

DEVELOPMENT OF A COMPUTER AIDED SIMULATION PROGRAM FOR THE GROUND SOURCE HEAT PUMP SYSTEM COMBINED COOLING TOWER AND ITS APPLICATION

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Abstract: In order to investigate design and operating method of the ground source heat pump (GSHP) system combined cooling tower, the authors developed a computer aided simulation program for the system. Performance of the GSHP system combined cooling tower, which has been installed in a new office building in Kitakyushu City, is predicted with the simulation program. The result shows the combined GSHP system can operate with high cooling SPF of 5.4. In addition, the authors investigated connection sequence and operation method of the combined GSHP system. The result shows that the GSHP system provides the highest cooling SPF when cooling tower is connected in series with ground heat exchangers (GHEXs) and the water as heat carrier fluid is circulated to the cooling tower ahead of the GHEXs. The system performance is additionally improved by set the water temperature in which the cooling tower is operated.

Key Words: Ground Source Heat Pump, Cooling Tower,
Combined System, Computer Simulation Tool

1 INTRODUCTION

In the most part of Japanese commercial buildings, the cooling load is much larger than the heating load. Therefore, one of the main issues to expand Japanese market of the ground source heat pump (GSHP) system is to prevent long-term ground temperature rise, which is caused by excess heat injection due to the cooling load.

Releasing the excess heat to atmosphere by using the cooling tower is effective method to cut down the issue. However, few studies have been reported on the design and operating method of the GSHP system combined cooling tower because there are not so many studies on the GSHP system in moderate climate region.

The authors previously reported several studies on the design tool for the GSHP system (Katsura et al., 2009, Katsura et al., 2008, Nagano et al., 2006). As the most advantage of the tool, several years' operation of the GSHP system that has multiple ground heat exchangers can be simulated in a few minutes. This tool also can calculate the long-term ground temperature rise caused by excess heat injection.

Combining calculation algorithm of the design tool and calculation algorithm of the cooling tower, a computer simulation tool for the GSHP system combined cooling tower is developed. In this paper, algorithm of the simulation tool is firstly described. Next, performance of the GSHP system combined cooling tower, which has been installed in a new office building in Kitakyushu city, is predicted by using the simulation tool. Additionally, design method and operation method of the GSHP system combined cooling tower are investigated.

2 OUTLINES OF GROUND SOURCE HEAT PUMP SYSTEM COMBINED COOLING TOWER

Figure 1 is a diagram of the GSHP system combined cooling tower. The water source heat pumps are connected to the ground heat exchangers and cooling tower in the primary side.

As the advantage of the combined GSHP system compared to the conventional GSHP system, the excess heat due to cooling load can be released to the atmosphere by using the cooling tower. This prevents long-term ground temperature rise caused by the excess heat injection to the ground and keeps high-efficiency of the GSHP system. Also, efficiency of the combined GSHP system in heating operation is much higher than the one of the water source heat pump system using only the cooling tower (heating tower).

