of grout, the design optimization, etc. (Delaleux et al. 2012, Selamat et al. 2016).

Phase change material (PCM) has the latent heat of fusion which is defined as the absorbed or released energy during solid-liquid phase transformation (Fleischer, 2015). This leads to the more thermal energy storage in the identical mass of PCMs compared with materials which only the sensible heat is used. PCMs are widely used for the thermal management and energy storage. However, relatively little attention was paid to the incorporation of phase change materials into as ground-source heat pump systems. Bottarell et al. (2015) investigated the performance of horizontal ground heat exchanger numerically when the micro-encapsulated PCMs were mixed directly with backfill material. They implied the potential increase in the coefficient of performance of the heat pump by showing the higher surface temperature in winter and lower in summer.

We are currently analyzing the effect of PCMs on the performance of horizontal ground heat exchanger when the panel form of PCMs is used. In addition, the thermal properties and various application forms of PCMs were investigated by a numerical study.

2. Numerical model

The numerical model was developed using COMSOL multiphysics which is a commercial FEM package. The following assumptions have been made for the present study:

- Container for PCMs was not considered.
- Properties of PCMs were identical in solid and liquid phases.
- PCMs were homogeneous and isotropic.
- The effect of natural convection was ignored.
- Ground was assumed as solid medium, not porous medium.

2.1 Governing equation

The generalized energy equation for a soil domain is given as follows:

$$\rho C_p \frac{\partial T}{\partial t} + \rho C_p u \cdot \nabla T + \nabla \cdot (-k \nabla T) = Q \tag{1}$$

where ρ is the density, C_p is the specific heat at a constant pressure, T is the temperature, u is the velocity, k is the thermal conductivity, and Q the is heat source.

The thermal properties of phase change material were followed in the apparent heat capacity formulation. The apparent heat capacity, effective thermal conductivity, and effective density are given as follows:

$$C_p = \frac{1}{\rho}\left(\theta \rho_{solid} C_{p,solid} + \left(1-\theta\right) \rho_{liquid} C_{p,liquid}\right) + L \frac{\partial \alpha_m}{\partial T} \tag{2}$$

$$k = \theta k_{solid} + \left(1-\theta\right) k_{liquid} \tag{3}$$

$$\rho = \theta \rho_{solid} + \left(1-\theta\right) \rho_{liquid} \tag{4}$$

where $\alpha_m = \frac{1}{2}\frac{(1-\theta)\rho_{liquid} - \theta\rho_{solid}}{\theta\rho_{solid} + (1-\theta)\rho_{liquid}}$ is the mass fraction, θ is the fraction of phase before transition, and L is the latent heat.

The generalized mass, momentum and energy equations for an incompressible fluid flowing in a pipe are given as follows:

$$\frac{\partial A\rho}{\partial t} + \nabla_t \cdot \left(A\rho u e_t\right) = 0 \tag{5}$$

$$\rho\frac{\partial u}{\partial t} = -\nabla_t p \cdot e_t - \frac{1}{2} f_D \frac{\rho}{d_h}|u|u + F \cdot e_t \tag{6}$$

$$\rho A C_p \frac{\partial T}{\partial t} + \rho A C_p u e_t \cdot \nabla_t T = \nabla_t \cdot \left(Ak\nabla_t T\right) + \frac{1}{2} f_D \frac{\rho A}{d_h}|u|u^2 + Q + Q_{wall} \tag{7}$$

where ρ is the density of fluid, u is the velocity of fluid, p is the pressure of fluid, e_t is the unit tangent vector to the pipe axis, f_D is the Darcy friction factor, d_h is the mean hydraulic diameter, F is the volume force term, Q is the general heat source, and Q_{wall} is the external heat exchange through the pipe wall.

