OPTIMAL OPERATION FOR HVAC SYSTEM WITH SEASONAL UNDERGROUND THERMAL STORAGE SYSTEM

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ABSTRACT

The present paper proposes an optimal operational strategy of an actual HVAC system with a seasonal underground thermal storage system using simulation. The simulation is a powerful tool for the system because it is difficult to try various operational methods experimentally in the actual system due to the long heat transfer time in the underground. This paper develops the whole system simulation by connecting an underground thermal model, which can simulate the heat exchange between the ground and the pipes buried in the foundation piles, and the models of mechanical components in the air-conditioning system. The optimum operational strategy found using simulation, which is to minimize the energy consumption of the whole system, can save the energy consumption by approximately 30 % and improve system COP from 3.02 to 5.04 compared with the values of the present operational method.

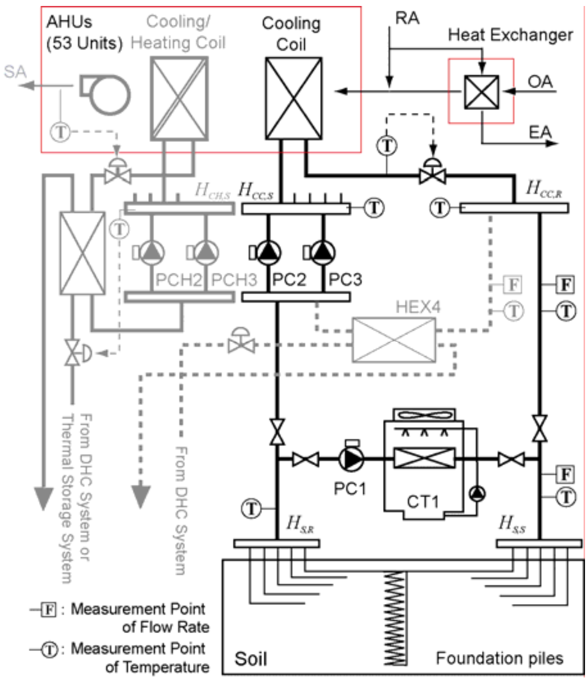

Figure 1 Air-conditioning system.

KEYWORDS

Commissioning, Underground Thermal Storage, Optimization, FEM, HVAC System

INTRODUCTION

The present paper proposes a method to optimize the performance of an HVAC system with seasonal thermal storage using simulation. In this system, water cooled by cooling towers is circulated through pipes pre-installed in foundation piles to cool the ground in winter, and in summer the cool water is fed to cooling coils of AHU to perform pre-cooling of supply air. Although the system utilizes natural energy, if operated improperly the system may consume more energy than a common HVAC system with refrigerators, due to using a large amount of energy for water circulation.

This paper discusses the optimal operational method of an actual HVAC system in Japan using whole system simulation. Conducting various case studies using the simulation, the optimum operational strategy is attained.

INFORMATION OF THE ANALYZED BUILDING

The building analyzed in this paper is located in Takamatsu, Japan, and has an HVAC system with underground thermal storage. Figure 1 shows an outline of the HVAC system. The building has no artificial heat source and is air-conditioned by means of cooling water from the underground and heating

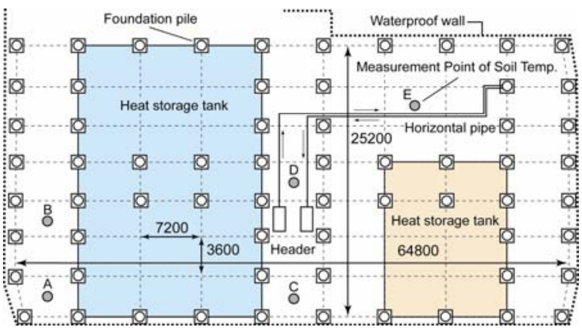

Figure 2 Layout plan of foundation piles.

Figure 3 Construction of a pile

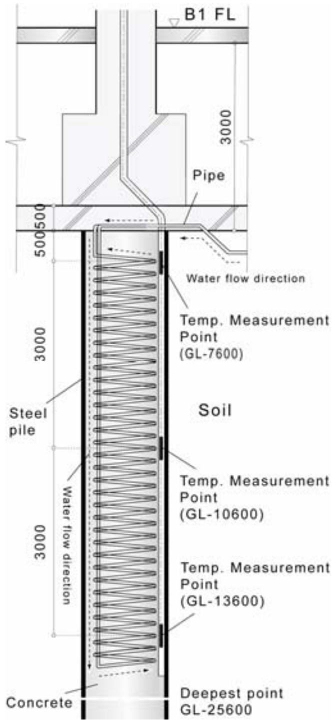

Figure 4 Cross-section of a pile

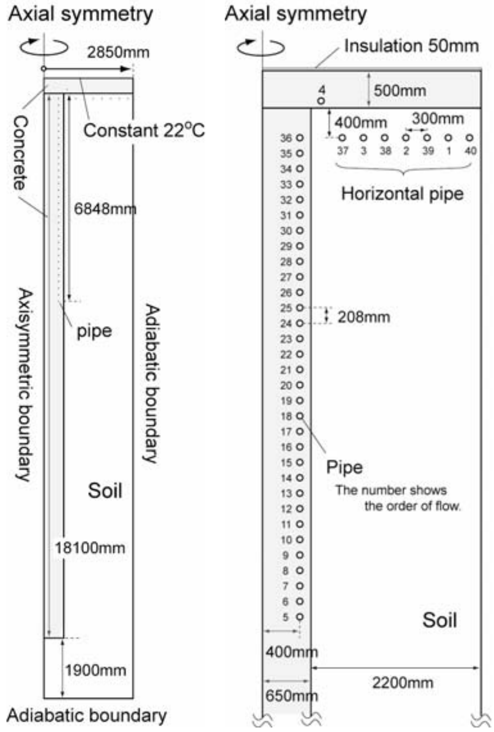

Figure 5 Underground Thermal Model (Left: Entirety, Right: Enlargement)

and cooling water supplied by a district heating and cooling (DHC) plant. There are 53 air-handling units and each unit has two coils; a cooling coil supplied with heat from underground and the DHC plant, and a cooling and heating coil supplied with heat form the DHC plant and water thermal storage systems placed in the building.