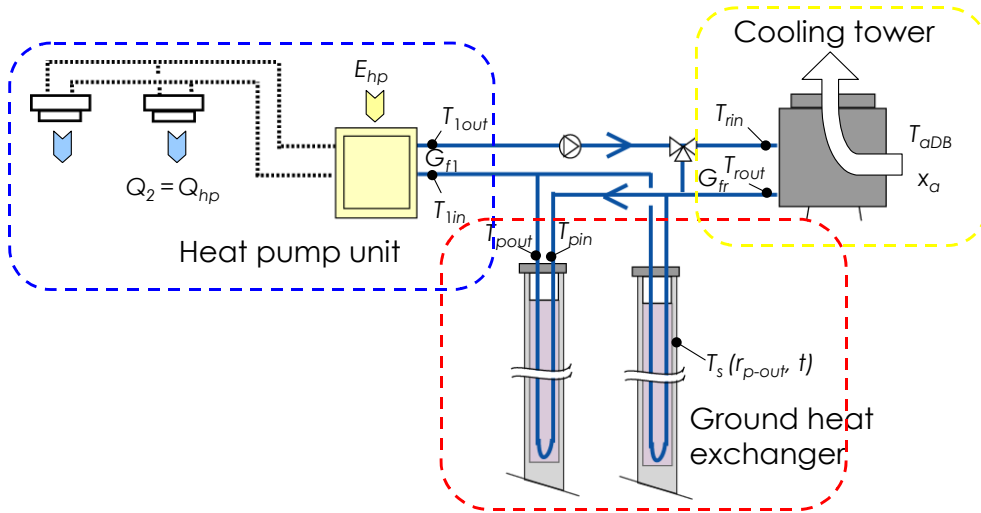


Figure 1: Diagram of GSHP system combined cooling tower

3 SIMULATION METHOD OF GROUND SOURCE HEAT PUMP SYSTEM COMBINED COOLING TOWER

As shown in Figure 1, the combined GSHP system is mainly composed of three parts, which are the heat pump unit, ground heat exchangers and cooling tower. The simulation methods of three parts are described below.

3.1 Simulation method of ground heat exchangers

3.1.1 Simulation method of temperature for heat extraction via multiple ground heat exchangers

As proposed in the previous report (Katsura et al., 2008), soil is treated as infinite isotropic constant solid and vertical ground heat exchanger (borehole or pile) is regarded as a heat source (line or cylindrical) with infinite length in the infinite solid. The constant n expresses the number of ground heat exchangers which are buried in a random layout as shown in Figure 2. The total temperature response on the surface of a certain ground heat exchanger $\Delta T_s(r_{p-out,i}, t)$ can be obtained by summing up all temperature responses (Katsura et al., 2008).

$$\Delta T_s(r_{p-out,i}, t) = \sum_{j=1}^n \Delta T_{s-L}(r_{d,ij}, t) + \Delta T_{s-C}(r_{p-out,i}, t) \quad (i \neq j) \quad (1)$$

In this time, i is the certain ground heat exchanger and j is one of the surrounding heat exchangers. In addition, when the generated heat on the surface of the cylinder Q' varies according to elapsed time, the temperature responses $\Delta T_{s-C}(r_{p-out}, t)$ and $\Delta T_{s-L}(r_d, t)$ are evaluated by the following equations (Katsura et al., 2008).

$$\Delta T_{s-C}(r_{p-out}, t) = \frac{r_{p-out} q_0 \Delta T_{s-C}^*(1, t^*)}{\lambda_s} \quad (2)$$

$$\Delta T_{s-L}(r_d, t) = \frac{r_{p-out} q_0 \Delta T_{s-L}^*(r_d^*, t^*)}{\lambda_s} \quad (3)$$

Also, $\Delta T_{s-C}^*(1, t)$ and $\Delta T_{s-L}^*(r_d, t)$ can be approximated by the following equations (Katsura et al., 2008).

$$\Delta T_{s-C}^*(1, t^*) \cong \sum_{\tau^*=0}^{t^*} q^*(t^* - \tau^*) \frac{\partial T_{s-C}^*(1, \tau^*)}{\partial \tau^*} \quad (4)$$

$$\Delta T_{s-L}^*(r^*, t^*) \cong 0 \quad (t^* < 0.05)$$

$$\Delta T_{s-L}^*(r^*, t^*) \cong \sum_{i=1}^m \Delta T_{s-Li}^* \quad (0.05 \leq t^* < 1.0)$$

$$\Delta T_{s-L}^*(r^*, t^*) \cong \sum_{i=1}^n \Delta T_{s-Li}^* + \sum_{\tau^*=1}^{t^*} q^*(t^* - \tau^*) \frac{\partial T_{s-C}^*(1, \tau^* r^{*2})}{\partial (\tau^* r^{*2})} \quad (t^* \geq 1.0) \quad (5)$$

Here,

$$\begin{aligned} \Delta T_{s-Li}^* &\cong \sum_{\tau^*=t_{i-1}^*}^{t_i^*} q^*(t^* - \tau^*) \frac{\partial T_{s-L}^*(r^*, \tau^*)}{\partial \tau^*} \\ &\cong \overline{q_i} (T_{s-L}^*(r^*, t_i^*) - T_{s-L}^*(r^*, t_{i-1}^*)) \\ q^*(t^*) &= q'(t) / q'_0 \end{aligned}$$

As the result, the ground temperature for heat extraction or injection via the multiple ground heat exchangers can be calculated with satisfied precision and speed.

