]. Thermo-syphon systems are formed by pressurizing a hollow metal cylinder with a working fluid, making helical anchors the perfect candidate to be converted into a self-running energy pile referred to as a thermo-syphon helical pile (THP) hereafter. This study introduces the concepts of THPs and evaluates the feasibility of implementing THPs with different geometries in different ground conditions into practice. Specifically, this study presents a simplified analytical framework to estimate the heat transfer rate from the tip to the head of the THP and a structural stability analysis to assess the impact of the pressurized working fluid on THP bifurcation. Although this study focuses on a building heating scenario involving heat extraction from the THP, the thermal and mechanical mechanisms discussed in this study are applicable to both building heating and cooling.

2. Proposed Thermo-syphon Helical Pile (THP) Concept

The definition of a thermo-syphon in physics is a system that employs natural convection to circulate a working fluid without a mechanical pump to achieve passive heat exchange. Since their first appearance in the 1800's, thermo-syphons have evolved significantly over the past two centuries [20]. Nowadays, two types of thermo-syphons are commonly in use: single- and two-phase syphons. Single-phase thermo-syphons rely on the density difference between hot and cold fluids of the same phase to drive natural convection and realize heat exchange [21]. In contrast, two-phase thermo-syphons utilize thermal and density gradients of the vapor and liquid phases of a fluid to spontaneously circulate the working fluid to transfer heat over longer distances. Two-phase thermal syphons are commonly used in cooling systems for computer central processing

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units (CPUs), soil freezing, solar water heating, neutron cooling for nuclear research, spacecraft and satellite cooling, and cooling/heating of foods [22-26].

Helical piles, sometimes called screw piles/anchors, are a popular ground piling and anchoring system to support buildings. Helical piles are typically composed of sections of hollow steel pipe with helical plates/blades welded to the exterior surface of selected sections. The installation of a helical pile is similar to advancing a screw into a piece of wood, primarily relying on torque. Invented in the 1830's, helical piles have become an important foundation alternative as they are suitable for 80 to 90% of the soil conditions encountered in construction. An advantage of helical piles is that they can conveniently be used for both new construction and foundation retrofitting [27].

THPs are formed by converting a vertically-oriented or inclined, closed-ended steel helical pile into a heat pipe by pressurizing carbon dioxide (CO_2) into the hollow annulus of one of the steel pipe sections. CO_2 is selected over other fluids used in heat pipes (e.g., ammonia, Freon, propane, etc.) in THPs because of its non-toxic and non-flammable nature [28, 29]. CO_2 under a pressure ranging from 3 to 5 MPa will exist in both liquid and vapor phases within the annulus of the section of the helical pile and will circulate due to natural convection associated with concurrent phase change and density gradients induced by the temperature difference between the tip and head of the THP. A conceptual drawing of the proposed THP is shown in Fig. 1. The THP includes three sections along its length: a condenser section, an adiabatic section and an evaporator section. During heating of a building, a cold fluid will be circulated through a passive heat exchanger at the head of the pile, making the head temperature become lower than the tip

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temperature. In response to this temperature gradient, CO_2 liquid will absorb heat from the tip of the pile, evaporate, and rise to the head of the pile due to buoyancy (i.e., warmer fluids have lower density), carrying heat from the subsurface by convection. When the warm CO_2 vapor reaches the colder head of the THP, it will condense and release latent heat, resulting in energy transfer from the tip to the head of the THP and warming the fluid circulating through the passive heat exchanger. Afterward, liquid CO_2 flows back to the tip due to gravity, absorbing heat from the subsurface and vaporizing again to repeat the cycle. In this building heating scenario, the head of the THP with a heat exchanger attached acts as the condenser section, while the tip filled with working fluid acts as the evaporator section. In most geographic locations, the soil temperature is considered constant below a depth of 5 m in the absence of an upward geothermal gradient. Thus, the section of the pile below 5 m can be considered as the evaporator section. In theory, an adiabatic section in the middle portion of the THP can be assumed where minimal heat transfer occurs. In reality, the natural ground temperature fluctuations in the upper 2-3 m from the ground surface may affect the heat transfer processes and the length of the adiabatic section is likely to be shorter unless insulation is implemented in this depth range. The self-running evaporation-condensation cycle will continue as long as the tip of the THP is warmer than its head, and if an appropriate pressure is maintained within the closed pipe.

THPs can also be used to provide building cooling by including a wicking material within their inner annuli to convey liquid CO_2 upward via capillary action, as shown in Fig. 2(a). An analogous natural process is the upward transport of liquid water within tree trunks through xylem micro channels. Wicking structures have been used in many heat pipes in appliances to achieve both heating and cooling [30-34]. Wicking structures in heat pipes are typically

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constituted of fabric or sintered metal with micro-pores ranging from 1 to 100 μm. The capillary rise of working fluids can reach up to tens of meters in the vertical direction against gravity [35-38]. A combination of heating and cooling capabilities would enable the long-term functionality of THPs, especially at geographic locations that have a balanced cooling and heating demand. In addition, the wicking structure inside a THP can further increase the heat transfer efficiency during heating by providing a larger contact area. This study focuses primarily on the behavior of THPs under building heating conditions, shown in Fig. 2(b), as this will constitute the simplest configuration of a THP, but the heat transfer processes can be readily extended to building cooling conditions.

Because a THP is an integration of a helical pile and a heat pipe used for both structural support and heat transfer, the geometry and arrangement of the THPs can be configured to reach different structural and heat transfer goals. The dual purpose of THPs takes advantage of the installation of helical piles to reduce the installation costs of geothermal heat exchange systems as the added cost of converting a helical pile to a THP is negligible compared to the overall cost of a helical pile and construction. More importantly, a THP can successfully circumvent the spatial conflict if a foundation system and ground source heat exchange system are constructed separately, which is important for urban areas. Helical piles are typically installed in 3 to 5 m-long sections of pipe using a hydraulic rotary driver, with the pipe sections bolted together to reach the desired level of penetration to meet the structural support requirements of the building. Helical blades are typically installed only on the lowest pipe section. When forming a THP, only one of the pipe sections needs to act as the CO_2 pressure vessel, connected via flexible tubing through the annuli of other sections. After installation into the ground, the pressure vessel will be vacuum

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evacuated and filled with pressurized CO_2. Connections at the THP head can be incorporated so that the working fluid pressure can be adjusted to alter the heat transfer characteristics of the system during operation.

3. Status of Research and Applications of Thermo-syphons

Even though thermo-syphons have been widely used in many applications with the basic physical principles of thermo-syphons established as early as 1800's; their complexity has led to sustained research into different aspects of heat transfer [24, 39-42]. Prior studies on thermo-syphons have employed laboratory model tests, numerical simulations, or simplified analytical models to investigate the thermodynamics and fluid mechanics aspects needed to understand the heat transfer mechanisms of thermo-syphons.