The building exchanges the heat with underground using foundation piles. Figure 2 shows the layout plan of the foundation piles, and Figure 3 is a photograph when the piles were constructed. Figure 4 shows a cross-section of a pile. The spiral pipes buried in the pile are connected to the headers $H_{S,S}$ and $H_{S,R}$ via pipes placed horizontally underground at a depth of 400mm. During winter the cooling tower CT1 and the pump PC1 are operated and the cooling energy is charged, and during summer the pumps PC2 and PC3 are operated and the cooling energy is discharged from underground. The flow rate of water passing through the cooling coil is controlled so that the outlet water temperature of the coil attains the temperature set point.

Table 1 Actual performance of the system

		1st year	2nd year
Storage	Period	2/1/05 to 4/30/05	12/1/05 to 4/30/06
	Operational condition	Ws = 50m^3/h	Ws = 45m^3/h
	Qss	213.4 GJ	373.6 GJ
	Ess	27.0 GJ	55.5 GJ
	ηss	7.9	6.7
Discharge	Period	7/1/05 to 10/30/05	7/1/06 to 10/30/06
	Operational condition	θχχ,ωo = 23°C, and water from DHC flowed through HEX4 so that the outlet water temperature of header Hss became 19°C	θwco = 25°C
	Qsd	145.1 GJ	201.4 GJ
	Esd	20.4 GJ	13.0 GJ
	ηsd	7.1	15.5
ηcop		3.06	2.94

The system began to be operated since February 2005. The operational conditions and the acutual performance of the system are shown in Table 1. The energy consumption, the efficiency of operation and the system COP η_{cop} are defined as follows.

$$E_{ss} = E_{ps} + E_{ct,f} + E_{ct,p} \tag{1}$$

$$E_{sd} = E_{pd} \tag{2}$$

$$\eta_{ss} = Q_{ss} / E_{ss} \tag{3}$$

$$\eta_{sd} = Q_{sd} / E_{sd} \tag{4}$$

$$\eta_{cop} = Q_{sd} / \left(E_{ss} + E_{sd}\right) \tag{5}$$

Before the discharge operation of the 2nd year began, simple case studies had been conducted and the discharge operational method was changed based on the results of the case studies. Though η_{sd} was improved from 7.1 to 15.5, η_{cop} was approximately 3.0, which was equal to the value of general current artificial heat pumps. Because the value seems to be improved if η_{ss} is improved, the inverter of the cooling tower fan was installed in December, 2006.

DEVELOPMENT OF SIMULATION

This section discusses how to develop the simulation of the HVAC system, which consists of a physical

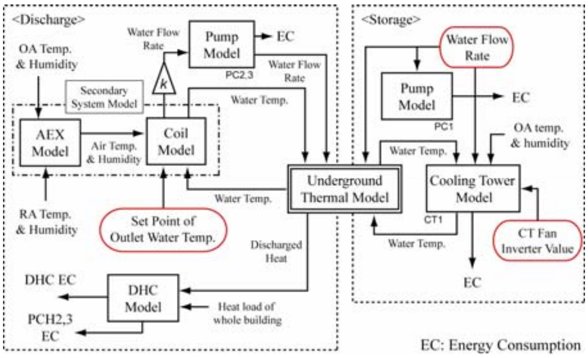

Figure 6 Diagram of the whole system simulation

model of the underground thermal storage through which heat is exchanged between circulated water in the pipes embedded in the building foundation piles and the ground, and models of components in the HVAC system such as a cooling tower, heat exchangers, cooling coils, and pumps. The schematic diagram of the whole system simulation is shown in Figure 6. By connecting these models, the system simulation is developed on MATLAB/Simulink (Miyata and Yoshida et.al 2005 and 2006).

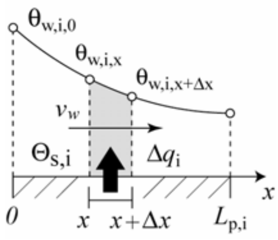

Figure 7 Heat balance of Pipe i

Underground Thermal Model

The shape of the model and the assumed boundary conditions are shown in Figure 5. This model is an axially symmetrical model and can estimate the heat transfer of a pile. The radius of the model r_{utm} is determined as follows.

$$r_{utm} = \sqrt{\frac{d_{ns} d_{ew}}{\pi}} \tag{6}$$

Although the actual water pipe in the pile is arranged in spirals, in the model independent annular-shape pipes are arranged in the vertical direction. Each pipe is numbered as shown in Figure 5, and the circulated water flows from Pipe 1 to Pipe 40 in order. The upper boundary condition of the model is set constant (22°C) because the measured temperature is constant.

