DEVELOPMENT OF A DESIGN AND PERFORMANCE PREDICTION TOOL FOR GROUND SOURCE HEAT PUMP SYSTEM

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ABSTRACT

This paper introduces a novel designing and performance prediction tool for the ground source heat pump (GSHP) system which authors have developed. This tool has some advantages and features, such as containing a user-friendly interface for input data and showing calculation result graphically to make operators understand. First, outlines of this tool were described. Next, a performance prediction and an installation effect feasibility of the GSHP system using steel foundation piles as ground heat exchangers in Tokyo was performed with this tool as an example. Results of the calculations show that the $LCCO_2$ of the GSHP system is 1338 kg-CO_2/year and it is less than half compared to conventional systems. Also, the GSHP system can reduce LCC by 50~90 thousand Japanese-yen/year. The cost payback time for increasing investment of the GSHP system is less than ten years against other conventional systems.

Key Words: ***ground source heat pump, cylindrical heat source theory, multiple ground heat exchangers, boundary element method, life cycle analysis.***

1 INTRODUCTION

GSHP system spreads in North America and Europe because optimum GSHP system generally operates with higher efficiency than conventional air-source heat pump (ASHP) system and fossil fuelled equipment for cooling, heating, and hot water supply. Only 120 units have been installed in Japan (NEDO 2003), despite 500,000 units have been installed in worldwide (IEA 2002). This main reason is that the capital cost of the GSHP system was extremely expensive than conventional systems. However, the GSHP system has been remarked as a system with large potential of reducing CO_2 emission after adopting the Kyoto Protocol in 1997 and installing of the GSHP systems has been promoted. For promoting the GSHP system effectively into the market, it is necessary to reduce the drilling cost, to produce a high-efficient and low-cost heat pump, and to develop a user-friendly design tool. In particular with regard to the design tool, there is no such tool with a big reason that the GSHP system has been still not so popular in Japan.

Authors have developed a novel designing and performance prediction tool for GSHP including LCA. Some advantages and features of this tool are described below.

1. A user-friendly data input procedure and graphical outputs based on MS Visual Basic program.
2. A short period of time for two year's calculation according to hourly heating and cooling loads (Calculation time is less than 40 seconds for a typical notebook PC).
3. Included database of CO_2 emissions, lifetimes and costs for various subjects can allow calculation of LCC (Life Cycle Cost) and $LCCO_2$ (Life Cycle CO_2).
4. A high speed algorithm for calculation of thermal response of multiple ground heat exchangers.
5. Available for short ground heat exchangers of big caliber by applying of modification coefficients method.
6. Calculation of internal thermal resistance in a borehole according to tube geometric arrangement by BEM (Boundary Element Method).

Nomenclature

A	Area	[m^2]
a	Thermal diffusivity	[m^2/s]
C	Modification coefficient	
c	Specific heat capacity	[kJ/kg/K]
d	Diameter	[m]
d_i	Distance	[m]
E	Electric power consumption	[W]
h_H	Performance of radiator (Amount of heat radiation per ΔT, which is difference between room temperature and hot water temperature)	[W/ K]
J_x	xth-order Bessel function of first kind	
K	Overall heat transmission coefficient	[W/m^2/K]
L	Length	[m]
m	Flow late	[m^3]
Q	Heat injection rate	[W]
q	Heat flux	[W/m^2]
r	Radius	[m]
r^*	Non-dimensional distance (=r_d/r_{bo})	[-]
T	Temperature	[^{o}C]
T^*	Non-dimensional temperature	[-]
t	Time	[h]
t^*	Fourier number	[-]
t^{***}	Fourier number (r_d-based)	[-]
u	Characteristic value	
V	Volume	[m^3]
Y_x	xth-order Bessel function of second kind	

Subscripts

b	Brine (Antifreeze solution)
bo	Borehole surface (Foundation pile surface)
d	Distance
s	Soil
$Utube$	U-tube
$2out$	Outlet of heat pump to secondary side
$1out$	Outlet of heat pump to primary side
$1in$	Inlet of heat pump to primary side

Greek letters

η	Heat pump COP	[-]
λ	Thermal conductivity	[W/m/K]
ρ	Density	[kg/m^3]

2 CALCULATION METHOD

2.1 Calculation of Ground Temperature

A vertical long ground heat exchanger (borehole type) can be treated as the infinite follow cylinder in the infinite solid. It is possible to apply the cylindrical heat source theory for calculation of the ground temperature. In this case, constant heat flux q_b is generated on the surface of the follow cylinder. The partial differential equation, boundary condition and capital condition are in equation (1)s.

$$\begin{aligned} &\frac{\partial T_s}{\partial t} = a_s\left(\frac{\partial^2 T_s}{\partial r^2} + \frac{1}{r}\frac{\partial T_s}{\partial r}\right) \\ &\text{BC}; -\lambda_s \left.\frac{\partial T_s}{\partial r}\right|_{r=r_{bo}} = q_{bo} \\ &\text{IC}; T_s = T_{s0} \quad (t=0) \end{aligned} \tag{1}$$

The solution is (Carslaw and Jaeger 1959)

$$T_s = T_{s0} - \frac{2q_{bo}}{\pi\lambda_s}\int_0^\infty (1-e^{-a_s u^2 t})\frac{J_0(ur)Y_1(u\,r_{bo}) - Y_0(ur)J_1(u\,r_{bo})}{u^2[J_1^2(u\,r_{bo}) + Y_1^2(u\,r_{bo})]}du \tag{2}$$

The equations shows ground temperature for optical radius r. **Figure 1** is the non-dimensional temperature response $T^* = \lambda_s T_s / r_{bo} q_{bo}$ on the surface according to Fourier number $t^* = a_s t / r_{bo}$. The superposition of this response is applied in the developed tool for hourly heat extraction and heat injection. Here, an approximate expression of **Figure 1** is used to calculate this response.