Larkin [43], Clements and Lee [44], Shiraishi et al. [45], Park et al. [42], Noie [46], Jiao et al. [47] investigated the effects of geometry, working fluid characteristics, working fluid temperature ranges and operational pressures, fluid flow rates, and applied heat flux on the behavior of a thermo-syphon, and concluded that the temperature difference and fluid volume have the most significant effects on the performance of thermo-syphons. Huang and El-Genk [48] and Huang [49] defined operational parameters of thermo-syphons to estimate the liquid pool height during operation. Cohan and Bayley [50], Dobran and Casarosa [39], Gross [51], El-Genk and Saber [52], Terdtoon et al. [53, 54] and Xu [55] studied the operational limits (i.e., flooding limits, boiling limits, sonic limits, dry-out limits, and entrainment limits) of two-phase thermo-syphons under different configurations and found that two-phase thermo-syphon were likely governed by entrainment, boiling and dry-out limits. El-Genk and Saber [56] described the

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operational envelope of a two-phase thermo-syphon to delineate the extreme operational conditions. Without studying the detailed heat transfer process in THPs, Reed [57] and Reed and Tien [58] treated a thermo-syphon as a black box and developed a control volume method based on their experimental data, which can be used to evaluate the overall performance of thermo-syphons in similar applications. Rohsenow [59] studied the flow pattern of the liquid film in two-phase heat pipes, Chen et al. [60] and Spendel [61] considered the interfacial interaction between vapor and liquid in two-phase heat pipes, and Xu [55] examined the flow pattern and bubble formation in two-phase heat pipes. Even though there has been significant advancement in the application and design of thermo-syphons since their advent, especially after the 1950's, solutions to thermo-syphon problems remain largely empirical due to the complex fluid mechanics involved and the difficulty in monitoring their performance during operation [55, 62, 63].

In the 1950s, the US Army introduced thermo-syphons into geotechnical engineering as a passive subgrade freezing system to ensure warm permafrost soil to stay frozen in the warm summers [64, 65]. Warm permafrost soil usually has a temperature slightly below 0 ^{o}C but is susceptible to thawing in summer, resulting in settlements of pavements. By using thermo-syphons, the temperature of the warm permafrost layer can be further lowered in winter to enhance its thawing resistance in summer [66, 67]. Such soil stabilization applications, known as "refrigerated foundations", have been used in Russia, Canada, China, and Alaska for slab-on-grade buildings, petroleum pipelines, and railroads as shown in Fig. 3 [67-69]. It should be emphasized that these refrigerated foundation systems were not truly a structural foundation as proposed in this study but only heat transfer devices to freeze soil.

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Completed research on refrigerated foundation is limited and has focused on characterizing the operational conditions based on experimental studies or field monitoring [67, 70]. Without an established model, the usage of refrigerated foundations in soil was usually evaluated case by case without the development of design guidance. Thus, the design of refrigerated foundations relies heavily on past and local experience [70, 71]. Often, a multi-year monitoring is needed to ensure its performance [69, 72]. In addition, the existing refrigerated foundations typically are typically shallower compared with a THP and uses ammonia as working fluid that can be easily condensed under pressure but imposes a hazardous concern upon leakage. As a result, the past experience from refrigerated foundations may not be valid for a THP making it necessary to develop a simple model to assess the feasibility of utilizing CO_2 as a working fluid and to consider different embedment depths and geometries.

4. Analytical Framework for Heat Transfer in THPs

4.1 Heat transfer rate

A heat transfer analysis was performed for THPs based on the established theories of thermodynamics and fluid dynamics as well as some experimentally-identified phenomena reported for thermo-syphons. A schematic of a simple THP used in this analysis is shown in Fig. 4 and the adopted assumptions are listed below:

- The system is assumed to be in dynamic equilibrium - that is, the evaporation rate is equal to the condensation rate. In other words, this study investigates a steady state condition and not transient conditions. This dynamic equilibrium assumption is valid when studying the heat transfer in the THP at any given instant in time. This assumption

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will allow this study to focus on the evaporation flow and heat transfer but not the concurrence of evaporation and condensation. If the long-term functionality of a THP is to be assessed, the potential effects of imbalanced heating and cooling demands on the transient response need to be taken into account.

- The CO_2 pressure is assumed uniform inside the THP and stable. A number of experimental studies have shown that the pressure at the condenser end is only slightly lower than that in other locations within a closed-loop, two-phase thermo-syphon and a uniform pressure is usually assumed in analyses for simplification [24]. In the long term, the soil temperature will change due to extraction of heat from ground, which will lead to a decrease in the pressure of the CO_2 inside a THP. However, the soil temperature may not change rapidly when the temperature difference between the head of the THP and the ground is small.
- The filling ratio, equal to the volume of liquid working fluid to that of the evaporator section, is assumed to be 0.5, which means that half of the evaporator section is filled with a liquid CO_2 pool. In this study, the filling ratio refers to an equilibrium status, i.e., when the heat transfer equilibrium is reached. The specific filling ratio of 0.5 is made in this study to estimate the heat transfer area for the evaporator section of the THP, although the filling ratio in THPs may be varied between 0.2 and 0.8 depending on geometry, pressure, temperature, and type of working fluid used to achieve the desired heat transfer conditions. Several studies have shown that the heat transfer mainly occurs near the liquid film rather than the liquid pool [56, 66], and in this study the heat transfer area is the total evaporator area less the area in contact with the liquid pool.

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- As the subsurface temperature is approximately constant below a certain depth from the surface, it is assumed that the THP section below a depth of L´ functions as the evaporator section, as shown in Fig. 4. The value of L´ used in an analysis will vary depending on the ground thermal properties and climate setting. A typical value of L´ is 5.0 m.
- Initial fluid flux momentum is assumed to be negligible. This is a reasonable assumption because the evaporation primarily occurs at the liquid film that has zero or nearly zero vertical velocity. In addition, for energy piles, the temperature difference between the condenser and evaporator sections is small compared with other thermo-syphon applications, so the possible turbulence of working fluid is assumed to be negligible.
- Evaporation is assumed to primarily occur at the liquid film, so evaporation from the liquid pool is insignificant compared to evaporation from the liquid film [56].

According to the above-mentioned assumptions, heat transfer between the condenser and evaporator section is assumed to occur instantaneously and thus represent steady-state conditions.

As the circulation of the working fluid within a THP is driven by buoyancy due to temperature effects on the density of the CO_2 vapor, the vertical flow velocity, w_m, (unit: m/s) can be estimated by the equation for a plume with cylindrical boundaries, as follows [73]:

$$w_m(z/B)^{1/3} = f(B^{1/3}z^{2/3}/v) \tag{1}$$

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where z is the vertical distance between THP head and the center of the liquid film at the condenser [i.e., for a depth to a constant ground temperature of L´ from the ground surface, z=(L+3L´)/4, where L is the length of the pile] (unit: m), B is the specific fluid flux due to buoyancy (unit: m^4/s^3), and ν is the kinematic viscosity of the fluid (i.e., CO_2 vapor in this study) (unit: m^2/s). The fluid flux due to buoyancy can be calculated as follows [73]:

$$B = g_o^{'}Q \tag{2}$$

where g'_o is the effective acceleration of the fluid due to the density gradient (unit: m/s^2) and Q is the evaporative volume flux (unit: m^3/s). The effective acceleration of the fluid due to the density gradient of CO_2 can be calculated as follows [74]:

$$g_o^{'} = g(\Delta\rho_o/\rho) \tag{3}$$

where g is the acceleration due to gravity, ρ is the density of CO_2 vapor (unit: kg/m^3), and $\Delta\rho_o$ is the spatial variation of the density of CO_2 vapor (i.e., the density difference of CO_2 vapor between the condenser and evaporator sections).