Because the independent, annular-shape pipes are arranged in the model, the temperature differences between the neighboring pipes are calculated by considering the heat balance over the infinitesimal distance Δx as shown in Figure 7.

$$\Delta q_i(t) = a_p\left(\Theta_{s,i}(t) - \theta_{w,i,x}(t)\right)\Delta x \tag{7}$$

$$\Delta q_i(t) = c_w v_w(t) \Delta\theta_{w,i,x}(t) \tag{8}$$

$$a_p = 2\pi r_{po}\alpha_{pe} \tag{9}$$

$$\Delta\theta_{w,i,x}(t) = \theta_{w,i,x+\Delta x}(t) - \theta_{w,i,x}(t) \tag{10}$$

$$\frac{1}{r_{po}\alpha_{pe}} = \frac{1}{r_{pi}\alpha_{pi}} + \frac{\log(r_{po}/r_{pi})}{\lambda_p} \tag{11}$$

Based on these equations, the inlet and outlet water temperature of the pipe can be calculated.

Table 2 Thermo physical properties

Materials	Thermal conductivity [W/m·K]	Specific heat [J/kg·K]	Density [kg/m^3]
Soil	1.29	1640	1964
Concrete	1.64	880	2450
Insulation	0.028	1300	700
Pipe	0.337	1900	940

The model is divided into approximately one thousand elements using FEM software (ANSYS 2007). The initial temperature of the each point is set at the average soil temperature measured at Point A to E in Figure 2. Table 2 shows the thermo physical properties of the materials used in the model, which are determined based on the result of boring exploration.

The inputs of the model are inlet water temperature $\theta_{s,wi}$, which is equal to the water temperature in Pipe 1, and water flow rate of the pipe w_s, and the output is the outlet water temperature $\theta_{s,wo}$, which is equal to the water temperature in Pipe 40.

Verification of the Underground Thermal Model

In order to verify the accuracy of the model, the outlet water temperature from underground is calculated inputting the measured inlet water temperature to the ground and the water flow rate to the model, and the simulated temperature is compared with the measured outlet water temperature. Data measured from 2/1/2005 to 10/30/2006 are used.

Figure 8 to 11 show a comparison between simulated

Table 3 Verification results of the underground thermal model

Period			Transferred Heat			Outlet Water Temp.	
			Measured	Simulated	Error	Error	RMSE
1st Year	Storage	From 2/1/05 to 4/30/05	213.4 GJ	221.85 GJ	3.96%	0.14 K	0.33 K
	Discharge	From 7/1/05 to 10/30/05	145.1 GJ	141.2 GJ	-2.69%	0.02 K	0.26 K
2nd Year	Storage	From 12/1/05 to 4/30/06	373.6 GJ	398.2 GJ	6.58%	0.18 K	0.36 K
	Discharge	From 7/1/06 to 10/30/06	120.0 GJ	125.8 GJ	4.83%	-0.13 K	0.35 K

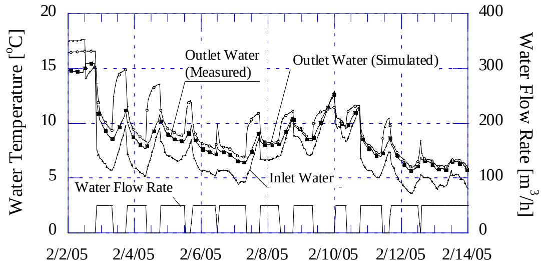

Figure 8 Water temp. (Storage period)

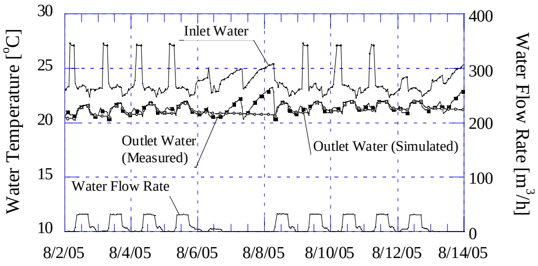

Figure 9 Water temp. (Discharge period)

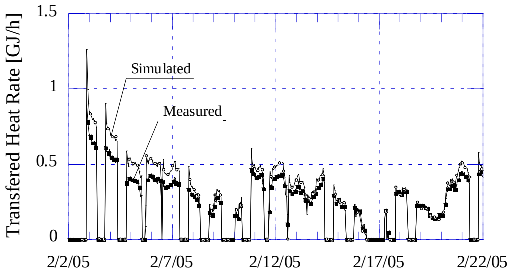

Figure 10 Transferred heat (Storage period)

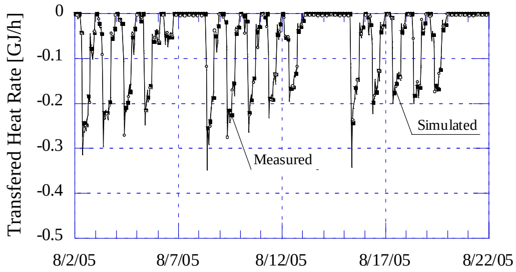

Figure 11 Transferred heat (Discharge period)

and measured data. Table 3 shows the error between the measured and simulated data. The error e_{ave} and the root mean squre error (RMSE) e_{rmse} are defined as follows.

$$e_{ave} = \sum_{n=1}^{N}\left(x_n - \hat{x}_n\right) \Big/ N \qquad (12)$$

$$e_{rmse} = \sqrt{\sum_{n=1}^{N}\left(x_n - \hat{x}_n\right)^2 \Big/ N} \qquad (13)$$

The error of the outlet water temprature ranges from –0.13K to 0.18K and RMSE ranges from 0.26K to 0.36K. The results show that the model can simulate the outlet water temperature accurately. The error of the transfferd heat amount is slightly large, especially during the storage period. As can be seen from Figure 10, the difference is larger in the beginning of the storage period. The reason may be the intial soil temperature or the boundary conditions of the model, etc. Clarification of the reason is a future task.

