Development of a Simulation Tool for the Commissioning of a HVAC system with Seasonal Thermal Storage

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Synopsis

The authors developed a simulation tool to test and optimize the performance of an HVAC system with seasonal thermal storage. The tool consists of physical models of a ground-coupled thermal storage system through which heat is exchanged between circulated water in tubes embedded in building foundation piles and the ground, and components of an HVAC system, such as cooling towers, cooling coils, pumps, etc. The results of applying the tool to a real building system are reported as a commissioning process to test system performance and improve operation strategies. Compared with the performance of the system in the first year, the efficiency of storage operation is improved from 7.9 to 12.2, and the efficiency of discharge operation is improved from 7.1 to 13.7. If the operational methods proposed by the tool are adapted, the system COP can be improved from 3.06 to 5.16.

About the Authors

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Introduction

This paper proposes a tool for testing and optimizing, through simulation, the performance of an HVAC system with seasonal thermal storage. In this system, water cooled by cooling towers is circulated through tubes 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. However, problems are encountered in trying various operational methods in an actual system, because the minimum cycle of the system operation is one year and testing requires too much time. The tool developed in this paper helps us to determine how the system should be operated. This paper discusses development of the tool and its application to the actual system.

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 is a photograph of the building, and Figure 2 shows an outline of the air-conditioning system. The building has no artificial heat source and is air-conditioned by means of cooling water from underground and heating and cooling water supplied by a district heating and cooling (DHC) plant. Each air-handling 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 from the DHC plant and water thermal storage systems placed in the

Fig. 1 Photograph of the Analyzed Building

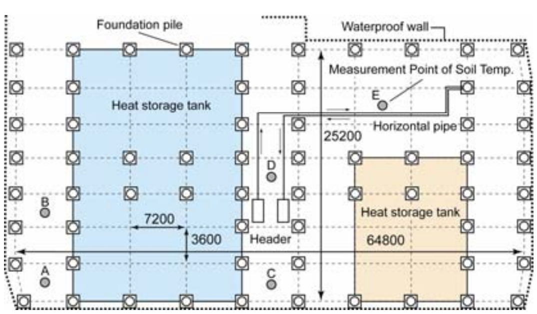

Fig. 3 Layout Plan of Foundation Piles

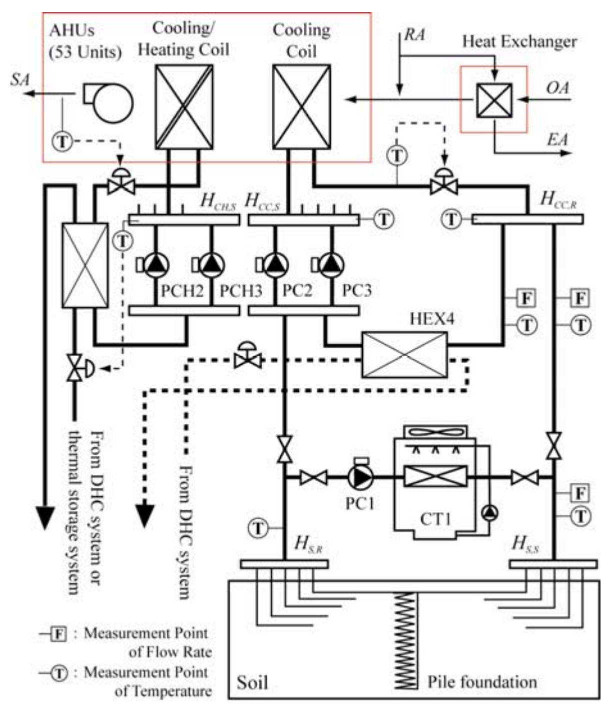

Fig. 2 Air-conditioning System.