Considering that the vapor flow is highly turbulent with a Reynolds number much greater than 4,000 [24, 75], the effect of the viscosity can be neglected. Accordingly, Equation 1 can be simplified as follows:

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$$w_m = b_1(B/z)^{1/3} \tag{4}$$

where b_1 is an empirical constant. The length of any THP is at least 5 m and diameter is usually no more than 0.3 m and the length/diameter ratio is at least 17, which makes longitudinal flow dominate and cross-sectional flow negligible, leading to a 1D flow problem. A 1D flow scenario occurs if the evaporator section is assumed to be a point source for which Rouse et al. [76] found that b_1 is equal to 4.7.

For the proposed THP, the vertical flow velocity of the CO_2 (w_m) can be estimated by:

$$w_m = b_1(B/z)^{1/3} = 4.7\left[\left(g\left(\Delta\rho_o/\rho\right)\right)(Q/z)\right]^{1/3} \tag{5}$$

where $= w_m A$, A is the area of liquid film. The equation can be further rendered to:

$$w_m = 10.2\sqrt{(g\Delta\rho_o A)/(\rho z)} \tag{6}$$

For the example THP in Fig. 4, the evaporator section is assumed to be located at a depth, L´ below the ground surface, so the area available for heat exchange with soil is $A = \pi D(L - L')/2$, L and L´ are the length of pile and distance from surface to constant temperature soil layer, respectively. Consequently, the heat transfer rate (h_e) will be:

$$h_e = m_v h_v + m_v C_p \Delta T = 10.2(h_v + C_p\Delta T)(\pi D(L - L')/2)^{\frac{3}{2}}\sqrt{(4g\rho\Delta\rho_o)/(L + 3L')} \tag{7}$$

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where m_v is CO_2 flow rate, $m_v = Q\rho = w_m A\rho$; h_v and C_p are the latent heat (unit: kJ/kg) and specific heat capacity of CO_2 (unit: kJ/($^oC \cdot$kg)), respectively.

According to Eq. (7), the heat transfer is a function of many factors such as the THP geometry, the operational pressure (as CO_2 vapor density depends on pressure), and the temperature difference. Crucial relationships between temperature and pressures are described by the CO_2 phase diagram shown in Fig. 5. At a certain operational pressure, P_1, the CO_2 will be in the vapor and liquid phase zones in the evaporator and condenser at their initial temperatures (i.e., T_1 and T_o), respectively. When heat transfer starts, the tip temperature (i.e., evaporator section) starts at a temperature T_1, assumed to be 15 °C (288 K) for the case that the ground temperature is 15 ^oC at a depth of 5 m or deeper, and gradually drops to T_2 when heat is extracted from the soil. Meanwhile, the head of the THP (i.e., condenser section) starts at a temperature T_o and gradually increases toward T_2 when heat is absorbed, as shown in Fig. 5. Whenever the temperature at the head or tip of the THP reaches T_2, the heat transfer will be halted because either evaporation or condensation ceases. Temperature T_2 is the critical temperature of a THP, which depends on the operational pressure, P_1. By increasing the operational pressure of the working fluid, the critical temperature can be increased to T_2' as shown in Fig. 5. Ideally, the operational pressure, P_1, should be lower than 5 MPa, which is the pressure of saturation liquid and vapor CO_2 at 15 ^oC. To improve the efficiency of a THP, P_1 should be determined in a way that the temperature of the head and tip reach T_2 at nearly the same time. For an instance, if T_2 is selected to be 10 ^oC, i.e., the operational pressure should be 4.5 MPa (45 bar), so the condenser section operates at a temperature $T_o \sim 10$ ^oC and evaporator section operates at a temperature of

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10 ~ 15 ^{o}C. It is worth emphasizing again that Eq. 7 represents heat transfer within the THP at any instant moment and can be used in the preliminary design to estimate the heat transfer at possible steady stages under different configurations.

To validate the derived formula, Eq. 7 was used to predict the results in a published experimental study, i.e., Fujita et al. (1988) [77]. Heat transfer in a narrow, constraint space was studied using distilled water as a working fluid under different cross-sections and temperatures. The evaporator was formed by two 30 × 30 mm^3 metal plates with varying gaps. The comparison of the heat transfer between calculation and experiments is provided in Table 1. It can be seen that Eq. 7 yields reasonably good predictions of the test results albeit with the calculated values consistently higher than the experimental data, perhaps because in the test, the metal plate resulted in some thermal resistance that could not be eliminated during testing. Nonetheless, the comparison in Table 1 confirms that Eq. 7 can accurately predict parametric effects for the heat transfer in thermo-syphons when the temperature difference between condenser and evaporator is small and the flow is primarily one-dimensional.

Since the derivation of Eq. 7 was based on heat transfer phenomenon inside a thermo-syphon, the simplified model presented in this study has a broader application potential as it is independent of surrounding or capsulation materials and can be used to evaluate thermo-syphons made from different materials. However, the derivation of the model presented in this study is based on the plume theory so further validation is needed to assess the boundary effect of the cylindrical space not accounted for in the model if the travel distance of fluid is very long.

4.2 Bifurcation

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The thermo-mechanical response of THPs is an important issue to consider, and it is expected that they will have similar behavior to drilled shaft energy piles due to the similar coefficient of thermal expansion of steel compared to concrete. However, another possible failure mode unique to THPs is bifurcation or "shell buckling", which occurs in thin-wall cylindrical vessels under combined axial compression and internal pressure. The internal pressure causes an expansion of the pipe diameter and, possibly, results in different degrees of wrinkles under axial compressive forces as shown in Fig. 6 [78, 79]. This deformation may lead to sudden collapse of the pipe if it fails to sustain the deformation. According to Lo et al. [80], Fung and Sechler [81] and Paquette and Kyriakides [82], low internal pressures will help the pipe wall to maintain an upright position. However, when the internal pressure exceeds a certain limit, it facilitates the propagation of wrinkles and causes bifurcation failure. Lo et al. [80] and Fung and Sechler [81] used the following equation to define a dimensionless limit of bifurcation, p_p:

$$p_P = \left(\frac{p}{E}\right)\left(\frac{R}{t}\right)^2 \tag{8}$$

where p is internal pressure, E is modulus of elasticity, R is radius, and t is the thickness of the pipe wall. Fung and Sechler [81] suggested that the dimensionless limit of bifurcation p_p should be limited to 0.2 to minimize the likelihood of bifurcation under the applied internal pressure and axial compression.