Cooling Tower Model

The cooling tower in the building is the indirect-contact cooling tower. The cooling tower model is expressed as follows.

$$Q_{ct} = f_{ct} g_{ct} (h_{ct,ai} - h_{ct,ao}) = c_w w_{ct} (\theta_{ct,wo} - \theta_{ct,wi}) \qquad (14)$$

$$U_{ct} = \frac{K_{ct} V_{ct}}{f_{ct} g_{ct}} = \int \frac{dh}{h_w - h_a} \qquad (15)$$

$$\frac{U_{ct} f_{ct} g_{ct}}{w_{ct}} = -c_w \int \frac{d\theta_{ct,w}}{h_w - h_a} \qquad (16)$$

$$Q_{ct} = c_w w_s (\theta_{s,wo} - \theta_{s,wi}) \qquad (17)$$

$$Q_{ct} = U_{ch} A_{ch} \frac{\left(\theta_{s,wo} - \theta_{ct,wi}\right) - \left(\theta_{s,wi} - \theta_{ct,wo}\right)}{\ln\left\{\left(\theta_{s,wo} - \theta_{ct,wi}\right) / \left(\theta_{s,wi} - \theta_{ct,wo}\right)\right\}} \qquad (18)$$

$$E_{ct} = \left(c_0 + c_1 f_{ct} + c_2 f_{ct}^{\ 2} + c_3 f_{ct}^{\ 3}\right) + E_{ct,p} \qquad (19)$$

The equations (14) to (16) are the typical model of cooling tower and calculate the heat transferred between the outdoor air and the water circulated in

Table 4 Specification of the components in the HVAC system

Equipment		Specification	Input(s)	Output(s)	Error	RMSE
CT1		Cooling Capacity 498.8 kW,Wet-bulb outdoor air temperature 27℃,inlet chilled water temperature 37℃,outlet chilled water temperature 32℃,Circulated water flow rate 1300 ℓ/min,Air flow rate 1188 m3/min,Rated energy consumption of fan 6.0kW,Rated energy consumption of pump1.6kW	Inlet water temp., Water flow rate, outdoor air temp. and humidity, Inverter value of fan	Outlet water temp., Energy consumption	0.06°C (2.91%)	0.21°C (10.2%)
Pump	PC 1	Rated energy consumption 15kW, Rotation speed 29.2rps, Pressure 30m, Water flow rate 25kg/s	Water flow rate, Inverter value	Energy consumption	-0.011kW (-0.35%)	0.05kW (1.54%)
	PC 2,3	Energy consumption 5.5kW, Rotation speed 29.0rps, Pressure 30m, Water flow rate 6.7kg/s	Water flow rate, Inverter value	Energy consumption	0.02kW (0.43%)	0.20kW (3.97%)
Cooling Coil		Rated heat transfer rate 4.2kW, Water flow rate 0.2kg/s, Number of rows of tubes 22, Number of tubles per row 4, Length of finned section 480mm, Height of finned section 1100mm, Width of finned section 180mm, Rated return air flow rate 3940m^3/h, Rated outdoor air flow rate 660m^3/h	Intet air temp. and humidity, Air volume, Inlet water temp., Water flow rate	Outlet air temp. and humidity, Outlet water temp.	-0.17°C (-5.95%)	0.33°C (11.1%)
Heat Exchanger		Rated energy consumptino 2.8kW,Air flow rate 3500m^3/h,Rated exchange efficiency of temperature 77%,Rated exchange efficiency of enthalpy 60.5%	Inlet outdoor air temp. and humidity, Inlet return air temp. and humidity, Air volume	Outlet air temp. and humidity	-0.19°C (-5.76%) -0.300g/kgDA (6.39%)	0.22oC (6.98%) 0.442g/kgDA (9.33%)

cooling tower (ASHRAE 2004). The equations (17) and (18) are the model of the heat exchanger and calculate the heat transferred between the circulated water and the water to the underground (ASHRAE 2001).

The model is calibrated determining A_{ch} and $c_0,...,c_3$ using the data measured in the building so that the difference of the simulated and the measured values becomes minimum. The specification of the cooling tower CT1, inputs and outputs of the model, and the verification results of the model accuracy using the measured data are shown in Table 4.

Pump Model

The model is expressed as follows (Clark 1985, Wang, Yoshida and Miyata 2004).

$$C_h = a_0 + a_1 C_f + a_2 C_f{}^2 + a_3 C_f{}^3 + a_4 C_f{}^4 \tag{20}$$

$$\eta_p = e_0 + e_1 C_f + e_2 C_f{}^2 + e_3 C_f{}^3 + e_4 C_f{}^4 \tag{21}$$

$$E_p = \frac{w_p \Delta P_p}{\eta_p \eta_{motor} \eta_{inv} \rho_w} \tag{22}$$

The model is calibrated determined η_{inv} using the measured data. The specification of PC1 and PC2,3 and the verification results of the model accuracies are shown in Table 4.