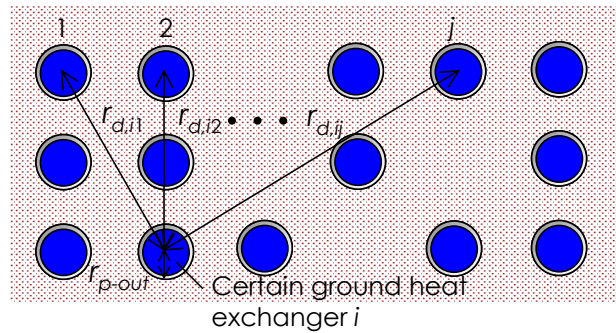


Figure 2: Multiple ground heat exchangers buried in random layout

3.1.2 Simulation method of fluid temperature in ground heat exchanger

First, it is assumed that heat loss in a circuit can be neglected. As shown in Figure 3, m numbers of parallel circuits are arranged and each circuit has n number of ground heat exchangers in series. The flow rate is divided according to the number of parallel circuits. If total flow rate is G_{1f} , the flow rate per one circuit is G_{1f} / m . On the other hand, n number of ground heat exchangers arranged in series is regarded as a ground heat exchanger which has length equal to the sum of n number of the ground heat exchangers. Heat capacity change of the heat carrier fluid in the ground heat exchanger is also considered. From assumption that all ground heat exchangers are in the same size, the volume of the heat carrier fluid in all ground heat exchangers and the surface area of the all ground heat exchangers are the same. When the volume and surface area are presented by V_f and A_{p-out} respectively, the following equation is obtained as respect to all circuits.

$$c_{1f} \rho_{1f} V_m \frac{dT_f}{dt} = -c_{1f} \rho_{1f} \frac{G_{1f}}{m} (T_{pout} - T_{pin}) + K_{p-out} A_{p-out} n (T_s(r_{p-out}, t) - T_f) \quad (6)$$

By using Equation (6) and giving other parameters, outlet temperature T_{pout} can be obtained. The ground temperature on the surface of ground heat exchanger $T_s(r_{p-out}, t)$ in Equation (6) is expressed by using average temperature of the surface temperatures of all ground heat exchangers calculated by Equation (1).

$$T_s(r_{p-out}, t) \cong T_s(r_{p-outave}, t) = T_{s0} + \frac{\sum_{i=1}^{m \cdot n} \Delta T_{si}(r_{p-out}, t)}{m \cdot n} \quad (7)$$

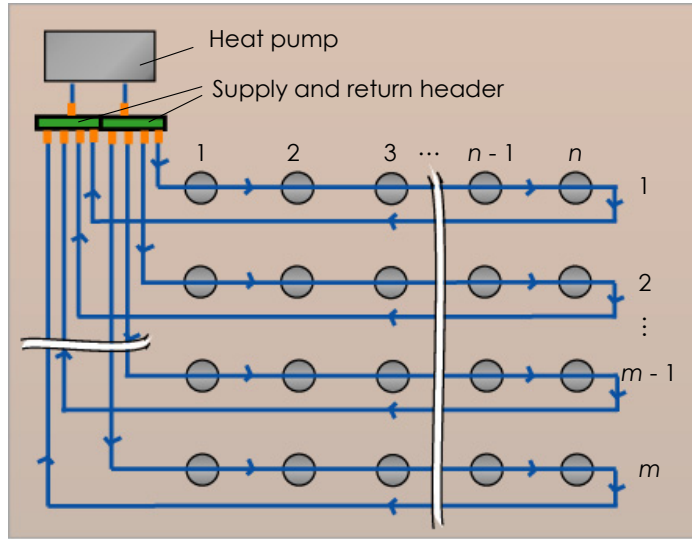


Figure 3: Multiple ground heat exchangers with parallel and series circuits

3.2 Simulation method of cooling tower

The cooling capacity of cooling tower Q_r is approximated as a function of the inlet temperature T_{rin} , the ambient wet bulb temperature T_{aWB} and flow rate G_r .

$$Q_r \cong f(n_r, T_{rin}, T_{aWB}, G_r) \quad (8)$$

Here, the number n_r is determined by specification of the cooling tower. The ambient wet bulb temperature T_{aWB} is calculated by using iterative method and giving ambient dry bulb temperature T_{aDB} and ambient absolute humidity x_a . The value T_{aDB} and x_a are obtained from weather data.