5. Results and Discussion

5.1 Heat transfer rate and thermal conductivity

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With the target application of helical piles for building heating, the heat transfer rate and equivalent thermal conductivity under various configurations are calculated and discussed in this section. The equivalent thermal conductivity is defined in this study as the heat transfer rate per unit length of pile per unit temperature difference between the evaporator and condenser sections, which can be estimated from Eq. 7. The prototype THP is selected with the following parameters as listed in Table 2, which account for the typical geometry of a helical pile as well as the practical temperature variation range. The tip temperature was assumed to be 15 ^oC to represent an initial condition. The temperature difference between the tip and head was selected to ensure the CO_2 would be located in liquid and vapor phase zones, respectively, at the operational pressure. The parameter used for each parameter representing the baseline case is asterisked in the table. The equivalent thermal conductivity of the THP was calculated based on the length and temperature difference of each THP configuration.

The heat transfer rates per unit length for THPs with different diameters under a range of temperature differences are shown in Figs. 7, 8 and 9. The results indicate that the heat transfer rate of a THP increases with the THP diameter or temperature difference. In addition, compared with the effect of temperature, the effect of diameter seems much more salient, which is explainable. According to Eq. 7, the heat transfer rate is related to $D^{3/2}$, while the effect of temperature on the heat transfer rate is mainly associated with the specific heat capacity, i.e., $\Delta T \cdot C_p$, which is insignificant compared with latent heat. It can be easily asserted that the heat transfer rate primarily depends on latent heat as the ratio of $h_v : C_p\Delta T$ is always greater than 4.0 for CO_2 if the temperature difference is no greater than 25 ^oC. In other words, the phase change

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from vapor to liquid leads to more heat transfer than simple conduction. Accordingly, heat loss between the condenser and evaporator sections can be reasonably assumed minimal.

Considering various temperature differences between the evaporator and condenser, the calculated equivalent thermal conductivity is more than 1,000 times of steel, which confirms the findings from other studies [18, 55, 58, 66, 83]. As the performance of thermo-syphons depend on many factors, the published experimental data in the literature on thermal conductivities cannot be used directly to validate this analytical solution. However, the calculated value of this study is highly consistent with the published experimental values of equivalent thermal conductivities, which were reported to be $10^3 \sim 10^4$ greater than most metals depending on geometry, working fluid, etc. [24, 55]. Based on the results, the equivalent thermal conductivity decreases with the temperature difference increases. By referring to Eq. 7, the equivalent thermal conductivity is inversely related to the temperature difference when heat transfer rate was divided by the temperature difference when converting from heat transfer rate to thermal conductivity. Accordingly, the larger diameter leads to higher equivalent thermal conductivity, which is intuitively correct as a larger THP holds more working fluid. It must be clarified that the heat transfer rate and equivalent thermal conductivity only apply to the THP itself and the heat transfer between soil and a THP is not considered in this study. This study implies that the transfer of heat from the ground to the building will be dominated by heat transfer in the soil, and that the THP will provide negligible impedance to heat transfer, which is different from both conventional drilled shaft heat exchangers and conventional GSHEs.

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The results in Figs. 7, 8 and 9 reflect the effect of an operational pressure of the working fluid on the heat transfer rate of a THP and emphasize that an increase in operational pressure may have some benefits up to a certain point, after which the benefits dissipate due to the nonlinear relationship between heat transfer rate and operational pressure. For example, when the CO_2 pressure increases from 3.9 to 4.2 MPa, the heat transfer rate increases by more than 200%. However, when the pressure increases from 4.2 to 4.5 MPa, the heat transfer rate increases by approximately 20%. The effect of operational pressure of the working fluid on the heat transfer rate lies mainly in its impact on CO_2 vapor density. With a higher operational pressure, the density of CO_2 vapor is greater. Under an operational pressure between 3 and 5 MPa, the CO_2 is within the transition zone from vapor to liquid and cannot be treated as an ideal gas as indicated in Fig. 5. The phase diagram in Figure 5 confirms that the effect of operational pressure on the density of CO_2 is highly nonlinear, and that it is not always effective to increase heat transfer rate by increasing operational pressure.

Nonetheless, the CO_2 operational pressure controls the total harvested energy through its governance on the critical temperature T_2 as marked in Fig. 5. Theoretically, the operational pressure of the working fluid should be selected such that the evaporator and condenser sections reach T_2 at the same time. This implies that both evaporation and condensation will cease at the same time. Consequently, the determination of an ideal operational pressure would require a complete heat transfer analysis of a THP that also includes heat exchange between the soil and the tip of the THP for a given heat extraction rate applied to the THP head. A deeper study on the impact of heat transfer from the soil to a THP is required to fully optimize the operational pressures of the working fluid used in THPs.

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The results in Fig. 10 indicate that the length of a THP has a significant effect on the heat transfer rate. Longer THPs result in a higher heat transfer rates due to a larger area for heat transfer in the evaporator section (i.e., the depth below 5 m for this study). In addition, a longer THP means more working fluid if the filling ratio is kept constant. These two factors contribute to enhanced heat transfer. For THP lengths of 6, 8, and 10 m, the contact length for CO_2 evaporation is 0.5, 1.5, and 2.5 m (i.e., (L-5)/2), respectively. The results in Figure 10 indicate that the increase of heat transfer rate appears to be approximately proportional to the increase of contact length, i.e., 1:3:5. This phenomenon is supported by Eq. (7), which indicates a nearly linear correlation between heat transfer rate and contact length. If a THP is very long (L >> 5 m), the heat transfer rate and total length L are approximately linearly related.

Evaluation of the different cases in the parametric analysis indicates that the equivalent thermal conductivity of the THP is positively related to operational pressure of the working fluid and the length and diameter of the THP. However, the equivalent thermal conductivity is inversely related to the temperature difference between the tip and head of the THP.

5.2 Bifurcation

Considering a possible operation pressure ranging from 3 to 5 MPa, which corresponds to critical temperatures equal to -5 to 15 ^{o}C, respectively, the maximum allowable radius to thickness ratio is plotted in Fig. 11 according to Eq. 8. The maximum allowable R/t ratio decreases approximately linearly with increasing operational pressure of the working fluid. According to Eq. 8, the R/t is quadratically related to the ratio of operational pressure of the working fluid to

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the steel modulus of elasticity. However, as R/t and P/E are both significantly greater than 1, the quadratic relationship regresses to an approximately linear relationship. Under the worst-case bifurcation scenario, the highest possible operational pressure of 5 MPa, which corresponds to a soil temperature of 15 ^{o}C before energy is extracted, the R/t should be less than 90. In other words, if the R/t is less than 90, bifurcation should not be a concern under any operational condition for THPs. Such a limit on the R:t ratio covers most helical piles available in the market, so bifurcation does not need to be considered in the design of THPs in most of cases.

