DEVELOPMENT AND VALIDATION OF NUMERICAL MODELS OF BOREHOLE HEAT EXCHANGER FOR GSHP (DOUBLE U-TUBE TYPE AND CONCENTRIC TUBE TYPE)

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1. INTRODUCTION

Since a GSHP (Ground source heat pump) system uses thermal heat of the ground as a stable source of heat, its high system efficiency is expected to help save energy and reduce CO₂ emissions, as well as alleviate the heat island effect because it does not release heat to the atmosphere. In Japan, the concept of GSHP systems began to attract interest from around 1980 as an energy-efficient technology, and the use of such systems has increased in recent years as concerns about environmental conservation have heightened. To promote the use of GSHP systems in Japan, it is essential to prepare sufficient engineering data and tools for predicting and evaluating system performance, while reducing the initial cost.

The performance of a GSHP system depends largely not only on the heat exchanger’s heat extraction and injection of efficiencies, but also on the heat pump’s operating performance characteristics and the load profile of the building. To predict and evaluate the system’s operating performance, it is necessary to consider the interrelated actions of three factors: the ground heat exchanger’s performance characteristics, the heat pump’s performance characteristics and the building’s load profile.

However, many of the engineering support tools for GSHP systems do not consider the temporal aspect of the building’s load profile and the heat pump’s operating performance (Nagano et al., 2006), making it difficult to quantitatively predict the performance and effectiveness of the installed system.

Our ongoing study on GSHP systems aims to quantitatively predict the performance and effectiveness of the installed system, and thereby to provide guidance on appropriate operation and control schemes. To do this, we require an analysis model that can simulate the thermal behavior of a ground heat exchanger in consideration of the building’s load profile and the heat pump’s performance profile. This paper reports on our development of transient analysis models that simulate the performance of ground heat exchangers (Borehole type, hereafter referred to as BHE) and our validation of these models through a comparison with the operation records of a system installed in a building.

2. OUTLINE OF THE INSTALLED GSHP SYSTEM

The GSHP system is installed in a building in Yokkaichi, Japan (latitude 35N, longitude 136E). The building’s gross floor space is about 338m². The building has two GSHPs (about 2.2kW each), for air conditioning and water heating. Four BHEs are used as heat source / sink for the heat pumps.

Figure 1 is the primary side diagram of the air conditioning system, which consists of BHEs and heat pumps. Each heat pump is lined up with two BHEs, a 50m pile and a 20m pile, connected in series. The heat source water for each heat pump flows through the 50m pile and then through the 20m pile as it circulates in the loop. The system uses double U-tube type and concentric tube type BHEs.

As shown in Fig. 1, HP-1 uses double U-tube type BHEs, while HP-2 uses concentric tube type BHEs.
The heat pumps are operated such that each heat pump is turned on and off according to the change of load on the secondary side. The average operating duration per day is around 8 hours. The flow rate of the heat source water (ethylene glycol solution, 20%wt.) is about 24L/min. with HP-1, and 19L/min. with HP-2. We sampled data from this GSHP system at one-minute intervals. The parameters we monitored included the following: heat source water inlet temperature (from each heat pump to BHEs), outlet temperature, soil temperature, heat source water flow rate and heat pump operating status.

![Figure 1: The primary side diagram of the air conditioning system](image)

3. NUMERICAL MODELING OF BHE

The two types of BHE used in this heat pump system, namely, the double U-tube type and the concentric tube type, are common in Japan and many have been installed. We therefore developed numerical analysis models for these two types of BHE.

3.1 Numerical Modeling of a Double U-Tube Type BHE

As shown in Fig. 2, the numerical model was designed to include a three-dimensional, rectangular block of soil (20m along the X and Y axes, 80m along the Z axis) around the BHE. The model was assigned different categories of soil at different depths, in accordance with geological survey data. Table 1 shows the relationships between soil categories and thermal peculiarities.

As shown in Fig. 3 (a), a double U-tube type BHE is configured by inserting a pair of 20A class polyethylene U-tubes (133mm in diameter) into a borehole, which is backfilled with mortar. We simplified the numerical model by substituting the round cross-section of the borehole, and that of each U-tube, by a square cross-section, as shown in Fig. 4 (a). In this substitution, we took care not to change the U-tube’s heat transmission area (on the circumference of each tube) and the volume of the backfill.