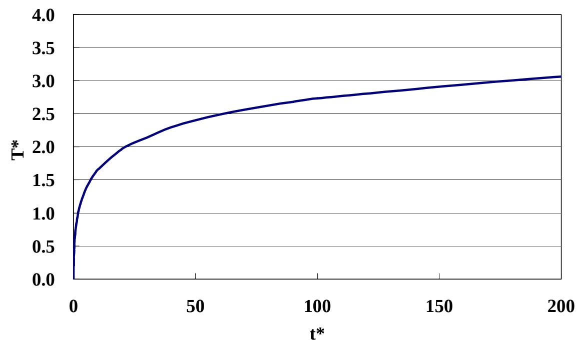


Fig. 1. Surface temperature of the follow cylinder in the infinite solid with zero initial temperature and constant heat flux at the surface

2.2 Calculation for Multiple Ground Heat Exchangers

As shown in **Figure 2**, n (n is constant) number of ground heat exchangers is buried in random layout. In this case, the surface temperature of i-th ground heat exchanger is by the following equation.

$$T_s \Big|_{r=r_{bo}} = -\frac{2}{\pi\lambda_s}\{q_{boi}\, I(r_{bo}) + \sum_{j=1}^{n} q_{boj}\, I(r_{dij})\} \tag{3}$$

Here, j (j is constant) is surrounding ground heat exchanger ($i \neq j$) and r_{dij} is distance between i and j. Heat flux q_{boi} is generated by heat exchange of i-th ground heat exchanger.

Temperature calculations for radius r_{bo} and r_{dij} are different and $I(r_{bo})$ can be calculated by using the approximate expression of **Figure 1**. In general, r_{dij} is larger than several meters and r^* is larger than ten. Thus temperature influence from the other ground heat exchangers can be calculated by applying the line heat source theory shown in **Figure 3**. This response gives temperature influences when distance r_{dij} and elapsed time t are substituted.

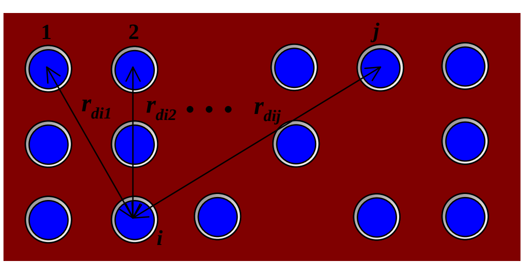


Fig. 2. Multiple ground heat exchangers buried in random layout

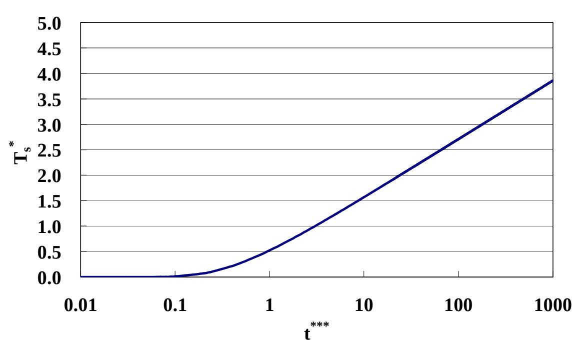


Fig. 3. Ideal borehole surface temperature of the infinite line heat source in the infinite solid for t^{***}

2.3 Calculation of the GSHP System Operation

Figure 4 shows the typical GSHP system diagram. It consists of three parts: They are radiators for cooling or heating, heat pump units, and ground heat exchanger. Heat balance equations of each part are formed.