Rotary Heat Exchanger Model

The model is expressed as follows.

$$h_{oa,o} = h_{oa,i} - \eta_h \left(h_{oa,i} - h_{ra,i} \right) \tag{23}$$

$$\theta_{oa,o} = \theta_{oa,i} - \eta_\theta \left(\theta_{oa,i} - \theta_{ra,i} \right) \tag{24}$$

$$x_{ao,o} = \frac{1}{c_v \theta_{oa,o} + L_w} \left(\frac{h_{oa,o}}{\rho_a V_{oa}} - c_a \theta_{oa,o} \right) \tag{25}$$

$$\eta_h = a_{h0} + a_{h1} V_{oa} \tag{26}$$

$$\eta_\theta = a_{\theta 0} + a_{\theta 1} V_{oa} \tag{27}$$

The specification of the heat exchanger, the inputs and outputs of the model, and the verification results are shown in Table 4.

Coil Model

The model is expressed as follows (Husaunndee et.al 1998).

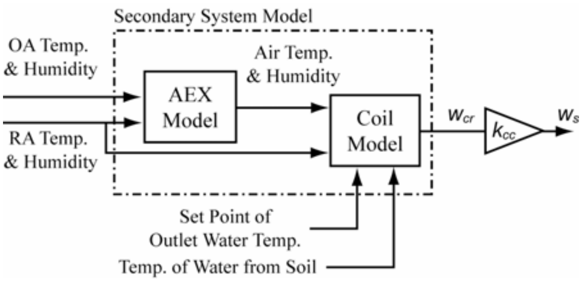

Figure 12 Secondary system model

$$Q_{cc} = \max(Q_{cd}, Q_{cw}) \tag{28}$$

$$Q_{cd} = \eta_{cd} C_{cd} (\theta_{cc,ai} - \theta_{cc,wi}) \tag{29}$$

$$Q_{cw} = \varepsilon_{cw} C_{cw} (\theta_{caw,i} - \theta_{cw,i}) \tag{30}$$

$$\varepsilon_{cd} = f(U_{cd}, C_{air}, C_{water}) \tag{31}$$

$$\varepsilon_{cw} = f(U_{cw}, C_{airs}, C_{water}) \tag{32}$$

$$C_{air} = m_{cc}(c_a + c_v x_{cc,ai}) \tag{33}$$

$$C_{airs} = m_{cc} c_{as} \tag{34}$$

$$C_{water} = w_{cc} c_w \tag{35}$$

The specification of the cooling coil and the verification results are shown in Table 4.

Secondary System Model

Although the building contains 53 cooling coils, the model of a typical coil is developed and the total water flow rate of all the coils is calculated by multiplying the estimated flow rate of the typical coil model by a coefficient k_{cc} as shown in Figure 12.

$$w_{cr} = f(\theta_{oa}, x_{oa}, V_{oa}, \theta_{ra}, x_{ra}, V_{ra}, \theta_{s,wo}, \theta_{cc,wo}) \tag{36}$$

$$w_s = k_{cc} w_{cr} \tag{37}$$

The coefficient k_{cc} is determined satisfing the following equation using measured data $\tilde{w}_s$.

$$\min_{k_{cc}} \left\{ \sum_{n=1}^{N} \left(k_{cc} w_{cr,n} - \tilde{w}_{s,n} \right)^2 \right\} \tag{38}$$

The value is determined from the measured operational data and is set to 47.4. Figure 13 shows the verification result of the model. RMSE of the model is 0.78°C (8.20%). The result shows that the model can calculate w_s accurately.

The inputs of the whole system simulation are the temperature and humidity of the outdoor air and the return air, the set point of water flow rate of the pump for storage, the inverter value of the cooling tower fan, and the outlet water temperature of the cooling coil for discharge operation.

OPTIMIZATION OF OPERATION

The present paper conducts case studies about the operaitonal method and seeks the optimal operational method which is to minimize the value of an

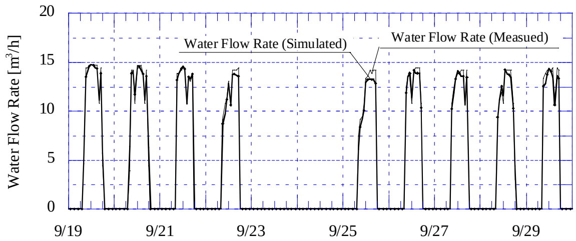

Figure 13 Verification result of the secondary system model

objective function. Although this is not proper optimization, this paper regard the operational method determined from the results of the case studies as the optimal method because the calculation for one year operation takes more than 30 hours and it is difficult to obtain the proper optimal solution.

This paper analyzes the water flow rate and the inverter value of the cooling tower fan for the storage operation and the coil outlet water temperature for the discharge operation. The calculation period is from December 1st to November 31st, and data measured from December 2004 to November 2005 is used to input the simulation. The period of the storage operation is from December to March and the period of the discharge operation is from June to October.

Objective Function

This paper defines the optimum operation as the operation method that minimizes the energy consumption of the whole system. Because the charged and discharged heat amount seem to be decreased when the enery consumpsion is decreased, the targeted discharge heat amount $Q_{sd,r}$ is set and the deficient heat from the artificial heat pump, whose system COP is η_{ahp} , is assumed to be supplied if the discharge heat amount is less than the target amount.