In addition, the outlet temperature of cooling tower T_{rout} is expressed by the following equation.

$$T_{rout} = T_{rin} - Q_r / (c_w \rho_w / 3600) \quad (9)$$

3.3 Simulation method of heat pump unit

As shown in the previous report (Katsura et al., 2009), the outlet temperature in primary side of heat pump unit T_{1out} is calculated by giving the inlet temperature in primary side T_{1in} , outlet temperature in secondary side T_{2out} , flow rate in primary side G_{1f} , flow rate in secondary side G_{2f} and heat demand of building Q_2 as calculated conditions. First, the coefficient of

performance (COP) of a heat pump unit η can be approximately expressed as a function according to T_{1in} , T_{2out} , G_{1f} , G_{2f} , and Q_2 .

$$\eta \cong f(T_{1in}, T_{2out}, G_{1f}, G_{2f}, Q_2) \quad (10)$$

Electric power consumption of the heat pump unit E_{hp} is calculated by the following equation.

$$E_{hp} = \frac{Q_2}{\eta} \quad (11)$$

Then, amount of heat extraction of the heat pump unit Q_1 is obtained by the next heat balance.

$$Q_1 = Q_2 - E_{hp} \quad (12)$$

On the other hand, Q_1 can be expressed by the following.

$$Q_1 = c_{1f} \rho_{1f} G_{1f} (T_{1in} - T_{1out}) \quad (13)$$

From Equation (12) and Equation (13), Equation (14) is derived.

$$T_{1out} = T_{1in} - \frac{Q_1}{c_{1f} \rho_{1f} G_{1f}} = T_{1in} - \frac{Q_2 - E_{hp}}{c_{1f} \rho_{1f} G_{1f}} \quad (14)$$

4 DESIGN AND PERFORMANCE PREDICTION OF GROUND SOURCE HEAT PUMP SYSTEM COMBINED COOLING TOWER

An office building shown in Figure 4 was completed in Kitakyushu City, on December 2010. The total area is approximately 10,000 m². The building has a lot of energy-saving technologies such as the hybrid ventilation system, PV system, high-efficiency lighting, building energy management system and so on. The GSHP system is also installed in the building. The design and construction works of GSHP system was carried out basis on the simulation result with the developed simulation tool.

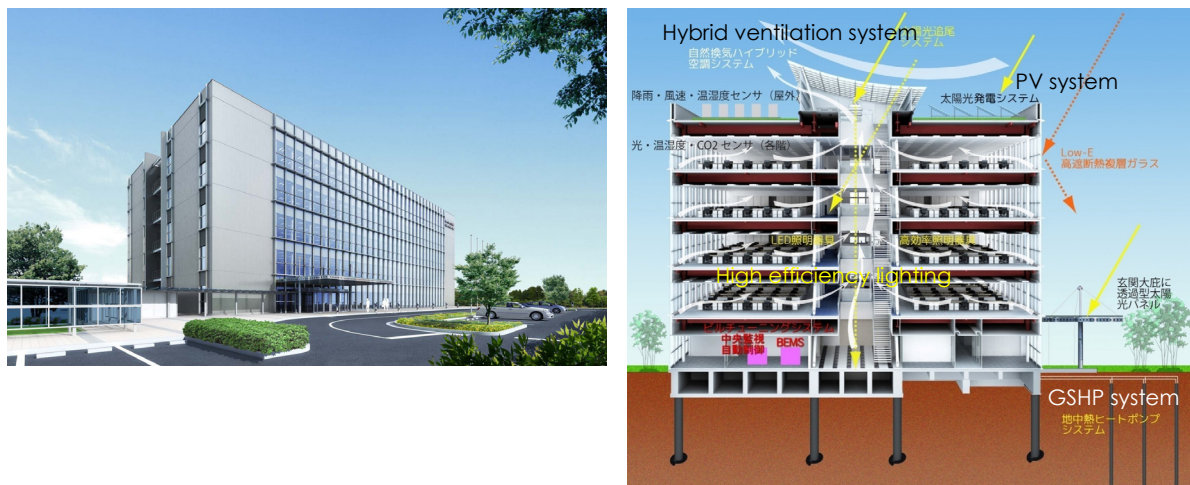


Figure 4: Appearance of new office building (Left) and energy saving technologies in the building (right)