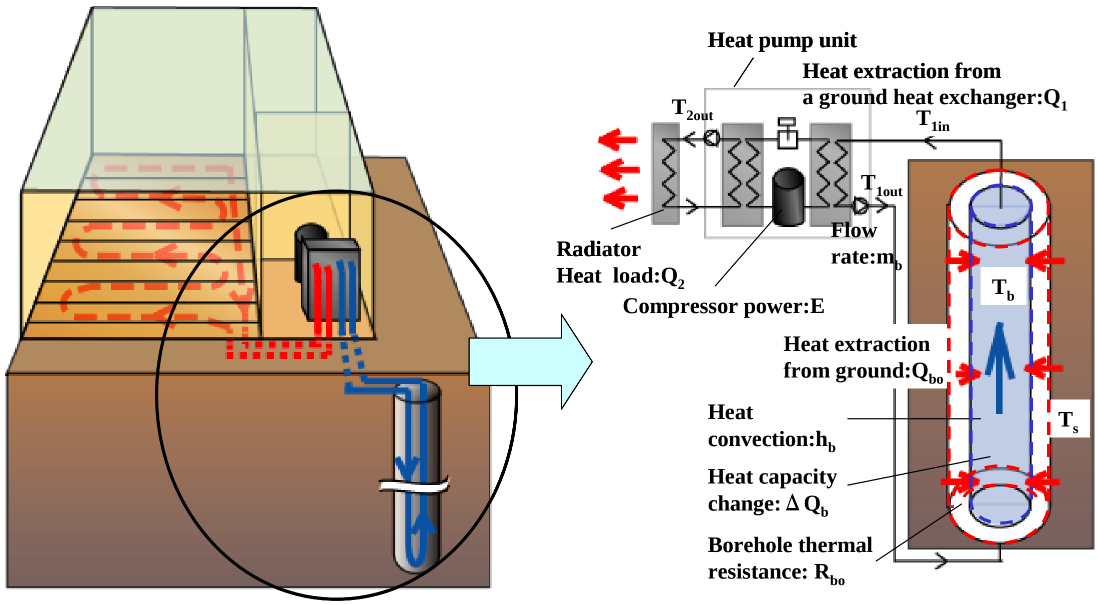


Fig. 4. Calculation diagram of typical GSHP system

1. Radiator

Overall heat transfer coefficient of radiators h_H and heat load of the building Q_2 are known. Then the temperature for heating and cooling T_{2out} is calculated by the next equation.

$$T_{2out} = \frac{Q_2}{\sum h_H} \tag{4}$$

2. Heat pump unit

The average coefficient of performance (COP) of a heat pump can be expressed as a function of dinlet to the heat pump T_{1in} and T_{2out}.

$$\eta = f(T_{1in}, T_{2out}) \tag{5}$$

Electric power consumption of the heat pump E is calculated in equation (6).

$$E = \frac{Q_2}{\eta} \tag{6}$$

In this time, heat balance equation at the primary side of the heat pump unit is obtained.

$$Q_1 = c_b \rho_b m_b (T_{1out} - T_{1in}) \tag{7}$$

On the other hand, the amount of heat extraction from the ground heat exchanger Q_1 can be shown even by the following equation.

$$Q_1 = Q_2 - E \tag{8}$$

From the equation (7) and (8), equation (9) is derived.

$$T_{1out} = T_{1in} - \frac{Q_1}{c_b \rho_b m_b} = T_{1in} - \frac{Q_2 - E}{c_b \rho_b m_b} \tag{9}$$

3. Ground heat exchanger

Considered heat balance in a ground heat exchanger, the amount of heat extraction from the ground heat exchanger can be shown as a sum of heat capacity change of the antifreeze solution in the ground heat exchanger ΔQ_b and heat transfer from ground to the antifreeze solution Q_{bo}.

$$Q_1 = \Delta Q_b + Q_{bo} \tag{10}$$

Heat capacity change ΔQ_b is expressed by temperature change of the antifreeze solution and written in equation (11)

$$\Delta Q_b = c_b \rho_b V_b \frac{dT_b}{dt} \tag{11}$$

Heat transfer Q_{bo} is calculated by the following heat balance equation if antifreeze solution temperature T_b is considered uniform.

$$Q_{bo} = K_{bo-out} A_{bo-out} (T_s|_{r=r_{bo}} - T_b) \tag{12}$$

Substituted equation (11) and (12) into equation (10), the following deferential equation is obtained.

$$c_b \rho_b m_b (T_{1out} - T_{1in}) = c_b \rho_b V_b \frac{dT_b}{dt} + K_{bo-out} A_{bo-out} (T_s|_{r=r_{bo}} - T_b) \tag{13}$$

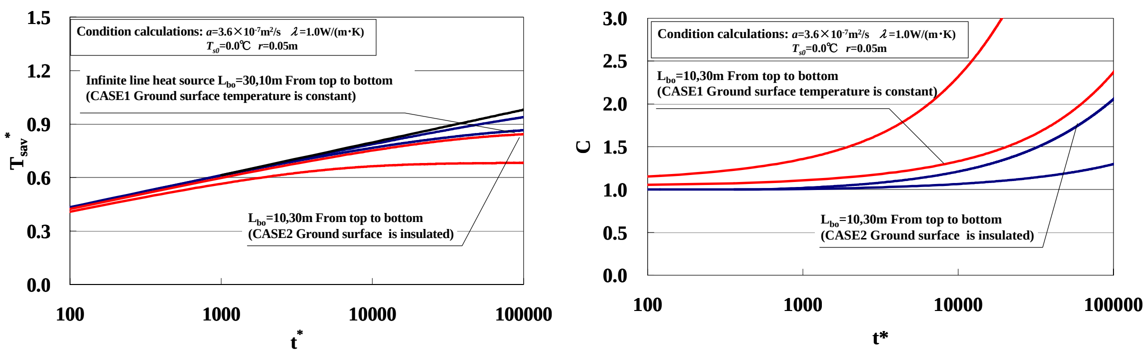
Here, T_b is treated as the mean temperature of T_{1in} and T_{1out}. As a result, it is possible to calculate T_b and T_{1in}. The surface temperature of the ground heat exchanger $T_s|_{r=r_{bo}}$ is calculated by using the equation (2). Heat flux q_{bo} of the equation (2) can be calculated by Q_{bo}/A_{bo}.