Churchill (1997) proposed an empirical equation for the Darcy friction factor which could be used in all fluid-flow regime.

$$f_D = 8\left[\left(\frac{8}{\mathrm{Re}}\right)^{12} + (c_A + c_B)^{-1.5}\right]^{1/12} \tag{8}$$

where $c_A = \left[-2.457\ln\left((7/\mathrm{Re})^{0.9} + 0.27(e/d_h)\right)\right]^{16}$, $c_B = (37530/\mathrm{Re})^{16}$, Re is the Reynolds number, e is the surface roughness, and d_h is the hydraulic diameter.

The value of e was set as 0.00015 mm because the HDPE pipe was assumed as drawn tubing pipe. Nusselt number for internal turbulent forced convection was obtained using the empirical equation proposed by Gnielinski (1976).

$$Nu = \max\left\{\begin{array}{c} Nu_{lam} = 3.66 \\ Nu_{turb} = \dfrac{(f_D/8)(\mathrm{Re}-1000)\mathrm{Pr}}{1+12.7(f_D/8)^{1/2}\left(\mathrm{Pr}^{2/3}-1\right)} \end{array}\right\} \tag{9}$$

where f_D is the Darcy friction factor , Re is the Reynolds number, and Pr is the Prandtl number.

The external heat exchange through the pipe wall is given as follows:

$$Q_{wall} = (hZ)_{eff}\,(T_{ext} - T) \tag{10}$$

where $(hZ)_{eff} = \dfrac{2\pi}{1/r_{in}h_{\mathrm{int}} + \left(\ln(r_{out}/r_{in})/k_{wall,pipe}\right)}$, r_{in} is the inner radius of pipe, r_{out} is the outer radius of pipe, h_{int} is the heat transfer coefficient on the inside of pipe, and k_{wall} is the thermal conductivity of pipe wall.

2.2 Initial and boundary conditions

A three-dimensional finite element model having size 10 width×20 length×6 height m was considered in the present study. To make the simulation more reasonable, the properties of soil and pipe were obtained from the experimental data in the field. Input parameters were listed in Table 1.

Table 1. Properties used for three-dimensional finite element model

Properties	Unit	Materials		
		Soil	Pipe	PCMs
Density	kg/m^3	810	958	1000
Thermal conductivity	$W/(m\cdot K)$	1.8	0.38	0.2
Specific heat	$J/(kg\cdot K)$	1654	2250	1500
Latent heat	kJ/kg	-	-	200

Energy balance at the ground surface is defined as

$$G_0 - R + H + LE = 0 \tag{11}$$

where G_0 is the soil heat flux, R is the net radiative energy flux, H is the sensible energy flux, and LE is the latent energy flux.

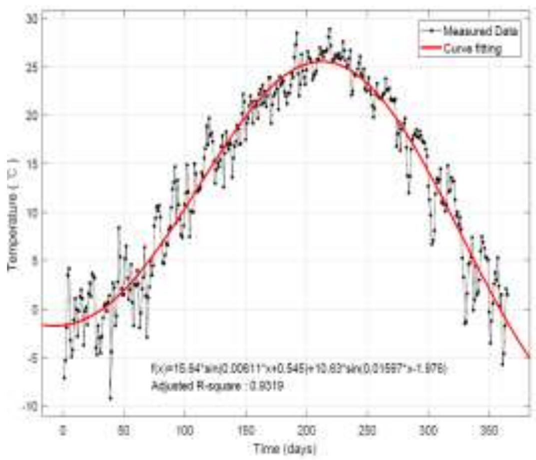
Bottarelli et al. (2015) obtained the net soil heat flux by comparison with measured and simulated data since enough data for estimating energy balance at the ground surface were not available. Jang et al. (2013) also applied the convection coefficient for the convective ground surface boundary for modeling soil freezing phenomenon. Similarly, the convective heat flux was used at the upper boundary in order to consider the temperature inside the ground which varied with depth and time.