The entire space for numerical analysis, including the BHEs (Fig. 3) and the block of soil around them (Fig. 2), was divided into a number of cells (59 along the X and Y axes, 80 along the Z axis) for the sake of numerical calculations with the following energy balance formulas:

Energy balance formula for the given block of soil

$$c_s \rho_s \frac{\partial \theta_s}{\partial t} = \lambda \left( \frac{\partial^2 \theta_s}{\partial x^2} + \frac{\partial^2 \theta_s}{\partial y^2} + \frac{\partial^2 \theta_s}{\partial z^2} \right)$$

Energy balance formula for double U-tube type BHEs that takes into consideration advection in the tube and surface heat transfer

$$c_s \rho_s \pi (D/2)^2 \frac{\partial \theta_u}{\partial t} = -c_s \rho_s \nu \pi (D/2)^2 \frac{\partial \theta_u}{\partial z} + U(\theta_u + \theta_o - 2\theta_s) dx + U(\theta_u + \theta_o - 2\theta_s) dy$$
Energy balance formula for the ground surface

\[-\lambda \frac{\partial \theta}{\partial y} \bigg|_{y=0} = \alpha_n (\theta_n - \theta_t) + \alpha J - \alpha d n\]  

(3)

In Formula (2) above, the advection term is assigned the cross-sectional area of the actual (round) U-tube, so as not to affect the calculated flow through the tube with the substitution of the round cross-section by a square cross-section.

Heat transfer within BHE was given as a function of Re number and Nu number (= 0.023Re^{0.8}Pr^{0.4}) in the turbulent flow region; the tube material was treated as a thermal resistant entity with negligible heat capacity (see Note 1). The heat transfer coefficient on the ground surface was assumed to be constant at 23W/m²K. The solar radiation on the ground surface was derived from average-year meteorological data in Nagoya.

<table>
<thead>
<tr>
<th>Thermal Peculiarity</th>
<th>Heat Conductivity(W/mK)</th>
<th>Volumetric Special heat(kJ/m³K)</th>
<th>Water Content (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sand gravel</td>
<td>0.62</td>
<td>1.580</td>
<td>60</td>
</tr>
<tr>
<td>Loam</td>
<td>0.70</td>
<td>3.260</td>
<td>19–22</td>
</tr>
<tr>
<td>Clay</td>
<td>1.74</td>
<td>2.510</td>
<td>10</td>
</tr>
<tr>
<td>Sand</td>
<td>1.39</td>
<td>2.009</td>
<td></td>
</tr>
<tr>
<td>Mortar</td>
<td>1.40</td>
<td>2.222</td>
<td></td>
</tr>
</tbody>
</table>

Table 1: Thermal peculiarity of soil and backfill (Watanabe et al., 1983)

Figure 2: Calculation area of numerical model

Figure 3: Schematic diagram of BHEs

Figure 4: Simplification of numerical model for BHEs

Figure 5: Diagram of heat transfer in concentric tube type
The coefficient of solar radiation absorption by the ground surface was assumed to be 0.35 (Shimoda et al., 1989; on lawn); the coefficient of nocturnal radiation was assumed to be 0.95.

Given these conditions, we performed preliminary calculations (Note 2) for a ten-year period to determine the soil temperature to be used as a reference value, and then simulated the thermal behavior of the BHE for the period between May 1, 2004 and April 30, 2005, which coincided with the installed system’s second year of operation (Note 3). For the heat source water inlet temperature (from the heat pump to the BHEs), heat source water flow and ambient temperature, we used data from field measurements.

3.2 Numerical Modeling of Concentric Tube Type BHEs

As shown in Fig. 3 (b), a concentric tube type BHE is composed of two steel pipes of different diameters, one inserted into the other, which are installed in a borehole and backfilled with sandy soil. Conditions other than those which concern the BHE equipment, namely the conditions of soil around the BHEs, the ground surface conditions and other conditions assumed in the calculations, are the same as those assumed in Section 3.1. The following description, therefore, is limited to how we modeled this type of BHE equipment.

Again with the concentric tube type BHEs, we simplified the model by substituting the round cross-section of the backfilled borehole and each tube by a square cross-section, as shown in Fig. 4 (b). The heat source water goes into the outer tube and comes back through the inner tube, and then goes into another BHE or returns to the heat pump.