6. Conclusions

This study demonstrates that THPs are a viable alternative energy pile concept and provides a simple analysis framework to assess their heat transfer rate and structural stability for a given geometric configuration and operational pressure of the working fluid. The results of the heat transfer and structural analyses performed in this study indicate that THPs may serve well as both foundation elements and heat transfer devices. The following conclusions can be drawn from the analyses presented in this study:

- The heat transfer rate and equivalent thermal conductivity of a THP are a function of pile length and diameter, temperature difference, working fluid operational pressure and type of working fluid. Under possible operational pressures, THP geometry of a single stem, and temperature difference, a THP has an equivalent thermal conductivity at least 1,000 times greater than that of most metals, reflecting its advantage as a geothermal heat transfer element.
- The operational pressure of the working fluid (CO_2) is the most important factor among those considered in this study. It has a profound impact on not only the heat transfer rate but also the total amount of heat that may be harvested from a THP. Thus, the determination of the

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operational pressure should consider a system that includes soil, THP and an attached heat exchanger at the pile head.

- Bifurcation of helical piles is not expected under the expected range of CO_2 operational pressures if the ratio of the radius to thickness (R:t) of the THP is less than 90. This is satisfied by most helical piles available on the market.

These observations confirm that there is a solid theoretical basis for the proposed new energy foundation, which makes it a promising alternative for existing cast-in-place concrete energy piles. However, there are still several additional studies required before THPs can be implemented into practice as foundations for commercial or residential buildings, including:

- Cooling applications: this study focuses on heating application and future studies are needed to investigate building cooling applications and the associated flow processes of liquid CO_2 through a wicking material. Building cooling applications of energy piles have not received as much attention as building heating applications of energy piles [3, 13].
- The soil-THP interaction and its influence on the loading capacity and displacement of a helical pile: the operational pressure applied on the internal boundaries of a THP causes it to deform, which is resisted by lateral soil pressure and shaft friction at the soil-THP interface.
- Long-term performance of THPs: although thermo-syphons have been used successfully in applications involving permafrost dominated by heat extraction, building heating and cooling cycles are expected to alternate seasonally. Although THPs are expected to respond quickly to thermal gradients due to their high equivalent thermal conductivity, the long-term efficiency in response to fluctuations in ground temperatures and building heating and cooling demands should be explored in future studies.

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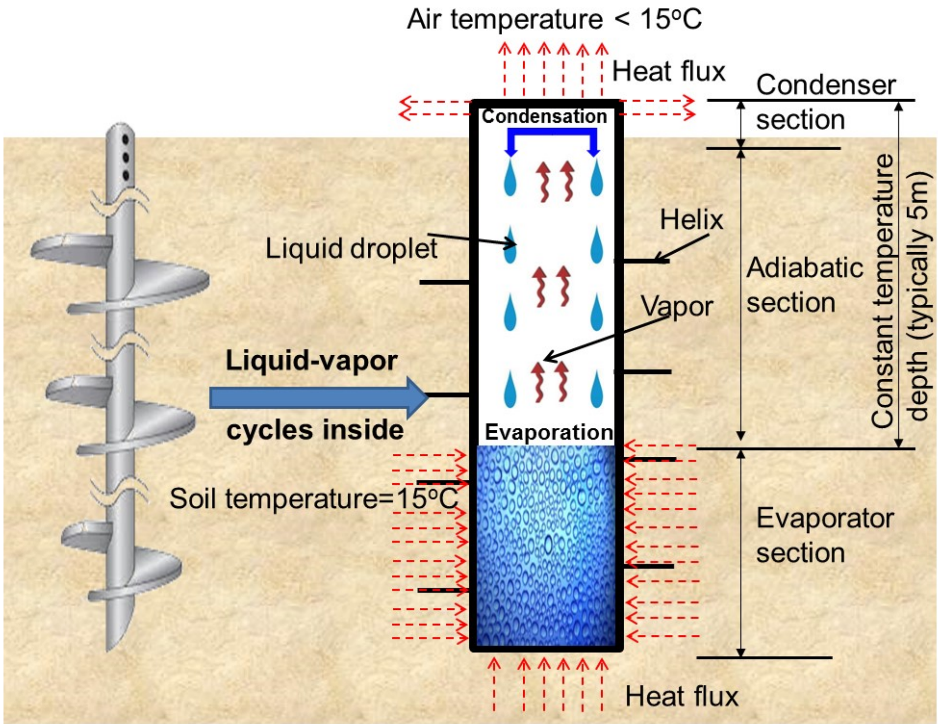

Fig. 1. Illustration of a THP in heating mode.