The convective heat flux applied to the upper boundary of domain is follows

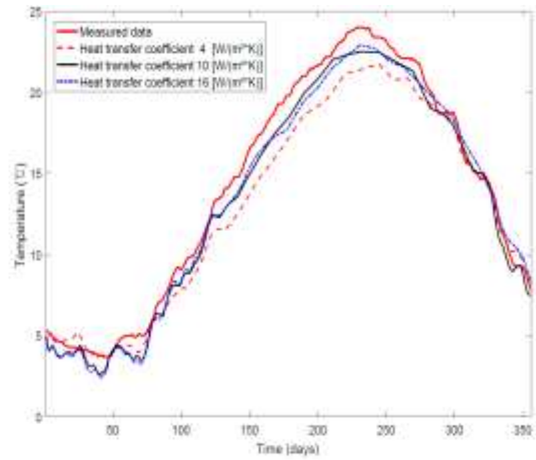
$$q = h\cdot(T_{ext} - T) \tag{12}$$

where h is the convective heat coefficient, T_{ext} is the external temperature, and T is the temperature of ground.

The external temperature was assumed as the air temperature. For obtaining an appropriate convective heat coefficient, the air and ground daily temperatures, which were measured by Korea Meteorological Administration, were collected. The collected data were measured from 1st of Jan to 31st of Dec in 2015 at Incheon city. Since the measured surface air temperature is affected by various factors such as wind, rain, location of measurement, etc., the temperature fluctuates over the day as shown in Fig. 1 (a). The dramatic variation of input data typically causes the numerical difficulties in the convergence process. For this reason, the fitted sine equation was obtained using the MATLAB by method of least squares. Initial condition of the ground temperature was set as the ground temperature measured in 1st of Jan in 2015. Fig. 1(b) shows the ground temperatures measured at the depth of 1.0m and the calculated ground temperatures for various convective heat coefficients. When the 10 $W/m^2 \cdot K$ of constant convective heat coefficient was used, the results had the good agreement with measured data. This result was also in good accordance with the study of Freitag and McFadden (1997) who proposed the range of free convection for air (6$W/m^2 \cdot K$ ~28$W/m^2 \cdot K$).



(a) Surface air temperature



(b) Ground temperature for measured and calculated data

Fig. 1 Surface air and ground temperature data used for simulation

All boundary except the upper boundary of domain was insulated. The velocity of inlet fluid was 0.5 m/s and the temperature with time was set as shown in Fig. 2. Fig. 3 shows the outlet fluid temperature with number of elements. The variation of temperature was reduced with increasing the elements. With the consideration of computation effort and accuracy of the solution, total number of 15,000 elements were considered for the present study.

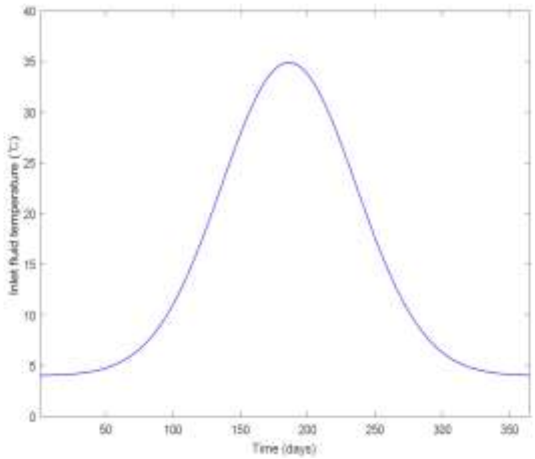


Fig. 2 Inlet fluid temperature with time

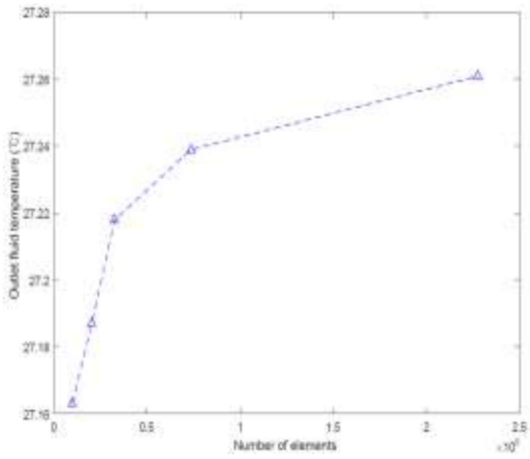


Fig. 3 Results of mesh convergence

3. RESULTS AND DISCUSSIONS

3.1 Effect of thermal properties of PCMs

The commercially available PCMs were considered to investigate the effect of thermal properties on the performance of horizontal ground heat exchanger. The thermal properties of PCMs were listed in Table 2.