With this design, we must consider the exchange of heat between the outer tube and the surrounding soil, along with the transfer of heat between the inner tube and the outer tube. We conceptualized the heat transfer in a concentric tube type BHE as given in Fig. 5, and applied the following energy balance formulas (Diener et al., 1986):

Energy balance formula for concentric tube type BHEs that takes into consideration the heat source water advections in and heat transfer with the outer tube:

\[
c_s \rho_s V \frac{\partial \theta_{ow}}{\partial t} = -c_s \rho_s V \frac{\partial \theta_{ow}}{\partial z} + \alpha_s (\theta_{ow} + \theta_{so} - 2\theta_{sw}) dx + \alpha_s (\theta_{ow} + \theta_{so} - 2\theta_{sw}) dy + U (\theta_{ow} - \theta_{so}) \pi D
\]

Energy balance formula for concentric tube type BHEs that takes into consideration the heat source water advection in and heat transfer with the inner tube:

\[
c_s \rho_s V \frac{\partial \theta_{iw}}{\partial t} = -c_s \rho_s V \frac{\partial \theta_{iw}}{\partial z} + U (\theta_{iw} - \theta_{si})
\]

4. VALIDATION OF ANALYSIS MODELS

We compared the simulation results with actual measurements for each type of BHE to assess the validity of our numerical models. The parameters compared are the temperature variation of the heat pump’s heat source water, and the amount of heat transfer rate (heat extraction or injection). A study of correlations between measured and predicted values revealed a similar tendency over the entire period, so the following descriptions are limited to the two one-month periods in which the maximum amount of heat either heat extraction or injection was recorded. The total amount of heat injected to the ground peaked in September 2004; the total amount of heat extracted from the ground peaked in January 2005.

4.1 Double U-Tube Type BHEs

With double U-tube type BHEs, the heat source water outlet temperature was unavailable. For the validation study, therefore, we referred to the tube surface temperature at the outlet of the underground portion of the tube, which was compared with the predicted heat source water outlet temperature.

HP-1 operated for about 230 hours in September 2004, during which the BHEs injected heat to the ground. During this period, with the double U-tube type BHEs, the heat source water inlet temperature and outlet surface temperature, as measured on the installed system, averaged 38.5°C and 31.5°C, respectively. The total amount of heat injected during the given period was about 2.18MWh. Against these measurements, the heat source water outlet temperature predicted by the model and averaged over the period was about 33.1°C. The deviation from the measurement was a mean of 1.6°C and maximum of 2.66°C. The predicted amount of total heat injected during the period was about 1.69MWh; the deviation from the measurement was about 22.4 percent.
HP-1 operated for about 113 hours in January 2005, during which the heat exchangers collected heat from the ground. During this period, with the double U-type type BHEs, the heat source water inlet temperature and outlet surface temperature, as measured on the installed system, averaged 7.1°C and 10.2°C, respectively. The total amount of heat collected during the given period was about 0.61MWh. Against these measurements, the heat source water outlet temperature predicted by the model and averaged over the period was about 10.8°C. The deviation from the measurement was a mean of 0.6°C and maximum of 1.1°C. The predicted amount of total heat extracted during the period was about 0.73MWh; the deviation from the measurement was about 19.4 percent.

Fig. 6 compares the predicted and measured pattern of changes in heat source water outlet temperature in a single day on account of the double U-tube type BHEs during the heat injection. As shown, the predicted temperature was higher than the measured temperature with the mean difference of 1.4°C; the amount of heat injected from the BHEs was smaller than predicted. Fig. 7 makes a similar comparison for the heat extraction. As shown, the heat source water outlet temperature was predicted rather accurately for hours during which the heat pump was operating, but a larger error of up to about 3°C was observed for hours during which the heat pump was not operating. The relatively large prediction error in this particular study is attributable to the following: the comparison referred to the temperature measured on the tube surface (which could be affected by the diffusion of heat to the soil); the heat pump was operated intermittently with each session lasting not longer than about 20 minutes; and the heat source water flow rate measurement could be inaccurate when the flow was low.

Figure 6: Comparison of the heat source water outlet temperature between prediction and measurement (Double U-type during the heat injection season)

Figure 7: Comparison of the heat source water outlet temperature between prediction and measurement (Double U-type during the heat extraction season)
Thus, the simulation results included relatively large deviations from the measurements due to measurement errors and intermittent heat pump operation. Nevertheless, regarding the trend of variation of heat source water temperature during heat pump operation, the prediction generally matched the measurement. We considered, therefore, that our numerical model can approximate the thermal behavior of double U-tube type BHEs.