Figure 4
Construction of a Pile

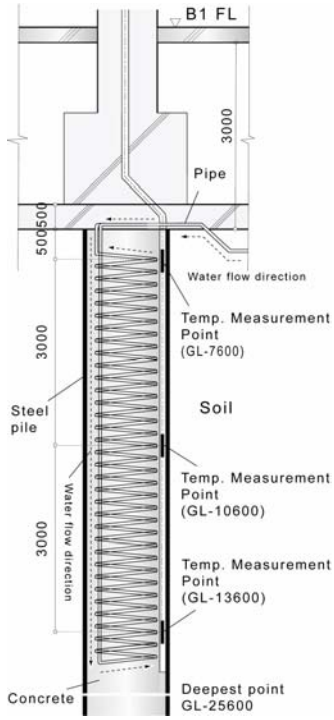

Figure 5 Cross-section of a Pile

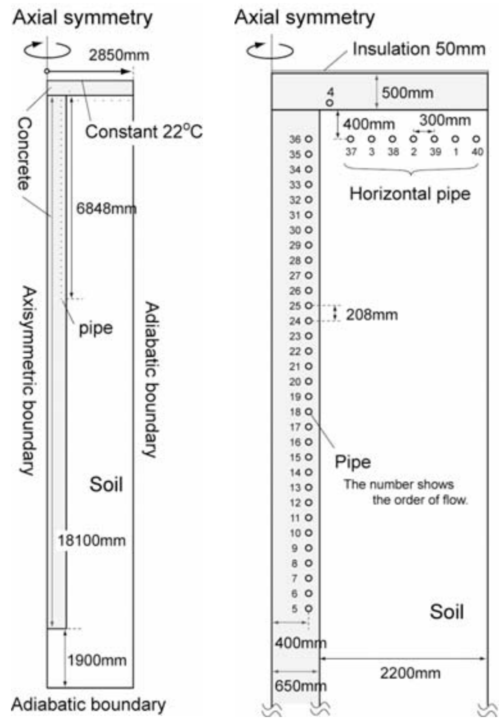

Figure 6 Underground Thermal Model
(Left: Entirety, Right: Enlargement)

building. Figure 3 shows the layout plan of the foundation piles, and Figure 4 is a photograph when the piles were constructed. Figure 5 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 400 mm.

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 (23^{o}C). The flow rate of the water flowed to HEX4 from the DHC plant is controlled so that the outlet water temperature of Header $H_{cc,s}$ attains the temperature set point (19^{o}C). If the outlet water temperature of $H_{cc,s}$ is less than the set point, the water from the DHC plant doesn't flow to HEX4.

Development of the Tool

The tool consists of a physical model of the underground thermal storage through which heat is exchanged between circulated water in the tubes 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.

An Underground Thermal Model

An underground thermal model is developed to simulate heat exchange between the pipes and the ground by means of the finite element method (FEM).

Development of the Model

Figure 6 shows the shape of the model and the assumed boundary conditions. This model is an axially symmetrical model and can estimate the heat transfer of a pile. The radius of the model r_d is determined as follows.

$$r_d = \sqrt{\frac{d_{ns} d_{ew}}{\pi}} \tag{1}$$

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 6, and the circulated water flows from Pipe 1 to Pipe 40 in order. The upper boundary condition of the model is set constant (22oC), because the measured temperature is constant.
The inputs of the model are inlet water temperature, which is equal to the temperature of water in Pipe 1, and water flow rate of the tubes, and the outputs are the outlet water temperature of the tubes and the heat transferred between the pipes and the ground. These data are measured by BEMS every 5 minutes.

The thickness of the pipe is disregarded since it is very small and the equivalent heat transfer coefficient α_{pe}, which is a summation of the heat thermal conductivity of the thickness of pipe and the heat transfer coefficient between the pipe and inside fluid, is used as the heat transfer coefficient of pipe. The coefficient α_{pe} is calculated as follows.

$$\frac{1}{r_{po}\alpha_{pe}} = \frac{1}{r_{pi}\alpha_{pi}} + \frac{\log(r_{po}/r_{pi})}{\lambda_p} \tag{2}$$

From thermo physical properties, r_{po}, r_{pi}, α_{pi}, and λ_p are 0.0135 m, 0.01025 m, 2545 W/m^2, and 0.337 W/m·K respectively, and α_{pe} is determined as 86.57 W/m^2 ·K.