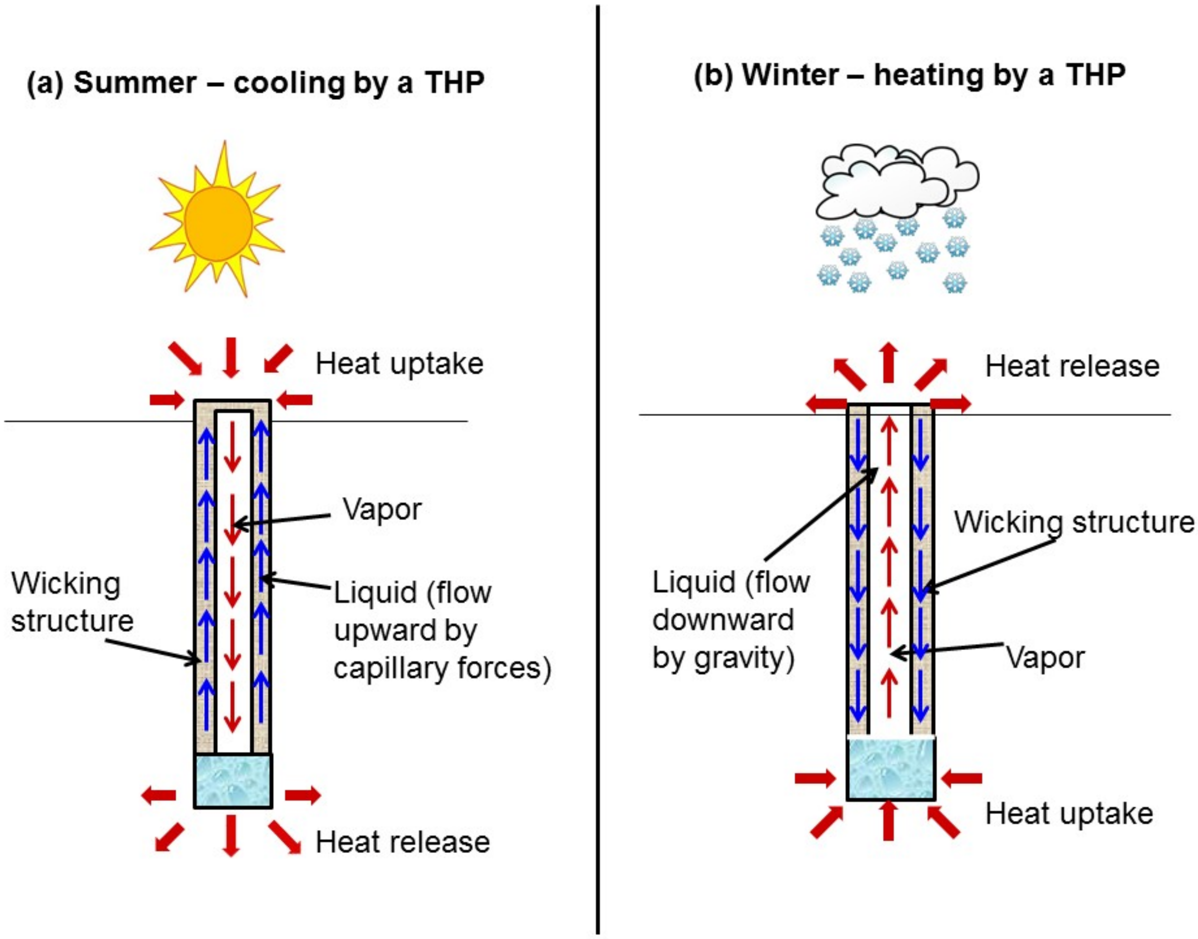

Fig. 2. Concept of a reversible THP energy pile: (a) cooling – fluid circulating by capillary; and (b) heating – fluid circulating by buoyancy & gravity.

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(a) (b)

Fig. 3. Thermo-syphon applications to maintain frozen soil conditions: (a) slab-on-grade building on permafrost in Alaska (modified from Jim Carlton/The Wall Street Journal), and (b) railroad throughout permafrost in Tibet, China (modified from China Railroad Academy).

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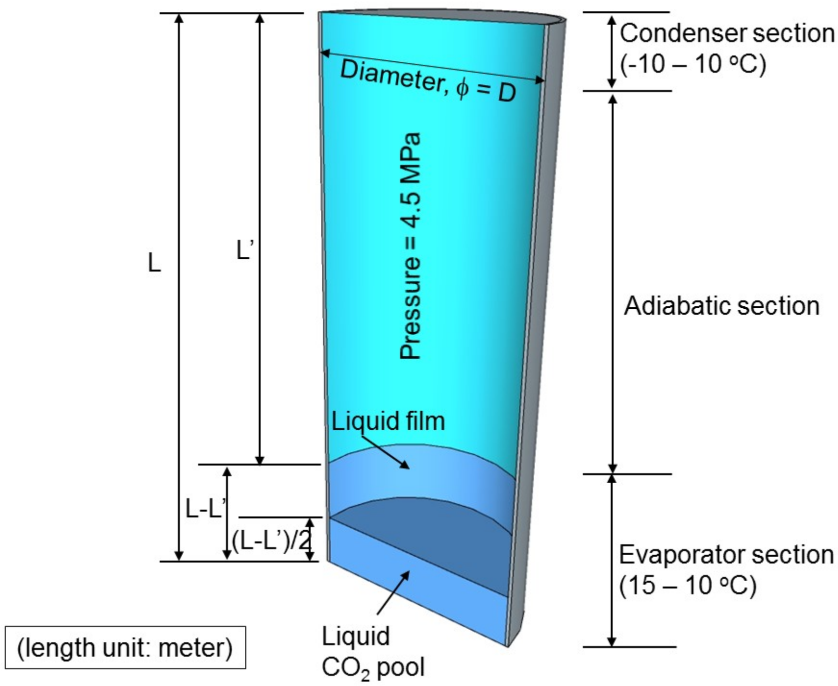

Fig. 4. Schematic of operation condition of an example THP.

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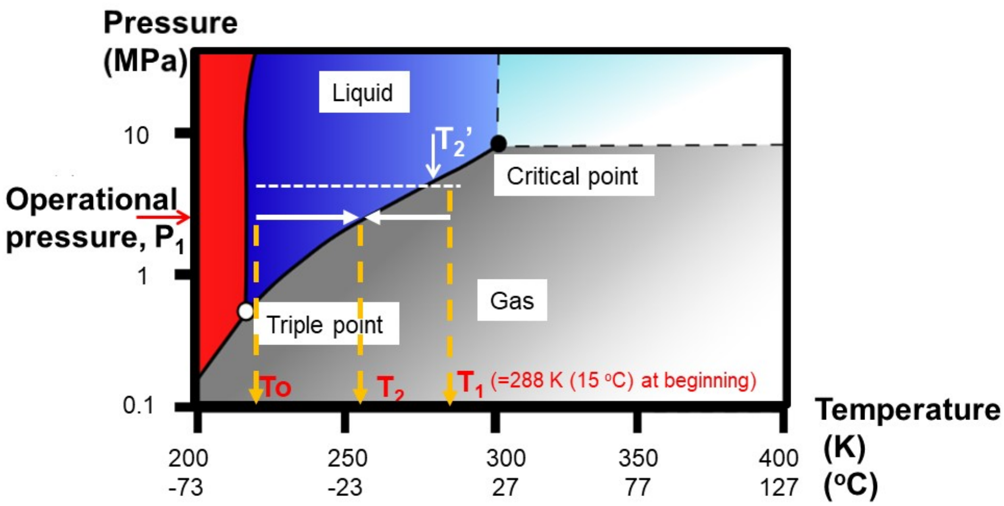

Fig. 5. Phase diagram of CO_2.

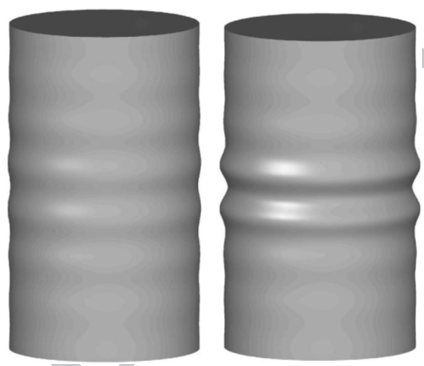

Fig. 6. Bifurcation of thin-wall pipes under compression (modified from Bardi et al.[75]).