The objective function η_t is defined as follows.

$$\eta_t = \frac{\left(E_{ss} + E_{sd} + E_{ahp}\right)}{\left(\dfrac{Q_{sd,r}}{\eta_{ahp}}\right)} \tag{39}$$

$$E_{ahp} = \left(Q_{sd,r} - Q_{sd}\right) / \eta_{ahp} \tag{40}$$

The operational method whose value of η_t is the smallest is defined as the optimized operational

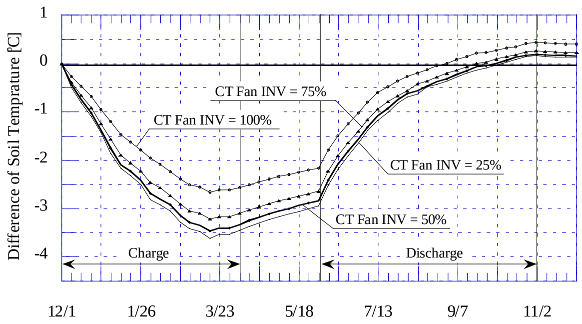

Figure 14 Soil temperature

method. In this paper, $Q_{sd,r}$ is set to 224GJ, which is the amount when w_s is 45m^3/h, f_{ct} is 100%, $\theta_{cc,wo}$ is 25°C, which are the actual operational method of the 2nd year.

Constrained Condition about Soil Temperature

It is desirable to maintain the soil temperature at the annual average outdoor air temperature in order to use the system cyclically over the years. If the soil temperature is raised more than the annual averaged temperature, the discharged heat amount might be decreased year after year. Therefore the end date of the discharged operation is determined so that the averaged soil temperature is not raisen than the temperature when the storage operation began. The soil temperature is defined as follows.

$$\overline{\theta_s}(t) = \frac{1}{N_{ele} V_{utm}} \sum_{n=1}^{N_{ele}} \left(\theta_{s,n}(t)\, V_{ele,n}\right) \tag{41}$$

Figure 14 show the calculation results of soil temperature when w_s is 45m^3/h and $\theta_{cc,wo}$ is 25°C and f_{ct} is changed as 25, 50, 75, 100%, and the end date of the discharge operation is set to October 31st. The soil temperature is fallen approximately 0.0368°C from the end date of the discharged operation until the beginning of the storage operation of next year. Thus, the end date of the discharegd

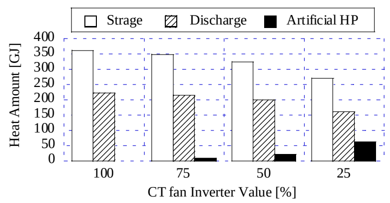

Figure 15 Heat Amount (w_s = 45m^3/h, $\theta_{cc,wo}$ =25°C)

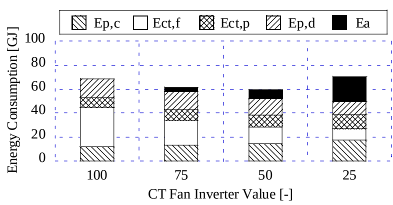

Figure 16 Energy consumption (w_s = 45m^3/h, $\theta_{cc,wo}$ =25°C)

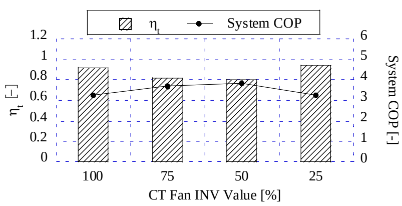

Figure 17 Efficiency (w_s = 45m^3/h, $\theta_{cc,wo}$ =25°C)

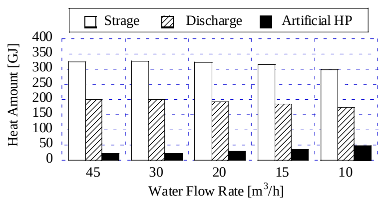

Figure 18 Heat Amount (f_{ct} = 50%, $\theta_{cc,wo}$ =25°C)

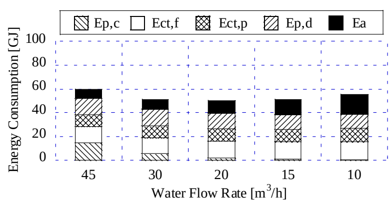

Figure 19 Energy consumption (f_{ct} = 50%, $\theta_{cc,wo}$ =25°C)

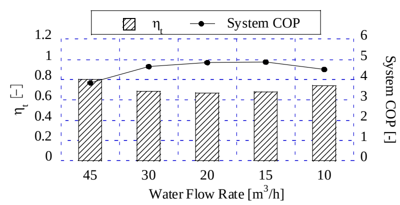

Figure 20 Efficiency (f_{ct} = 50%, $\theta_{cc,wo}$ =25°C)