Because the independent, annular-shape pipes are arranged in the model, the temperature differences between neighboring pipes are calculated by considering the heat balance over the

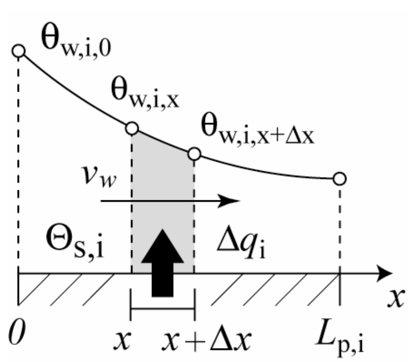

Figure 7 Heat Balance of Pipe *i*.

Table 1 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

infinitesimal distance Δx as shown in Figure 7.

$$\Delta q_i(t) = a_p\left(\Theta_{s,i}(t) - \theta_{w,i,x}(t)\right)\Delta x \tag{3}$$

$$\Delta q_i(t) = c_{pw} v_w(t) \Delta\theta_{w,i,x}(t) \tag{4}$$

$$a_p = 2\pi r_{po}\alpha_{pe} \tag{5}$$

$$\Delta\theta_{w,i,x}(t) = \theta_{w,i,x+\Delta x}(t) - \theta_{w,i,x}(t) \tag{6}$$

Using Equation (3) and (4), Equation (7) is obtained in consideration that $\theta_{w,i,0}(t)$ is equal to $\theta_{w,i-1,L_{p,i-1}}(t)$.

$$\theta_{w,i,L_{p,i}}(t) = \Theta_{s,i}(t) - \left(\Theta_{s,i}(t) - \theta_{w,i-1,L_{p,i-1}}(t)\right)\exp\left(-\frac{a_p L_{p,i}}{c_{p,w} v_w(t)}\right) \tag{7}$$

$$L_{p,i} = 2\pi r_{d,i} \tag{8}$$

Then, the averaged water temperature of Pipe i is defined as follows.

$$\overline{\theta_{w,i}}(t) = \Theta_{s,i}(t) - \frac{1}{L_{p,i}}\int_0^{L_{p,i}}\left\{\Theta_{s,i}(t) - \theta_{w,i,x}(t)\right\}dx \tag{9}$$

From the above equations, the water temperature of pipes can be calculated as follows. Here, the initial conditions of water temperature $\overline{\theta_{w,i}}(t)$ are given.

1) Set the water temperatures of the pipes as $\overline{\theta_{w,i}}(t)$ and calculate the surface temperatures of the pipes in next time step $\Theta_{s,i}(t+\Delta t)$ by means of FEM.
2) Calculate $\theta_{w,i,L_{p,i}}(t+\Delta t)$ by inputting $\Theta_{s,i}(t+\Delta t)$ into Equation (7). Here, $\theta_{w,0,L_{p,i}}(t+\Delta t)$ is water temperature of the header $H_{s,s}$.
3) Calculate $\overline{\theta_{w,i}}(t+\Delta t)$ from Equation (9).
4) Set the water temperatures of the pipes as $\overline{\theta_{w,i}}(t+\Delta t)$ and recalculate the surface temperature of the pipe in the next time step. Repeat from Step 1).

The model is divided into approximately one thousand elements using FEM software ANSYS [1]. The initial temperature of the each nodal point is set at the average soil temperature measured at Points A to E in Figure 3. The temperature of nodal points whose depths range from 0 m to 2 m is set to 19.18^{o}C, the temperature of points whose depths range from 2 m to 5 m is set to 19.60^{o}C, and the temperature of points whose depths are greater than 5 m are set to 19.16^{o}C. Table 1 shows the thermo physical properties of the materials in the model, which are determined from the result of boring exploration.

Verification of the Model

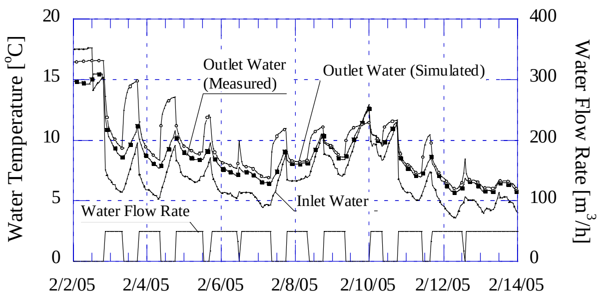

Figure 8 Water Temp. (1st year, Storage).