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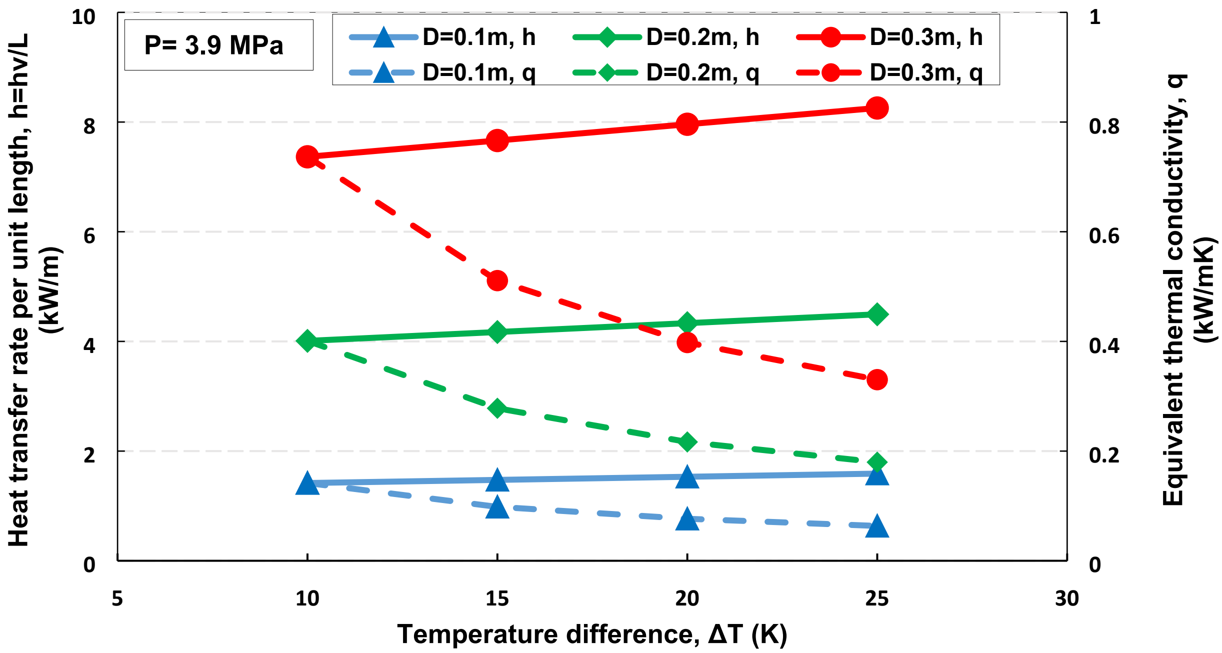

Fig. 7. Heat transfer rate per unit length and equivalent thermal conductivity for THPs with different diameters (P= 3.9 MPa, L= 8 m)

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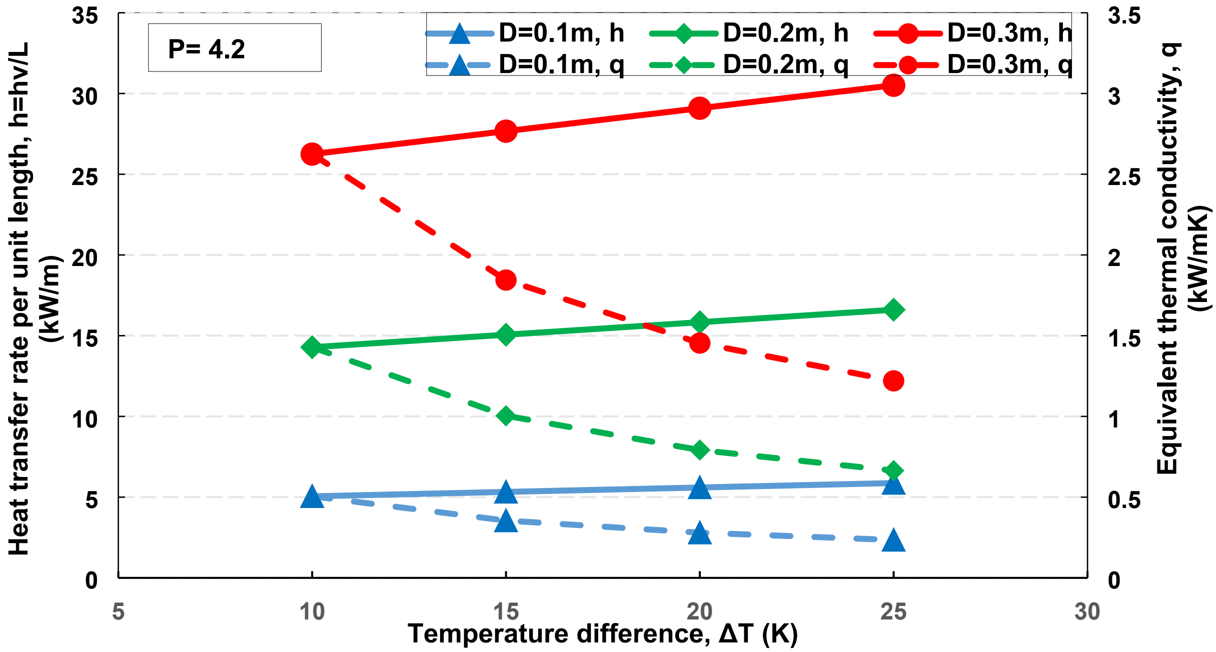

Fig. 8. Heat transfer per unit length and equivalent thermal conductivity for THPs with different diameters (P= 4.2 MPa, L= 8 m)

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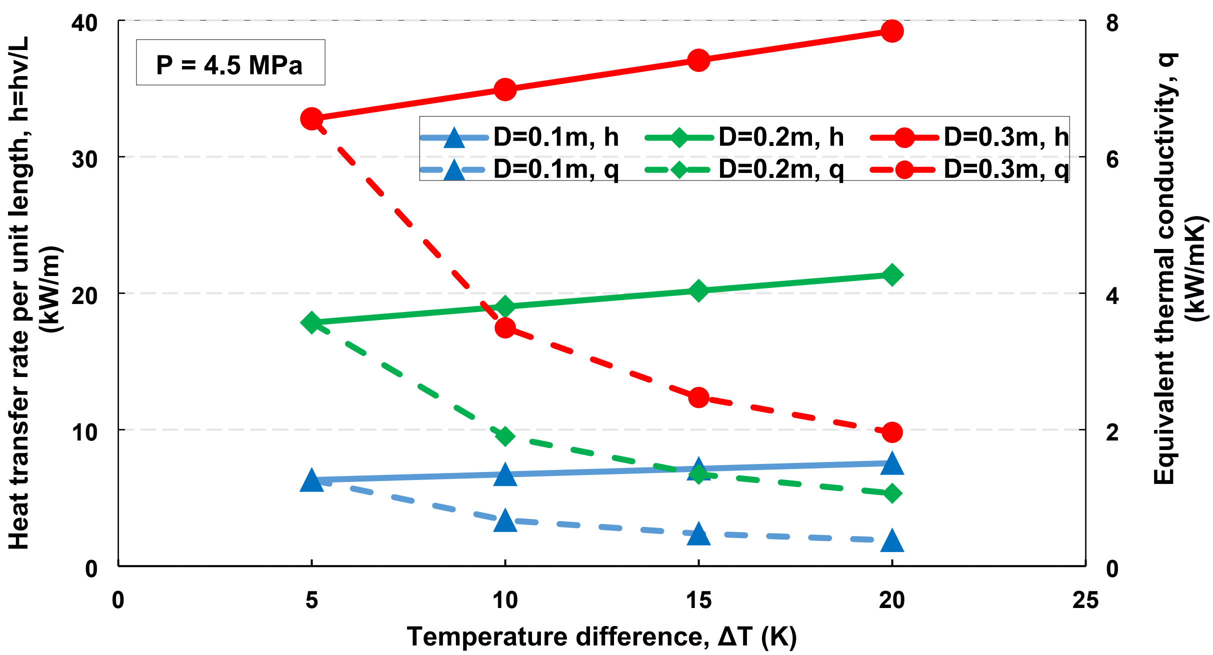