Table 2. Thermal properties of commercially available PCMs

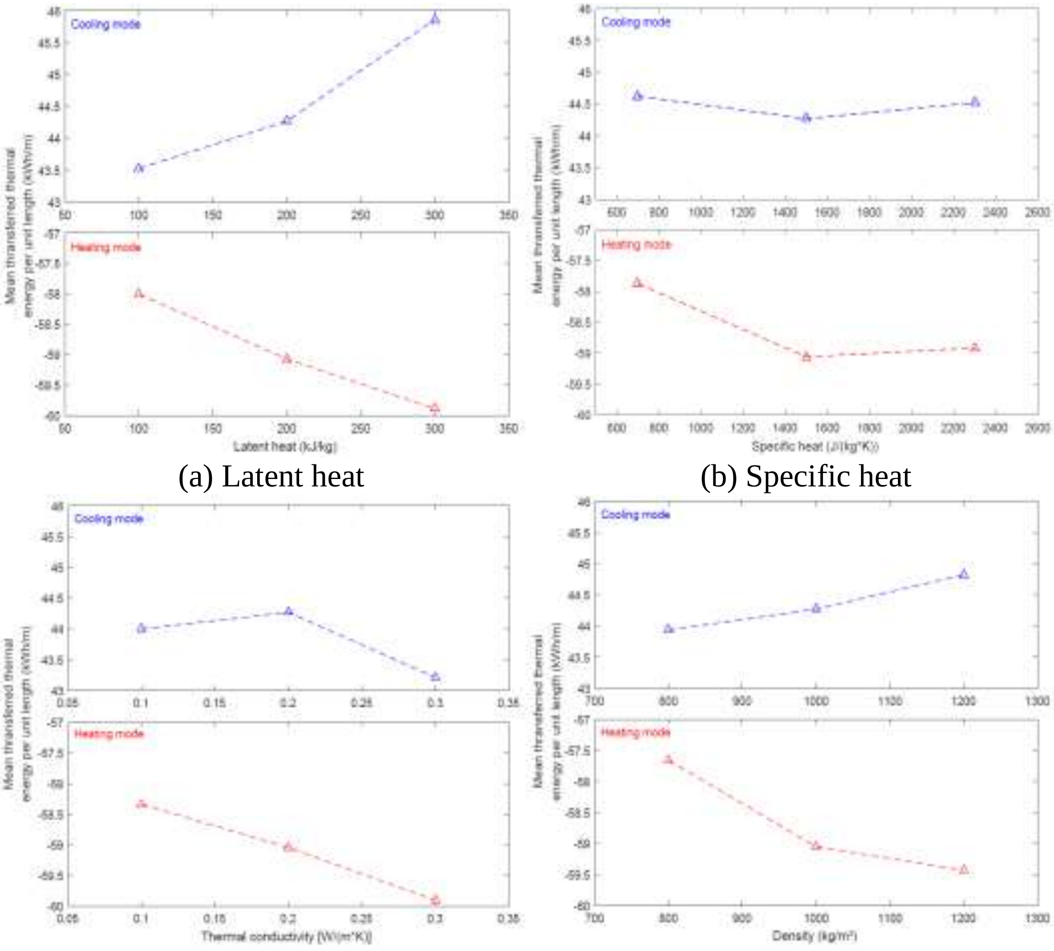
Properties	Unit	Values
Latent heat	kJ / kg	100, 200, 300
Specific heat	$J / (kg \cdot K)$	700, 1500, 2300
Thermal conductivity	$W / (m \cdot K)$	0.1, 0.2, 0.3
Density	kg / m^3	800, 1000, 1200

The performance of horizontal ground heat exchanger was quantified as the mean transferred thermal energy per unit length which is defined in Eq. (13).