4.2 Concentric Tube Type of BHEs

HP-2 operated for about 193 hours in September 2004, during which the heat exchangers injected heat to the ground. During this period, with the concentric tube type BHEs, the heat source water inlet temperature and outlet temperature, as measured on the installed system, averaged 30.8°C and 26.5°C, respectively. (Note that the heat source water outlet temperature was directly measurable with the concentric tube type BHEs.) The total amount of heat injected during the given period was about 1.09MWh. Against these measurements, the heat source water outlet temperature predicted by the model and averaged over the period was about 26.8°C. The deviation from the measurement was a mean of 0.41°C and maximum of 2.6°C. The predicted amount of total heat injected during the period was about 1.03MWh; the deviation from the measurement was about 6.3 percent.
HP-2 operated for about 568 hours in January 2005, during which the heat exchangers extracted heat from the ground. During this period, with the concentric tube type BHEs, the heat source water inlet temperature and outlet temperature, as measured on the installed system, averaged 3.6°C and 6.4°C, respectively. The total amount of heat extracted during the given period was about 1.92MWh. Against these measurements, the heat source water outlet temperature predicted by the model and averaged over the period was about 6.2°C. The predicted amount of total heat extracted during the period was about 1.79MWh; the deviation from the measurement was about 6.6 percent. Fig. 8 and Fig. 9 compare the predicted and measured pattern of changes of heat source water inlet/outlet temperature in a single day on account of the concentric tube type BHEs. As shown, the predicted heat source water outlet temperature was slightly higher compared with the measurement for those hours during which the heat pump was not operating, but the prediction was quite accurate for those hours during which the heat pump was operating. Thus, it was confirmed that the model was capable of accurately reproducing the thermal behavior of the BHEs.

5. CONCLUSION

Based on the comparisons given above, we concluded that, with regard to the double U-tube type BHEs, our numerical model could approximate the thermal behavior of the BHEs in spite of somewhat large prediction errors, because it successfully captured the trend of changes in the heat source water for those hours during which the heat pump was operating. With regard to the concentric tube type BHEs, there was close agreement between the prediction and the measurement.

Using the numerical models of BHEs reported in this paper, we will continue to study and evaluate the operating performance of heat pumps in consideration of the heat pump operating characteristics and the load profile changes on the building. We also plan to propose operation and control schemes optimized to the heat pump and load characteristics.

NOMENCLATURE

c : special heat [kJ/kgK], \( \rho \) : density [kg/m^3], \( \theta \) : temperature [K], \( \lambda \) : thermal conductivity [W/mK], 
\( U \) : thermal transmittance [W/m^2K], \( v \) : flow velocity in pipe [m/s], \( D \) : diameter [m], \( t \) : second [Sec.], 
\( J \) : intensity of solar radiation [W/m^2], \( a \) : solar absorptivity (0.35), \( d_n \) : nocturnal radiation (0.95), 
\( \alpha \) : heat transfer coefficient [W/m^2K], \( Re \) : Reynolds number, \( Pr \) : Prandtl number

-Subscript s : soil, w : heat source water, tu : upper cell adjacent to a pipe, tb : lower cell, tl : left cell, tr : right cell, ss : ground surface, i : inner pipe, o : outer pipe

\( \alpha_o \) is expressed as follows,

\[
\alpha_o = \frac{D_o}{D_i} \left( \frac{\alpha_{su}}{D_o - D_i} \right) \left( \frac{J_{tu}}{J_{tb}} \right) \left( \frac{J_{tl}}{J_{tr}} \right) \left( \frac{J_{ss}}{J_{w}} \right) \left( \frac{\alpha_{su}}{\alpha_{tu}} \right) \left( \frac{\alpha_{tu}}{\alpha_{tb}} \right) \left( \frac{\alpha_{tb}}{\alpha_{tl}} \right) \left( \frac{\alpha_{tl}}{\alpha_{tr}} \right) \left( \frac{\alpha_{tr}}{\alpha_{ss}} \right) \left( \frac{\alpha_{ss}}{\alpha_{w}} \right) \left( \frac{\alpha_{w}}{\alpha_{in}} \right) \left( \frac{\alpha_{in}}{\alpha_{out}} \right) \left( \frac{\alpha_{out}}{\alpha_{inner}} \right) \left( \frac{\alpha_{inner}}{\alpha_{outer}} \right) \left( \frac{\alpha_{outer}}{\alpha_{pipe}} \right) \left( \frac{\alpha_{pipe}}{\alpha_{wall}} \right) \left( \frac{\alpha_{wall}}{\alpha_{ground}} \right) \left( \frac{\alpha_{ground}}{\alpha_{surface}} \right)
\]

Notes:
1) Even though the material of double U-tubes and concentric tubes has a high volumetric specific heat capacity, the tube wall is so thin that its heat capacity is very small compared with the heat capacity of soil, so we treated the tube material as a thermal resistant entity with negligible heat capacity.
2) Assuming that the BHEs were not installed, we repeated calculations with successive entries of ambient temperature measurements from May 1, 2004 through April 30, 2005.

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