4.1 Calculated condition

The calculated condition of GSHP system is shown in Table 1. Multi-split system (Katsura et al., 2008) is applied in the secondary side. Air conditioning area provided by the GSHP system is 2,271 m², which is 21.7% of the whole building. Figure 5 shows cooling and heating load of the GSHP system. The maximum cooling load and heating load are 254 kW and 145 kW, respectively. The water source multi-split heat pump has scroll compressors, whose total power is 100 HP. The COP of heat pump varies according to inlet temperature, flow rate and load factor as shown in Figure 6. Although the COP is changed by the room temperature, the temperature is regarded as constant in this simulation. Therefore, Equation (10) is modified as the following.

$$\eta \equiv f(T_{in}, G_{f1}, Q_1) \quad (15)$$

The combined GSHP system also has variable water control (VWV) in the primary side (Katsura et al., 2011). The set temperature difference is 5.0 °C during cooling operation and 3.5 °C during heating operation, respectively.

Borehole double U-tube heat exchangers (BHEX) with the length of 80 m are used and silica sand is filled in the BHEXs. The ground temperature is set as 16.5 °C. The typical soil effective thermal conductivity of 1.5 W/m/K and thermal capacity of 3,000 kJ/m³ are given.

Table 1: Calculated condition of GSHP system installed in new office building

Air conditioning area		2271m ²
Heat pump and subsystem	Heat pump	Water source multi-split heat pump Compressor : 100HP Rated cooling output : 280 kW
	Cooling tower	Closed type
	Circulation pump in primary side	Rated flow rate : 960 L/m Rated electric power : 7.5 kW VWV control
Ground heat exchanger	Specification	Borehole double U-tube
	Borehole diameter	150 mm
	Grout	Silica sand (Effective thermal conductivity : 1.8 W/(m·K))
	U-tube	HDPE 25A
Soil property	Length	80 m
	Average ground temperature	16.5 °C
	Effective thermal conductivity	1.5 W/(m·K))
	Thermal capacity	3000 kJ/(m ³ ·K))

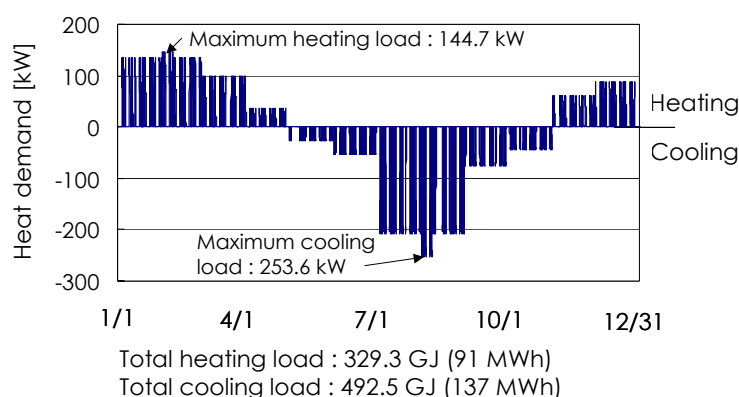


Figure 5: Heating and cooling load of the GSHP system

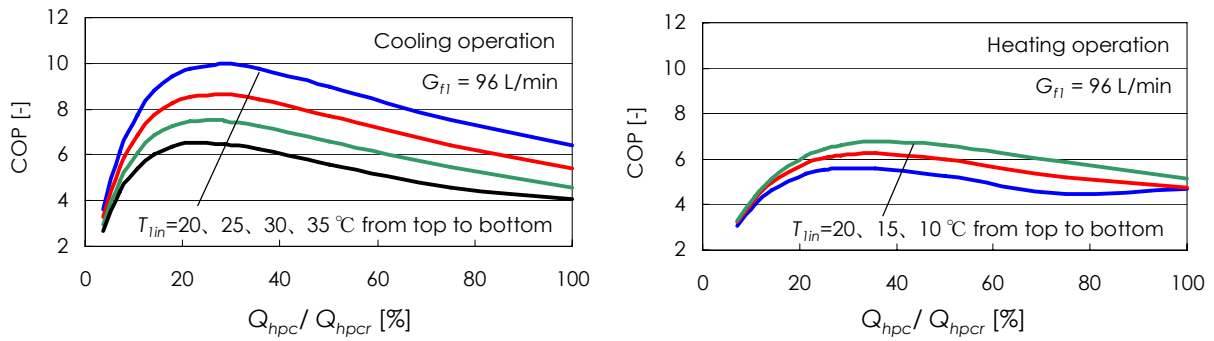


Figure 6: COP of multi-split heat pump installed in new office building

4.2 Design process using simulation program

Basis on the calculated condition, the number of BHEXs and specification (size) of cooling tower were determined according to the following procedure.