2.4 Life Cycle Analysis of the GSHP System

The electric power consumption due to the heat pump unit and circulation pumps is calculated. Then the annual primary energy consumption, CO_2 emission, and running costs can be calculated from the electric power consumption. The life cycle primary energy consumptions, CO_2 ($LCCO_2$), and costs (LCC) are estimated by using industrial input-output tables and a pile up approach. The industrial input-output tables are cited from Architecture Institute of Japan (AIJ). The object items are classified into the department independence. The capital energy consumptions and CO_2 emissions are estimated by multiplying the weight to the basic unit of energy consumption and CO_2 emission in the industry input-output tables. The weight and capital costs are the value obtained by researching the catalogs. These for maintenance and renovation are also estimated by these methods. The intervals for maintenance and renovation are quoted from the handbook of the Society of Heating, Air-conditioning and Sanitary Engineers of Japan (SHASE 1995). This tool can also compare $LCCO_2$ and LCC with these occurred by operating the conventional systems by using the pile up approach.

2.5 Evaluation of Average Surface Temperature of Ground Heat Exchanger

Influence of heat transfer from the edge of the ground heat exchanger and the ground surface is large when length of the ground heat exchangers is small to radius. The surface temperature variation is excessively evaluated if the ground heat exchanger is treated as infinite line heat source or infinite cylindrical heat source. It is important in case where foundation piles are used as ground heat exchangers. Eskilson focused this point and developed the calculation method by using equivalent radius (Eskilson 1987). It is obtained by multiplying modification coefficient to the ideal ground heat exchanger radius of finite line heat source and middle point temperature for equivalent radius equal average temperature for the real ground heat exchanger radius. Nagasaka calculated hourly modification coefficient and obtained approximate equation on the hypothesis that ground surface temperature stay initial ground temperature T_{s0} constant (Nagasaka et al. 1995). While, it is necessary to insulate the ground surface when foundation piles are used as ground heat exchangers. Modification coefficient for condition that ground surface is insulated is calculated by utilizing axial symmetry of insulated boundary. Calculated hourly average non-dimensional temperatures and modification coefficients for each condition are compared in **Figure 5** (a) and (b). However, these modification coefficients are what to obtain equivalent radius by multiplying them to the ideal ground heat exchanger of infinite line heat source or infinite cylindrical heat source because of the infinite cylindrical heat source theory is applied for calculation. The average non-dimensional temperature calculated under the condition of insulated ground surface (CASE2) is larger than one of CASE1. On the ten meters line heat source, the average non-dimensional temperature and modification coefficient of CASE1 are 0.66 and 2.33, those of CASE2 are 0.77 and 1.2 for t^*=10000.



(a) Non-dimensional temperature **(b) Modification coefficient**

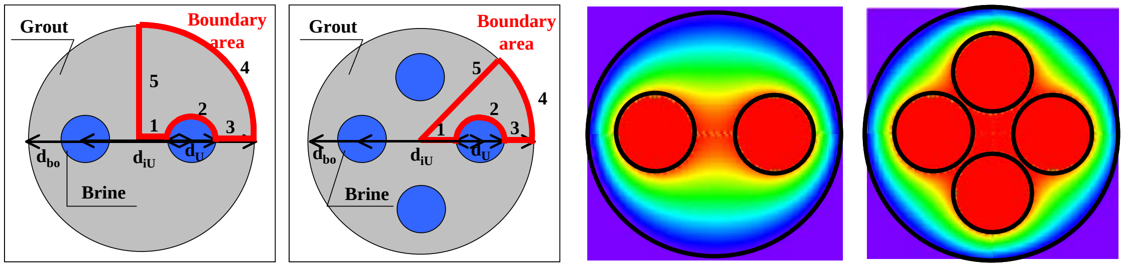
Fig. 5. Variation of non-dimensional temperature and modification coefficient for ground surface conditions

2.6 Calculation of Internal Thermal Resistance between Borehole Surface and U-tube external surface

Overall heat transfer coefficient $K_{bo\text{-}out}$ is needed for calculation of heat balance expressed by equation (12). Geometric U-tube installation makes a calculation of thermal resistance between borehole surface and U-tube external surface difficult. Kavanaugh showed the calculation method that treats a U-tube as a follow cylinder by using effective radius on the hypothesis that part of the U-tube external surface completely contacts the borehole surface (Kavanaugh 1984). Here, boundary element method (BEM) is applied to evaluate quantitatively thermal resistance under the condition that position of the U-tube differs. Boundary area calculated by BEM is shown in **Figure 6 (a)**. Boundary area is surrounded by thick line and distinguished to five elements. Adiabatic condition is given element1, element 3, and element 5. Element2 and element4 take potential condition. The potential is one for element2 and zero for element4. Borehole (grout) thermal conductivity λ_{bo}, U-tube diameter d_{Utube}, borehole diameter d_{bo} and distance from center of one pipe to center of another pipe of U-tube d_{iUtube} is given and calculate is carried out. Using heat flux $q_{bo\text{-}out}$ of the external surface obtained by BEM, thermal resistance in the borehole $R_{bo\text{-}out}$ is calculated by the following equation.