$$\frac{Q}{L} = \frac{\dot{m} C_p \left(T_{inlet} - T_{outlet} \right)}{L} \qquad (13)$$

Fig. 4 shows the variations of mean transferred thermal energy per unit length with varying the thermal properties of PCMs. As the latent heat and density increased, the mean transferred thermal energy also increased in both modes. The mean transferred thermal energy had the highest value in a cooling mode, but the lowest value in a heating mode when the specific heat was 1500 J/(kg*K). However, the total mean transferred thermal energy increased with increasing the specific heat. The total mean transferred thermal energy with 800 J/(kg*K), 1500 J/(kg*K), 2300 J/(kg*K) of specific heat were 102.49 kWh/m, 103.33 kWh/m, and 103.44 kWh/m, respectively.

Correlation between the mean transferred thermal energy and thermal conductivity showed positive in the heating mode. In cooling mode, the highest value of mean transferred thermal energy was obtained when the thermal conductivity was 0.2 W/(m*K). Ranges of the total mean transferred thermal energy for the latent heat, specific heat, thermal conductivity, and density were 4.21 kWh/m, 0.94 kWh/m, 0.98 kWh/m, 2.66 kWh/m, respectively. It implies that the latent heat of PCMs have the strongest influence on the performance of horizontal ground heat exchanger.



(a) Latent heat

(b) Specific heat

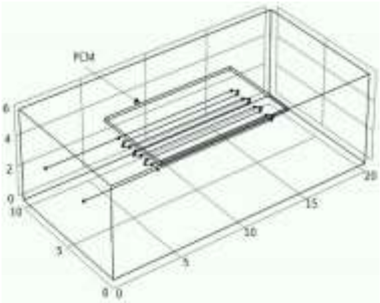
(c) Thermal conductivity

(d) Density

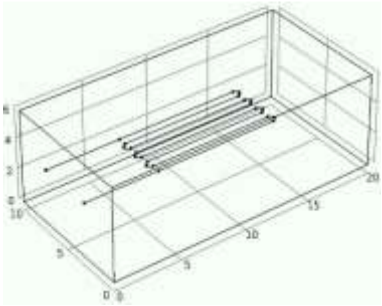
Fig. 4 Effect of thermal properties on the mean transferred thermal energy

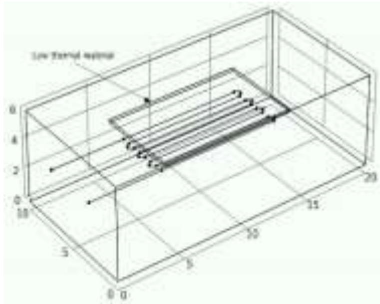
3.2 Effect of different form of PCM application

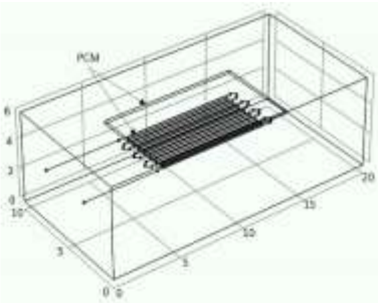
Effect of different form of PCM application on the performance of horizontal ground heat exchanger was investigated. Six cases were considered as shown in Fig. 5.



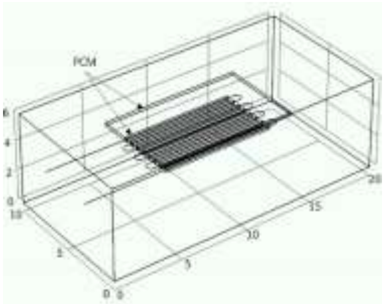
(a) Case 1


(b) Case 2

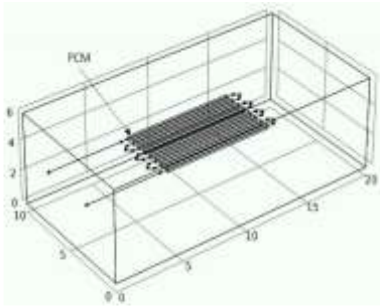

(c) Case3



(d) Case 4



(e) Case 5



(f) Case 6

Fig. 5 Various forms of PCMs application on the horizontal ground heat exchanger

Descriptions of each case are following as:

The location of horizontal ground heat exchanger was identical for all cases.