Fig. 9. Heat transfer per unit length and equivalent thermal conductivity for THPs with different diameters (P= 4.5 MPa, L= 8 m)

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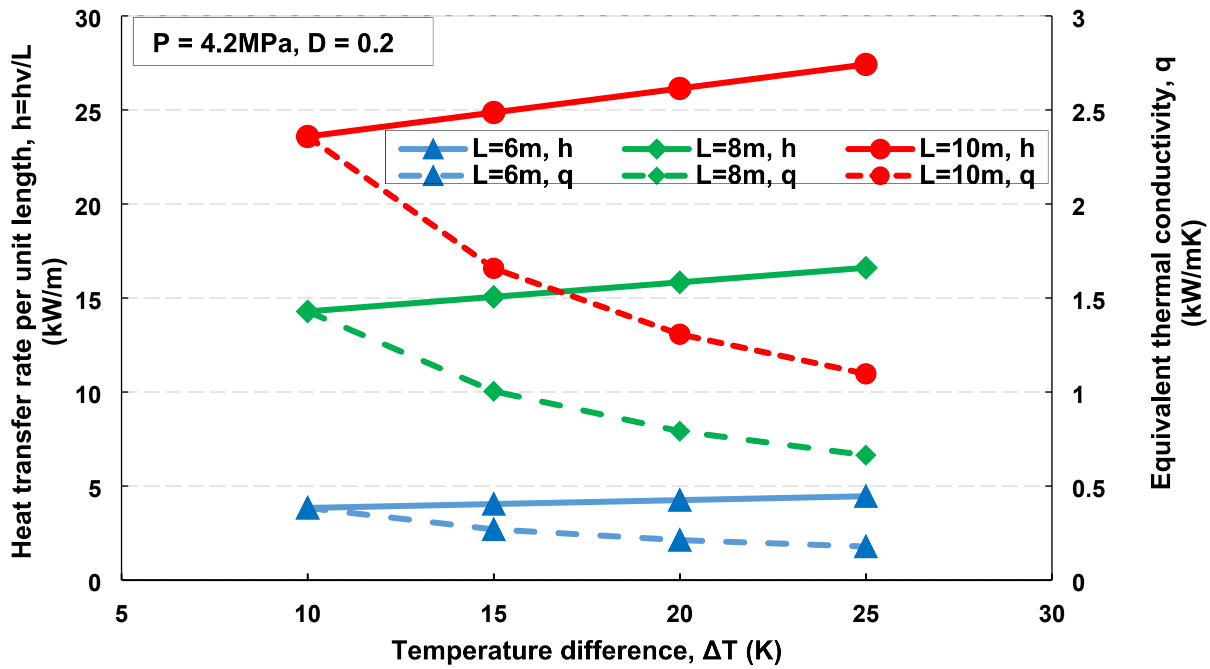

Fig. 10. Heat transfer per unit length and equivalent thermal conductivity for THPs with different lengths (P= 4.2 MPa)

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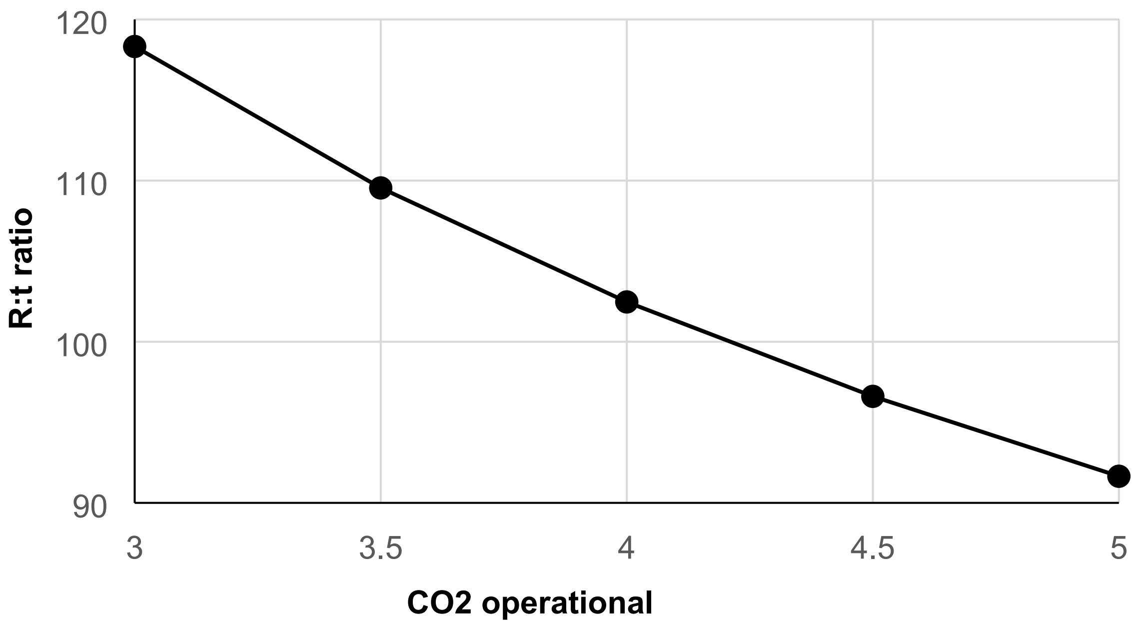

Fig. 11. Maximum allowable THP radius/thickness ratio (R/t) for different operational pressures

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Table 1

Heat transfer comparison

	Heat transfer (kW)					
Gap (mm)	$\Delta T = 5\ ^oC$			$\Delta T = 10\ ^oC$		
	Calculated	Experiment	Calculated/experiment ratio	Calculated	Experiment	Calculated/experiment ratio
5	83	65	1.28	121	105	1.15
2	77	68	1.13	112	108	1.03

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Table 2

Parameters of various THPs investigated in this study

Parameters	Value
CO_2 operational pressure (MPa)	3.9, 4.2*, 4.5 (corresponding to critical temperatures of T_2 = 5, 8, and 10 ^oC, respectively)
Temperature difference (^oC) between the THP tip and head	5, 10, 15*, 20, 25
THP length (m)	6, 8*, 10
THP diameter (m)	0.1, 0.2*, 0.3*

* Parameters used for the baseline case

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A Novel Energy Pile: The Thermo-Syphon Helical Pile

Jie Huang, John S. McCartney, Howard Perko, Drew Johnson, Chao Zheng, and Qingwen Yang

Highlights:

- A new type of energy pile is proposed based on the combined principles of a helical piles and a thermo-syphon to harvest shallow geothermal energy;
- The principles of thermodynamics and fluid dynamics governing the behavior of thermo-syphons were reviewed to understand the operational behaviors of the new energy pile.
- An analytical model was developed to assess the efficiency of heat transfer of the proposed energy pile;
- The internal stability of the new energy pile under operational pressure was confirmed to not be a major issue for the typical geometries of helical piles.

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