$$R_{bo-out} = \frac{1}{q_{bo-out}} \quad (16)$$

As examples of the calculation, contour charts of cross section, whose internal temperature are calculated by BEM, are demonstrated in **Figure 6 (b)**. Calculated thermal resistances in the borehole per unit area of the external surface are shown in **Table 1** when grout thermal conductivity λ_{bo} is assumed as 1.8 W/m/K. It is quantitatively shown that the thermal resistance is smaller when the U-tube is nearer to the borehole surface. For example, the ratio η_{single} is 0.86 for case1-2 and 0.61 for case1-3, respectively. In addition to mention, the thermal resistances of the double U-tube are 70% compared to ones of the single U-tube.



Single U-tube **Double U-tube**

(a) Boundary area

Single U-tube **Double U-tube**

(b) Contour charts

Fig. 6. Boundary area and calculation result of BEM

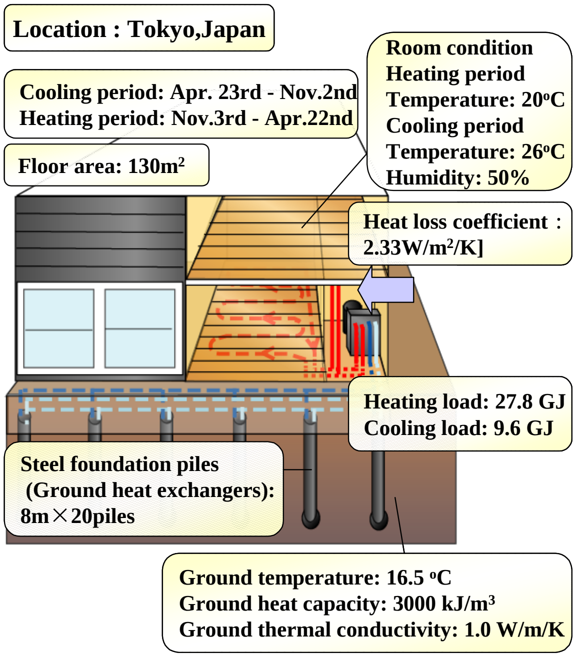
Table 1. Thermal resistance in the borehole for per area of the external surface

	d_{Utube} [mm]	d_{bo} [mm]	d_{iUtube} [mm]	$R_{bo\text{-}out}$ (Single U-tube) [(m^{2}K)/W]	η_{single} (=$R_{bo\text{-}out}/R_{bo\text{-}out,case1\text{-}1}$)	$R_{bo\text{-}out}$ (Double U-tube) [(m^{2}K)/W]	η_{double} (=$R_{bo\text{-}out}/R_{bo\text{-}out,case1\text{-}1}$)
case1-1			52	2.23×10^{-2}	1.00	1.52×10^{-2}	1.00
case1-2	32	120	62	1.91×10^{-2}	0.86	1.23×10^{-2}	0.81
case1-3			72	1.35×10^{-2}	0.61	7.58×10^{-3}	0.50
case2-1		120		2.23×10^{-2}	1.00	1.52×10^{-2}	1.00
case2-2	32	100	52	1.10×10^{-2}	0.49	6.48×10^{-3}	0.43
case2-3		150		3.99×10^{-2}	1.79	2.99×10^{-2}	1.97
case3-1	32		52	2.23×10^{-2}	1.00	1.52×10^{-2}	1.00
case3-2	20	120	40	3.57×10^{-2}	1.60	2.64×10^{-2}	1.74
case3-3	44		64	1.04×10^{-2}	0.47	6.04×10^{-3}	0.40

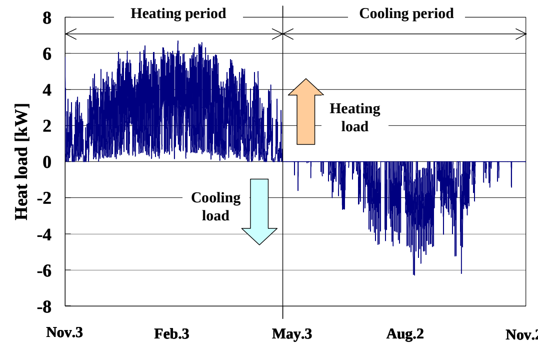
2 PERFORMANCE AND INSTALLATION EFFECT FEASIBILITY OF THE GSHP SYSTEM WITH STEEL FOUNDATION PILES AS GROUND HEAT EXCSHNGERS