1. The GSHP system operation without the cooling tower is simulated for one year. Then minimum length of the BHEXs is determined in the condition of minimum $T_{1inmin} \geq 10$ °C.
2. The GSHP system combined the cooling tower is calculated for five years. The optimum size of the cooling tower is determined in the range of maximum inlet temperature $T_{1inmax} \leq 35$ °C. The optimum size is the size that yields the highest cooling seasonal performance factor (SPF).

Consequently, the number of BHEXs was 50 boreholes and rated capacity of cooling tower was 91 kW (20 RT).

4.3 Result of performance prediction

As the simulation result of the GSHP system operation combined cooling tower, Figure 7 indicates variations of extracted heat from the ground QT_{exg} , injected heat to the ground QT_{ing} and released heat to the air via the cooling tower QT_{inr} . Approximately 13~17 % of cooling exhaust heat ($=QT_{ing} + QT_{inr}$) is released to the air via the cooling tower. Next, variations of maximum inlet temperature T_{1inmax} and minimum inlet temperature T_{1inmin} are shown in Figure 8. The maximum inlet temperature T_{1inmax} raises only 0.6 °C from 1st year to 5th year. The maximum inlet temperature T_{1inmax} is less than 35 °C and minimum inlet temperature T_{1inmin} is more than 10 °C. Also, intense temperature variation is not observed. Variations of heat pump COP and SPF are shown in Figure 9. SPF is calculated by the following equation.

$$SPF = \frac{QT_{hp}}{ET_{hp} + ET_p + ET_r} \quad (16)$$

The calculated SPF during cooling period is around 5.5. Thus, the GSHP system can operate with high efficiency long term.

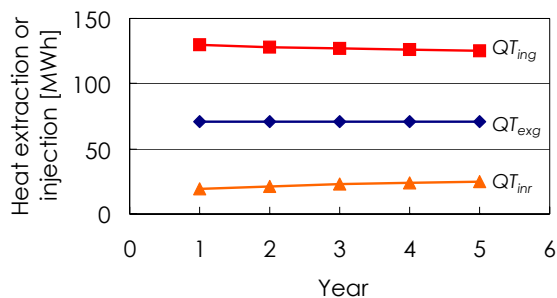


Figure 7: Variation of QT_{exg} , QT_{ing} and QT_{inr}

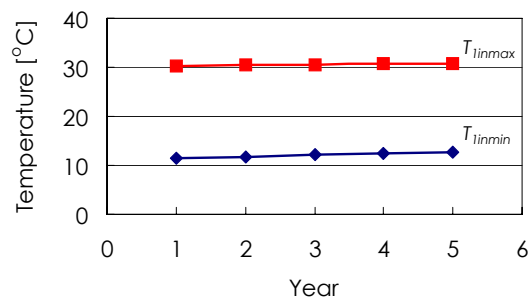


Figure 8: Variation of T_{1inmax} and T_{1inmin}

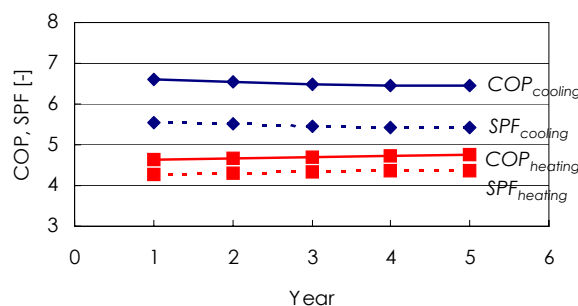


Figure 9: Variation of COP and SPF

4.4 Investigation of system connection and operation

Design and operating method of the GSHP system combined cooling tower is investigated by using the simulation tool. In this paper, system connection sequence and operation are changed and variations of SPF and injected heat to the ground according to elapsed time are compared. Table 2 indicates the calculated conditions. Other calculated conditions are the same as the ones in Chapter 4.1. CASE0 is not combined the GSHP system. The cooling tower is not used and only the GSHP system is applied. In CASE1 and CASE2, cooling tower is connected in series with BHEXs. The water is circulated to the cooling tower ahead of the BHEXs in CASE1 and to the BHEXs ahead of cooling tower in CASE2. The BHEXs and cooling tower are connected in parallel in CASE3. In CASE4, the water is circulated to the BHEXs only from July to August. Then the only cooling tower with capacity of 182 kW (40RT) is used as the heat source on the other months.