Case 1: Above ground heat exchanger pipes
Case 2: Without PCMs
Case 3: Use the material with low thermal conductivity
(same as that of PCMs, other properties were identical to the soil)
Case 4: Between and above ground heat exchanger pipes
Case 5: Surrounded and above ground heat exchanger pipes
Case 6: Surrounded ground heat exchanger pipes

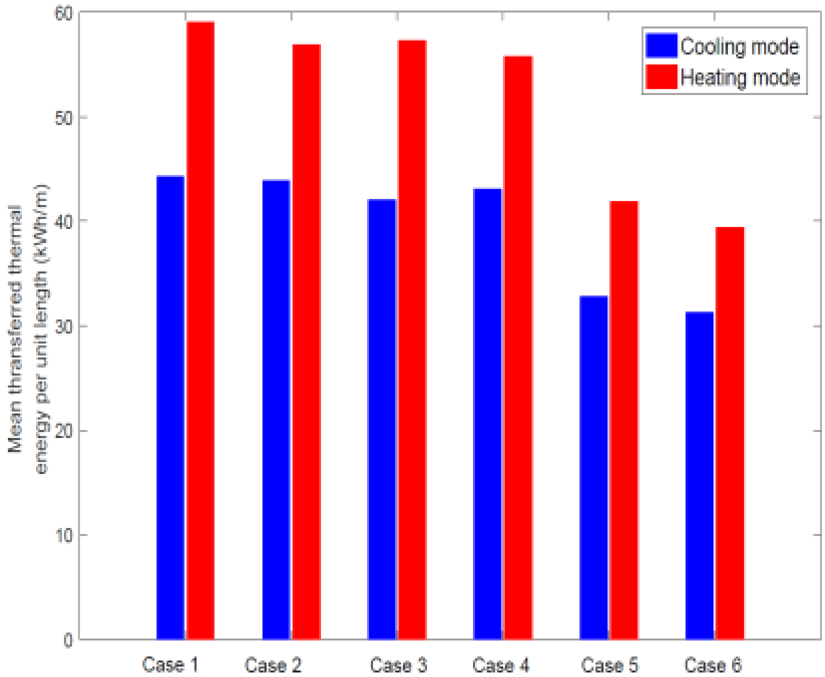


Fig. 6 Mean transferred thermal energy for each case

Fig. 6 shows the results of mean transferred thermal energy unit length for each case. Total mean transferred thermal energy per unit length was 103.33 kWh/m for Case 1, 100.85 kWh/m for Case 2, 99.31 kWh/m for Case 3, 98.87 kWh/m for Case 4, 74.73 kWh/m for Case 5, and 70.65 kWh/m for Case 6. Except the Case 1, all cases had the lower performance than Case 2 in which PCMs were not employed. However, the results of Case 3 showed higher performance than those of Case 2 in a heating mode. This implies that it would be helpful in an area where the heating load is dominating. Compared with the results of Case 2, Case 1 had the 3.8% higher performance in a heating mode, 0.8% higher performance in a cooling mode.

4. CONCLUSIONS

Benefit of phase change material on the performance of horizontal ground heat exchanger was investigated numerically. The use of higher latent heat, specific heat, and density for PCMs improved the performance of horizontal ground heat exchanger. The effect of thermal conductivity on the performance of horizontal ground heat exchanger showed the no coherent trend. Among the thermal properties, the latent heat seems to be the most sensitive in the performance of horizontal ground heat exchanger.

Through the various cases of PCMs application, the use of PCMs with the horizontal ground heat exchanger gave a positive effect on the total mean transferred thermal energy. Extensive studies about the container for PCMs, the free convection effect when melting, the porous medium for soil, and the energy balance at the ground surface would be needed for the real use.

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