2.1 Conditions for Calculations

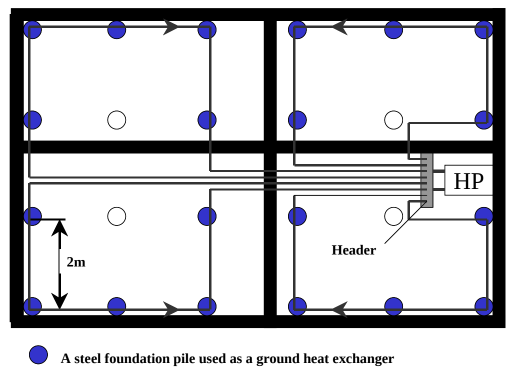
Performance prediction and installation effect of the GSHP system with foundation piles as the ground heat exchangers were examined with the developed tool. As an example, performance of the GSHP system for a typical residential house in Tokyo was calculated. Outlines of calculation conditions are shown in **Figure 7**s. The average floor area of 130m2 in Japan is given for the calculation. The room temperature is set 20°C during heating period and 26°C during cooling period. Only for cooling period, the set value of the room humidity is less than 60%. The heat loss coefficient is calculated under the conditions that the overall heat transfer coefficients of each part are less than standard values, with the result that it is 2.33 $W/m^2/K$. The heating and cooling load are 27.8 GJ and 9.6 GJ, which are calculated with a transient building heating and cooling load calculation software "SMASH for Windows (developed by Institute for Building Environment and Energy Conservation (IBEC) in Japan)". **Figure 7 (b)** shows hourly heating and cooling loads. Twenty steel piles, whose length is 8m, are used as ground heat exchangers and total length is 160m. Installed arrangement of twenty piles is shown in **Figure 7 (b)**. Five piles are connected in serial each other and there are four parallel circuits. Initial ground temperature is 16.5 °C. Typical average effective thermal conductivity 1.0 W/m/K and thermal capacity 3000 kJ/m^3 are given. **Figure 7 (d)** shows a system diagram of heating and cooling. Radiating floor heating is $70m^2$ and two fan-coil units of cooling power 3.6kW are used. The standard COP of the heat pump unit at the condition of T_{1in} =0 °C and T_{1in} =35 °C is 4.5 (Stiebel Eltorn 2000). We assumed that the heat pump unit has a variable speed compressor and so it can change T_{2out} according to heating demands. On the other hand, constant T_{2out} =7 °C is circulated due to dehumidification during cooling period The steel piles are used as indirect type heat exchanger in which water is filled and plastic tubes inserted into water. The tubes are coupled to a heat pump unit and heat extraction or injection is carried out by circulating antifreeze solution.



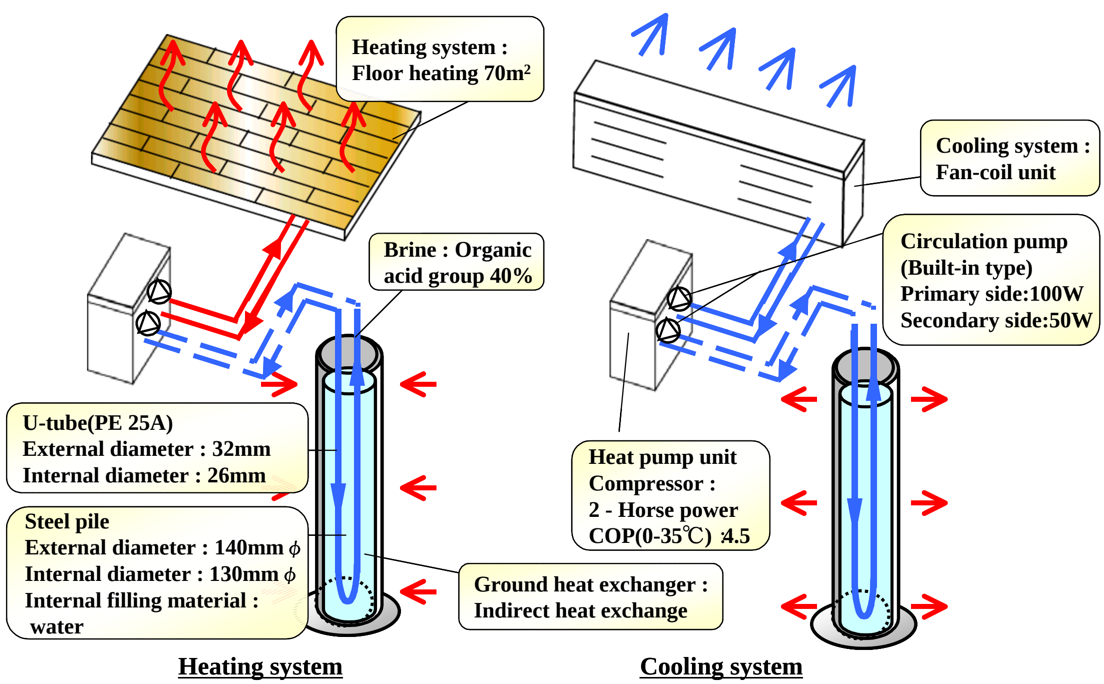
(a) Outline of subject a residential house