Figure 10 shows variation of released heat to the air via the cooling tower QT_{inr} and cooling SPF against CASE0~CASE4. The SPF of CASE1 is the largest. It suggests that the best system connection is that the BHEX and cooling tower are connected in series and the water is circulated to the cooling tower ahead of the BHEX. On the other hand, the SPF of CASE4 is the smallest. This indicates that it is better to operate both the BHEXs and cooling tower at the same time.

In addition, the authors assumed introduction of the cooling tower control by inlet temperature of heat pump unit T_{1in} as shown in Figure 11. When the temperature T_{1in} becomes higher than set temperature T_{rout} , the cooling tower is turned on. The temperature T_{rset} was changed and cooling SPF with variation of T_{rset} was calculated.

Figure 12 is cooling SPF of fifth year's operation with variation of T_{rset} . This result indicates the cooling tower control can improve the system performance. The highest SPF of 5.415 is obtained in condition of $T_{rset} = 25$ °C.

Table 2: Condition of cooling tower and system connection sequence

	Rated capacity of cooling tower	System connection sequence
CASE0	Nothing	
CASE1	91 kW (20 RT)	
CASE2	91 kW (20 RT)	
CASE3	91 kW (20 RT)	
CASE4	182 kW (40 RT)	July, August: Other months:

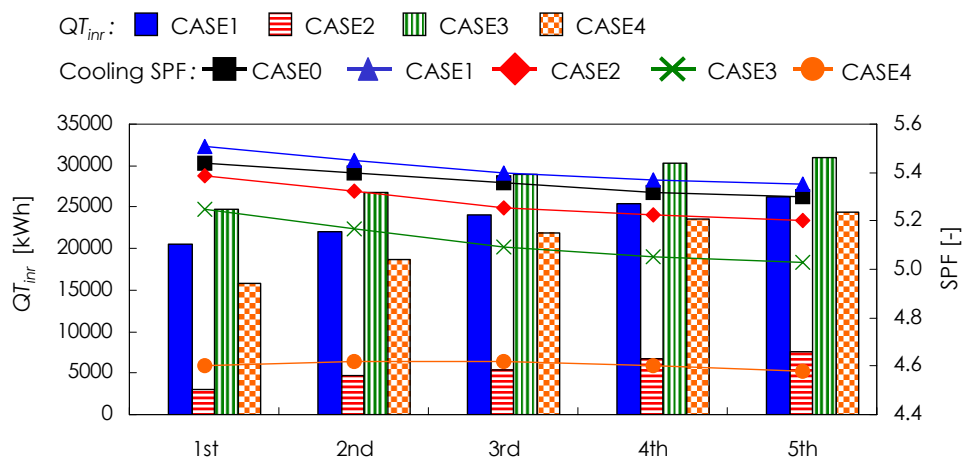


Figure 10: Variation of QT_{inr} and cooling SPF

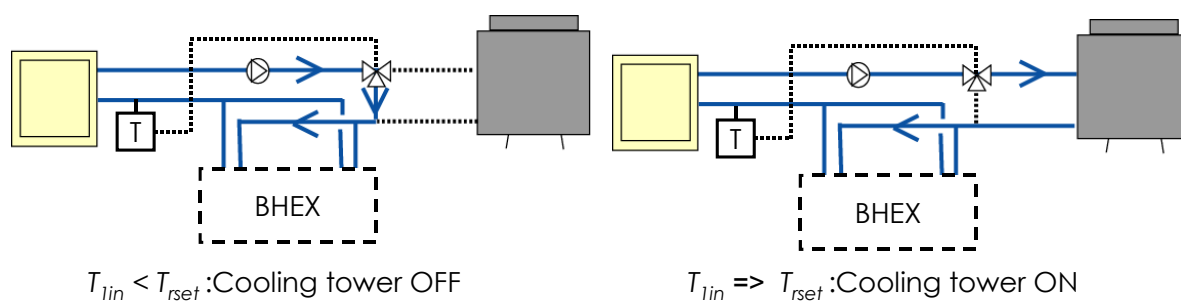


Figure 11: Concept diagram of cooling tower control by temperature T_{1in}

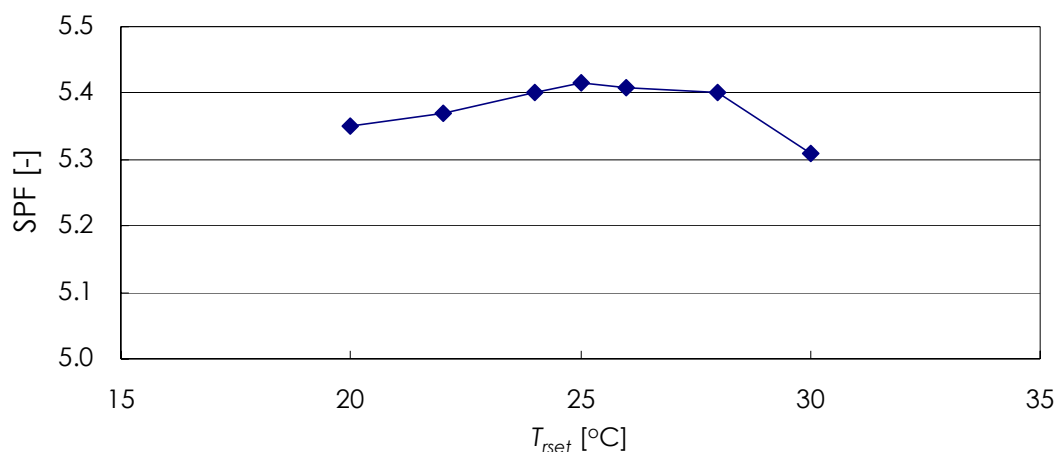


Figure 12: Variation of cooling SPF with variation of T_{rset}

5 SUMMARIES

- 1) A computer aided simulation tool for the GSHP system combined cooling tower was developed. The outlines of the simulation were introduced
- 2) The GSHP system combined cooling tower installed to an office building in Kitakyushu City was predicted. The result showed the combined GSHP system could operate with high cooling seasonal performance factor of 5.4.
- 3) Connection sequence and operation method of the combined GSHP system were investigated. The result showed that the GSHP system provides the highest cooling SPF when cooling tower is connected in series with BHEXs and the water as heat carrier fluid is circulated to the cooling tower ahead of the BHEXs. The system performance is additionally improved by applying cooling tower control.

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7 ACKNOWLEDGEMENT

This work is supported by the technological development project "Development of low flow circulation ground source heat pump multi-split system and its design and operation method" to prevent global warming of the Ministry of Environment and NEDO project "Experimental project of next generation energy saving architecture".

The authors appreciate Daikin Industry Cooperation for assistance of developing, operating, evaluating the heat pump unit. We also grateful to Mr. Y. Fujiwara in the Fujiwara Environmental Science Institute Ltd, Mr. S. Hori, Mr. T. Okawada and Mr. S. Nagae of master course and undergraduate student in Hokkaido University at the time.

NOMENCRATURE

A : Area [m^2], c : Specific thermal capacity [$kJ/kg/K$], E : Electric power [W], ET : Total electric power [Wh], G : Flow late [m^3/s], K : Overall heat transfer coefficient [$W/m^2/K$], n_r : coefficient of cooling tower, Q : Heating or cooling output, injected or extracted heat [W], QT : Total heating or cooling output, total injected or extracted heat [Wh], q^* : Non-dimensional heat flux ($= q / q_0$), q : Heat flux [W/m^2] [-], q_0 : Unit heat flux ($=1$) [W/m^2], T : Temperature [$^{\circ}C$], T^* : Non-dimensional temperature ($= \lambda \Delta T / r / q$) [-], t : Time [h], t^* : Fourier number ($= at / r^2$) [-], V : Volume [m^3], x : absolute humidity [$kg/kg(DA)$], η : Heat pump COP [-], ρ : Density [kg/m^3]

Subscript

a : ambient air, aWB : ambient air wet bulb, ave : average, d : Distance, f : Thermal medium, hp : Heat pump, p : Ground heat exchanger or pump, r : Cooling tower, s : Soil, $s0$: Soil initial, WB : Wet bulb, 1 : Primary side, 2 : Secondary side, in : Inlet, out : Outlet, $-out$: Outside, $-C$: For cylindrical heat source, $-L$: For line heat source