Table 5 Calculation results for the storage operational method

Operational Method			Day when discharge is ended	Heat Amount [GJ]				Energy Consumption [GJ]						System COP	η_t
ws	fct	θwco		Strage	Discharge	Ratio	Artificial HP	Strage: Pump	Strage: CT Fan	Strage: CT Pump	Discharge	Artificial HP	Sum.		
Only Artificial HP			-	0	0	-	224.0	0	0	0	0	74.66	74.66	-	1.000
45m³/h	100%	25°C	10/16	361.5	224.0	62.0%	0.0	12.08	32.94	7.98	15.68	0.00	68.68	3.26	0.920
	75%		10/9	348.7	214.8	61.6%	9.1	13.23	21.16	8.75	14.98	3.05	61.16	3.70	0.819
	50%		9/29	326.0	200.5	61.5%	23.5	14.95	13.30	9.88	13.96	7.84	59.94	3.85	0.803
	25%		9/2	271.6	161.1	59.3%	62.8	17.76	9.06	11.74	10.91	20.94	70.41	3.26	0.943
30m³/h	100%		10/14	362.1	222.7	61.5%	1.3	4.77	33.77	8.18	15.52	0.42	62.67	3.58	0.839
	75%		10/8	349.3	214.2	61.3%	9.7	5.17	21.45	8.87	14.96	3.24	53.69	4.25	0.719
	50%		9/29	326.6	199.9	61.2%	24.1	5.81	13.39	9.95	13.87	8.03	51.05	4.65	0.684
	25%		9/3	273.4	162.9	59.6%	61.1	6.85	9.06	11.74	11.09	20.35	59.09	4.21	0.792
20m³/h	100%		10/14	356.7	219.8	61.6%	4.2	1.82	35.88	8.69	15.48	1.39	63.26	3.55	0.847
	75%		10/5	344.3	208.9	60.7%	15.0	1.96	22.59	9.34	14.54	5.01	53.44	4.31	0.716
	50%		9/21	322.4	193.2	59.9%	30.8	2.17	13.94	10.36	13.31	10.27	50.05	4.86	0.670
	25%		9/1	270.7	158.5	58.5%	65.5	2.49	9.15	11.86	10.78	21.83	56.11	4.62	0.752
15m³/h	50%		9/19	315.5	187.0	59.3%	37.0	1.05	14.42	10.72	12.26	12.33	50.78	4.86	0.680
10m³/h	50%		9/12	299.0	174.9	58.5%	49.1	0.36	15.23	11.31	11.96	16.37	55.23	4.50	0.740

operation is determined when the average soil temparature becomes 0.0368 °C and the discharged heat amount assumed to be zero after the date.

Optimization of Storage Operational Method

The case studies about the storage operation are conducted. w_s is assumed to be 45, 30, 20, 15 and 10m^3/h and f_{ct} is assumed to be 100, 75, 50 and 25%. In all cases, $\theta_{cc,wo}$ is set at 25 °C.

Figure 15 to 17 show the results of the case studies about f_{ct} when w_s is set at 45m^3/h. The results indicate that if f_{ct} is changed from 100% to 75, 50 and 25%, the storage heat amount is decreased approximately 3.5%, 9.8% and 24.9%, and the discharge heat amount is decreased approximately 4.1%, 10.5% and 28.1% respectively. Although the energy consumption of the cooling tower fan can be saved approximately 35.8%, 59.6% and 72.5 respectively, the energy consumptions of the pump and the cooling tower pump are increased because the operating time becomes longer. To compare with these four cases, η_t is minimum when f_{ct} is set at 50%.

When f_{ct} is set at 50%, w_s is changed from 45 to 30, 20, 15 and 10m^3/h. Figure 18 to 20 show the analyzed results about w_s. Although the energy consumption of the pump becomes smaller when w_s gets small, E_{ahp} gets larger because the discharged heat amount is smaller. To compare with these five cases, η_t is minimum when w_s is set at 20m^3/h.

Table 5 shows the calculation results of all cases. Based on the results, the operational method of w_s = 20m^3/h and f_{ct} = 50% is optimal.

Optimization of Discharge Operational Method

The case studies about the discharge operation are conducted. $\theta_{cc,wo}$ is set at 23, 24, 25 and 26°C and two storage operational methods (w_s = 30m^3/h, f_{ct} = 50% and w_s = 20m^3/h, f_{ct} = 50%) are tried.

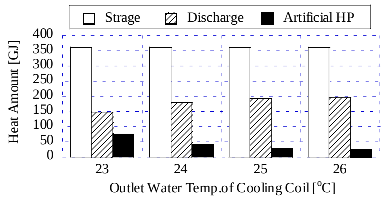

Figure 21 Heat Amount (w_s = 20m^3/h, f_{ct} = 50%)

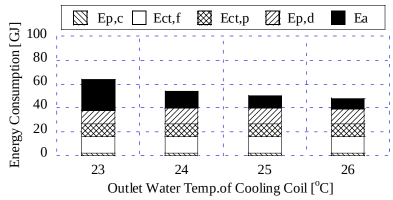

Figure 22 Energy consumption (w_s = 20m^3/h, f_{ct} = 50%)

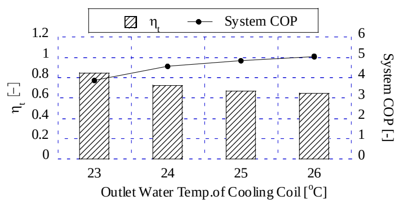

Figure 23 Efficiency (w_s = 20m^3/h, f_{ct} = 50%)

Table 6 Calculation results for the discharge operational method

Operational Method			Day when discharge is ended	Heat Amount [GJ]				Energy Consumption [GJ]						System COP	η_t
ws	fct	θwco		Strage	Discharge	Ratio	Artificial HP	Strage: Pump	Strage: CT Fan	Strage: CT Pump	Discharge	Artificial HP	Sum.		
Only Artificial HP			-	0	0	-	224.0	0	0	0	0	74.66	74.66	-	1.000
30m³/h	50%	23°C	10/31	326.6	150.3	46.0%	73.7	5.8	13.4	10.0	11.7	24.56	65.43	3.68	0.876
		24°C	10/31		184.1	56.4%	39.9				13.4	13.31	55.85	4.33	0.748
		25°C	9/29		199.9	61.2%	24.1				13.9	8.03	51.05	4.65	0.684
		26°C	9/4		197.4	60.4%	26.6				12.5	8.87	50.49	4.74	0.676
20m³/h		23°C	10/31	322.4	147.3	45.7%	76.6	2.2	13.9	10.4	11.5	25.55	63.52	3.88	0.851
		24°C	10/31		181.0	56.1%	43.0				13.3	14.32	54.05	4.56	0.724
		25°C	9/21		193.2	59.9%	30.8				13.3	10.27	50.05	4.86	0.670
		26°C	9/5		196.5	61.0%	27.4				12.5	9.15	48.13	5.04	0.645

Figure 21 to 23 show the calculation results when w_s and f_{ct} are set at 20m^3/h and 50% respectively. The results indicate that if $\theta_{cc,wo}$ is changed from 23°C to 24, 25, 26°C, the discharged heat amount is increased 22.9%, 31.2% and 33.4% respectively. The value of η_t is minimum when $\theta_{cc,wo}$ is set at 26°C.