(b) Hourly variation of heat load



(b) Arrangement of steel foundation piles



(d) System diagram of heating and cooling

Fig. 7. Conditions for calculations

2.2 Result and Discussion

2.2.1 Performance of the GSHP system

Figure 8 shows hourly temperature variations of T_{2out}, T_{1out} and T_{bo} for the second years result. In addition, **Table 2** indicates an evaluation at the time of the 2nd year's operation of the GSHP system. T_{1out} is rarely below 0 °C and it is not necessary to worry about freezing the filled water in steel piles. Differences of T_{1out} and T_{bo} at start of the second year's operation Nov.3rd and end of the second year's operation Nov.2nd are hardly observed. Although ratio of the amount of heat extraction from the ground and the heat injection from the ground is approximately 2:1, we have proved that these temperatures have almost achieved periodical steady state. This result suggests the possibility of long year's operation. Average T_{1out} during heating period is 7.8 °C and it is higher than average air temperature 6.8 °C from Dec.1st to Mar.31st (National Astronomical Observatory 2000). Average T_{1out} during cooling period is 16.5 °C and it is lower than average air temperature 22.2 °C from May.1st to Oct.31st. From these, it is expected that the GSHP system can operate higher efficiency than conventional ASHP system from both of heating and cooling. This is demonstrated from high average COP of 5.1 and 5.3 shown in **Table 2**. Although average COP during cooling period is larger than one

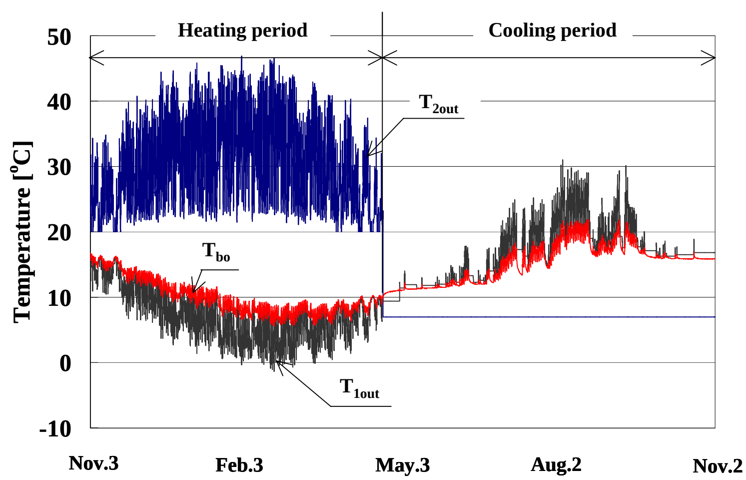


Fig. 8. Hourly temperature variations of the each part

Table 2. Evaluation at the time of operation of the GSHP system

Heating period				
Amount of the heat extraction from the ground [GJ]	Electric power consumptions of the heat pump [kWh]	Electric power consumptions of the circulation pump [kWh]	Average COP	Average SCOP
22.6	1508	395	5.1	4.1
Cooling period				
Amount of the heat injection to the ground [GJ]	Electric power consumptions of the heat pump [kWh]	Electric power consumptions of the circulation pump [kWh]	Average COP	Average SCOP
11.5	436	178	5.3	3.5

during heating period, average SCOP during cooling period is smaller. This reason is that the power of the compressor is controlled according to heat demand, but the power of the circulation pumps is almost constant when heat demand occurred in the calculation conditions. The SCOP during cooling period is small by increasing ratio of electric power consumptions of circulation pump. Therefore this indicates that controlling circulation pump is necessary to improve the SCOP.

2.2.2 Life cycle analysis

The GSHP system was compared to conventional systems with the LCA indicated in **Chapter.1.4**. **Table 3** shows four types of hot-water heating and conventional air conditioning systems (AC) classified by equipment composition. **Table 3** also shows capital CO_2 emission, cost, and lifetime of the each facilities and executions. CASE1 is the GSHP system and others are conventional systems, which are an oil boiler and an AC (CASE2), a gas boiler and AC (CASE3), and an ASHP system (CASE4). The cost of the GSHP system doesn't include material and burial cost of the steel foundation piles, which are categorized in foundation construction cost. As a condition of LCA, each systems produce heat for heating and cooling to heat load of the house and consumptions of the electric power, oil, and gas are calculated with conversion factors shown in **Table 4**. The boiler coefficient is set 0.8. The average ASHP unit COP is 2.5 for heating and 2.7 for cooling and that of AC is same value. Calculation result of $LCCO_2$ and LCC is shown in **Figure 9** for life time of 60 years. The $LCCO_2$ of the GSHP system is 1338 kg-CO_2/year and less than half compared to conventional systems. We can know that main part of $LCCO_2$ is occurred from the operation of the system from **Figure 9**. The LCC of the GSHP system can be reduced by 50~90 thousand Japanese-yen/year. The running costs of the CASE1 are smaller than the other conventional systems. Also the renewal costs of CASE1 are most inexpensive because of drilling costs and ducting costs aren't generated after the GSHP system installed. **Figure 10** shows yearly variation of integrating annual cost. The cost payback time for increasing investment of the GSHP system will be six years against CASE2. It is four years against CASE3 and CASE4, respectively. From these calculations, we confirmed that the GSHP system with the foundation piles as ground heat exchangers has the great potential from both of environmental and economical view.

Table 3. Four types of hot-water heating and air conditioning (AC) systems and its lifetime, capital CO2 emissions, and cost

	Element	Lifetime [year]	CO_2 emissions [kg-CO_2]	Cost [Japanese yen]
CASE1 GSHP system	Heat pump unit	20	469.0	450,000
	Brine(Organic acid type) [/l]	20	2.0	500
	Ground heat exchanger piping	60	4.0	160,000
	Ductwork and setting	60	22.0	250,000
	Floor heating	30	5845.0	840,000
	Fan-coil unit	20	268.0	72,000
CASE2 Oil boiler and AC	Oil boiler unit	15	63.8	200,000
	Boiler maintenance (Burner etc)*	7.5	29.0	80,000
	Oil tank	20	134.1	50,000
	Copper pipe for oil	10	0.4	3,500
	Ductwork and setting	60	2.0	50,000
	Air conditionig for cooling	10	214.4	280,000
	Floor heating	30	5845.0	840,000
	Fan-coil unit	20	268.0	72,000
CASE3 Gss boiler and AC	Gas boiler unit	15	81.2	270,000
	Boiler maintenance (Burner etc)*	7.5	29.0	80,000
	Air conditionig for cooling	10	214.4	280,000
	Ductwork and setting	60	2.0	50,000
	Floor heating	30	5845.0	840,000
	Fan-coil unit	20	268.0	72,000
CASE4 ASHP system	Air source heat pump unit	15	582.9	680,000
	Ductwork and setting	60	2.0	50,000
	Floor heating	30	5845.0	840,000
	Fan-coil unit	20	268.0	72,000

*When boiler unit is needed to renovate, maintenace isn't carried out

Table 4. Conversion factors

	Primary energy consumptions	CO_2 emissions [kg-CO_2]	Basis rate [Japanese yen /month]	Unit cost [Japanese yen]
Electric power [/kWh]	9.42	0.36	560	under 120kWh:14.82 under 300kWh:19.66 over 300kWh:21.13
Oil [/L]	36.70	2.5	0	55
Gas [/m^3]	41.10	2.1	2300	91

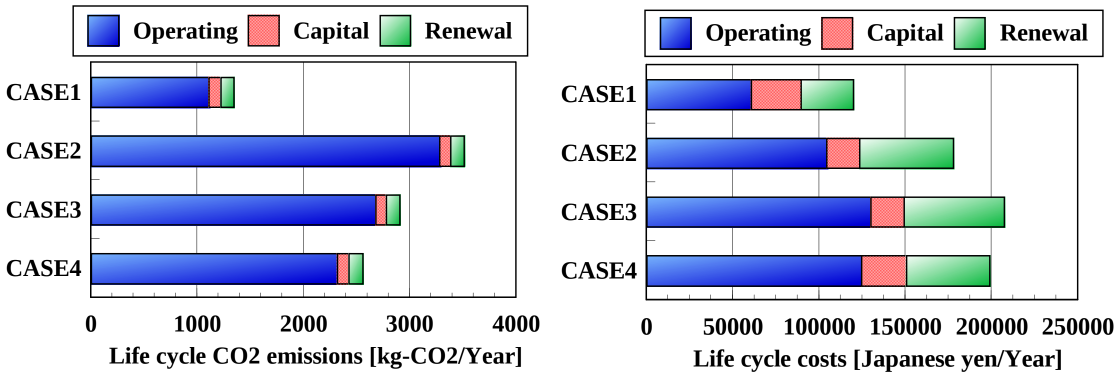


Fig. 9. Comparisons of $LCCO_2$ and LCC of each system (Renewal contains maintenance and renovation)

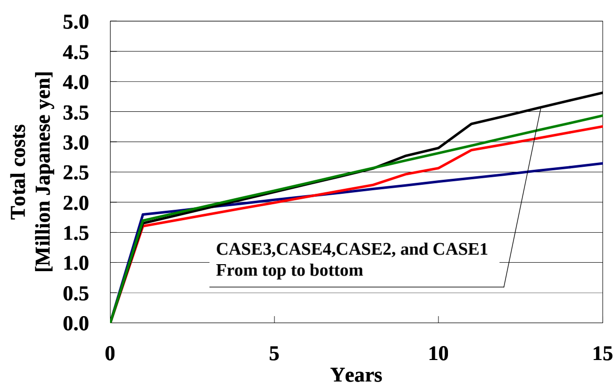


Fig. 10. Yearly variations of integrated annual costs

3 CONCLUSIONS

1. Authors have developed a novel designing and performance prediction tool for the GSHP system which applies infinite cylindrical heat source theory with high calculation algorism. The tool can perform LCA of GSHP system to each other conventional system. This tool also includes calculation routine to evaluate thermal resistance of the borehole by using BEM. The outlines of calculation method were introduced.

2. A performance prediction and an installation effect feasibility of the GSHP system using steel foundation piles as ground heat exchangers was performed with the developed tool. This result shows that the GSHP system with steel piles can be operated long year's with higher efficiency. The $LCCO_2$ of the GSHP system is 1338 kg-CO_2/year and more than half compared to other systems. Results of the calculations show that the $LCCO_2$ of the GSHP system is 1338

kg-CO_2/year and it is less than half compared to conventional systems. Also, the GSHP system can reduce LCC by 50~90 thousand Japanese-yen/year. The cost payback time for increasing investment of the GSHP system is six years against the oil boiler and the AC, four years against the gas boiler and the AC, and the ASHP system. As a result, we could show that the GSHP system with the foundation piles as ground heat exchangers has great potential from both of environmental and economical view.

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