Table 6 shows the calculation results of all cases. To compared with these eight cases, η_t is minimum when w_s = 20m^3/h, f_{ct} = 50% and $\theta_{cc,wo}$ = 26°C. The optimized operational method improves the system COP to 5.04.

CONCLUSION

The present paper proposes the method to optimize the operational method of an actual HVAC system with underground seasonal thermal storage using simulation. The simulation consists of a physical model of an underground thermal storage system through which heat is exchanged between circulated water in the pipes embedded in the building foundation piles and the ground, and the component models of the HVAC system.

The optimal operational method is obtained by conducting case studies using simulation. This paper analyzes the water flow rate and the inverter value of the cooling tower fan for the storage operation and the coil outlet water temperature for the discharge operation. The results shows the optimal operational method is that w_s = 20m^3/h, f_{ct} = 50% and $\theta_{cc,wo}$ = 26°C. The method can save the energy consumption of the system by approximately 30% and improve system COP from 3.06 to 5.04.

NOMENCLEATURE

A : Area [m^2], c : Specific heat [J/kg·K], C : Capacity rate [kW/K], C_h : Dimensionless pressure head [-], C_f : Dimensionless flow rate [-], d : Distance between piles [m], E : Energy consumption [J], f : Inverter value [-], g : Design air mass flow rate [kg/s], h : Enthalpy [J/g], K : Unit conductance [W/m^2·K], m : Air mass flow rate [kg/s], N : Number [-], r : Radius [m], Q : Heat amount [J], U : Number of Transfer Units [-], V : Volume [m^3], w : Water flow rate [kg/s], x : Inlet water humidity [kg/kgDA], α : Thermal transfer coefficient [W/m^2·K], η : Efficiency [-], λ : Thermal conductivity [W/m·K], θ : Temperature [°C], ρ : Density [kg/m^3], ΔP : Pressure head [kPa]

Subscript:

a : Air, ahp : Artificial heat pump, ai : Inlet air, ao : Outlet air, as : Saturation air, cc : Cooling coil, cd : Cooling coil in dry regimes, cr : Typical coil model, ch : Heat exchanger in cooling tower, cw : Cooling coil in wet regimes, ct : Cooling Tower, ct,f: Fan of CT1, ct,p: Pump of CT1, ele : Element of the underground thermal model, ew: East-west direction, i : Inlet, inv : Inverter, mor : Motor, ns: North-south direction, o : Outlet, oa : Outdoor air, p : Pump, pd: PC2 and PC3, pi : Inside of pipe, po : Outside of pipe, ps: PC1, ra : Return air, s : Soil, sd: Discharge operation, ss: Storage operation, utm : Underground thermal model, w : Water, wi : Inlet water, wo : Outlet water

REFERENCES

ASHRAE: Fundamentals Handbook (2001), Chapter 3 Heat Transfer, 2001

ASHRAE: HVAC Systems and Equipment Handbook (2004), Chapter 35 Cooling Tower, 2004

ANSYS: http://www.ansys.com/

Cane, R. L. D., D. A. Forgas: Modeling of ground source heat pump performance, ASHRAE Transactions 97(1), pp.909–925, 1991.

Clark, R. D.: HVACSIM+ Building Systems and Equipment Simulation Program Reference Manual, NBSIR 84-2996, January 1985

Husaunndee A., Riederer P., et Visier J.C.: Coil modelling in the Simbad toolbox - numerical and experimental validation of the cooling coil model, SSB'98 System Simulation in Building Conference, Liege, December 14-16, 1998

M. Miyata, H. Yoshida, M. Asada, K. Fujii, S. Hashiguchi: Estimation of Excessive HVAC Energy Consumption due to faulty VAV units, Building Simulation, 9th International Conference, pp.777-786, Canada, 2005

M. Miyata, H. Yoshida, M. Asada, T. Iwata, Y. Tanabe, T. Yanagisawa: Estimation of Energy Baseline by Simulation for On-going Commissioning and Energy Saving Retrofit, International Conference of Enhanced Building Operation, Shenzhen, Vol.6-6-2, 2006

Rottmayer, S. P., W. A. Beckman, and J. W. Mitchell: Simulation of a single vertical U-tube ground heat exchanger in an infinite medium. ASHRAE Transactions 103(2), pp.651–658, 1997

Wang, F., H. Yoshida, M. Miyata: Total Energy Consumption Model of Fan Subsystem Suitable for Continuous Commissioning, ASHRAE Transactions, Volume 110, Part-1, pp.357-364, 2004

Yavuzturk, C., J.D. Spitler, S.J. Rees: A Transient Two-dimensional Finite Volume Model for the Simulation of Vertical U-tube Ground Heat Exchangers, ASHRAE Transactions 105(2), pp.465-474, 1999