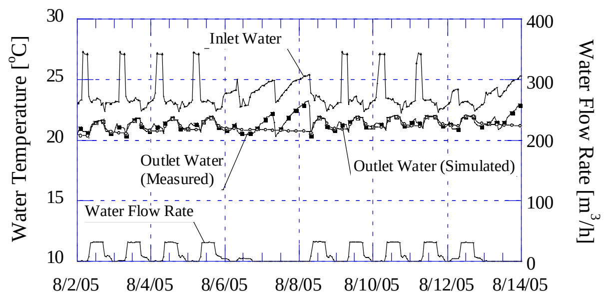

Figure 9 Water Temp. (1st year, Discharge).

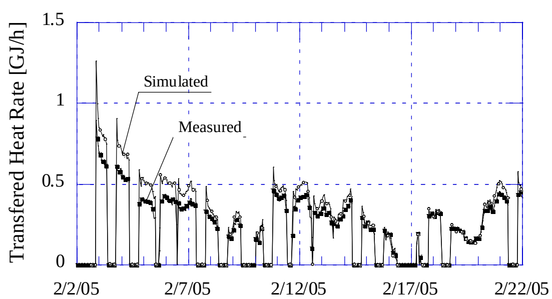

Figure 10 Transferred Heat (1st year, Storage).

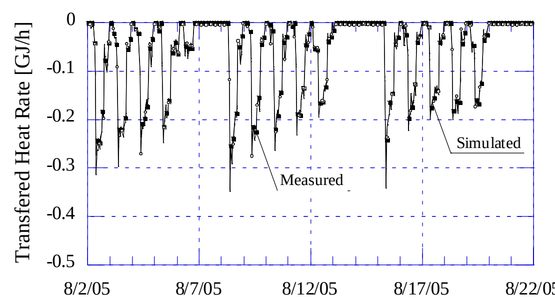

Figure 11 Transferred Heat (1st year, Discharge).

In order to verify the accuracy of the model, the outlet water temperature from underground is calculated by inputting the measured inlet water temperature to soil $\theta_{ws,i}$ and the water flow rate v_{ws} 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.

Figures 8 to 11 show a comparison between simulated and measured data of $\theta_{ws,i}$, $\theta_{ws,o}$, v_{ws} and the storage and discharge heat amount per pile q_{soil}. Tables 2 and 3 show the difference between measured and simulated $\theta_{ws,o}$ and the integrated heat amount of all pipes Q_{soil}. The difference of $\theta_{ws,o}$ ranges from –0.13 K to 0.18 K, and RMSE ranges from 0.26 K to 0.36 K. The results show that the model can simulate the outlet water temperature accurately. The difference of Q_{soil} is slightly large, especially during the storage period. As can be seen from Figure 10, the difference is large in the beginning of the storage period. The reason may be the initial soil temperature or boundary condition of the model, etc. Clarification of the reason is a future task.

Air-conditioning System Model

Figure 12 shows the outline of the air-conditioning system model developed in the present study.

Table 2 Comparison of Outlet Water Temp.

	Period	Error	RMSE
1st Year	Storage From 2/1/05 to 4/30/05	0.14 K	0.33 K
1st Year	Discharge From 7/1/05 to 10/30/05	0.02 K	0.26 K
2nd Year	Storage From 12/1/05 to 4/30/06	0.18 K	0.36 K
2nd Year	Discharge From 7/1/06 to 10/30/06	-0.13 K	0.35 K

Table 3 Comparison of Transferred Heat.

	Period	Measured	Simulated	Difference
1st Year	Storage From 2/1/05 to 4/30/05	213.4 GJ	221.85 GJ	3.96%
1st Year	Discharge From 7/1/05 to 10/30/05	145.1 GJ	141.2 GJ	-2.69%
2nd Year	Storage From 12/1/05 to 4/30/06	373.6 GJ	398.2 GJ	6.58%
2nd Year	Discharge From 7/1/06 to 10/30/06	120.0 GJ	125.8 GJ	4.83%

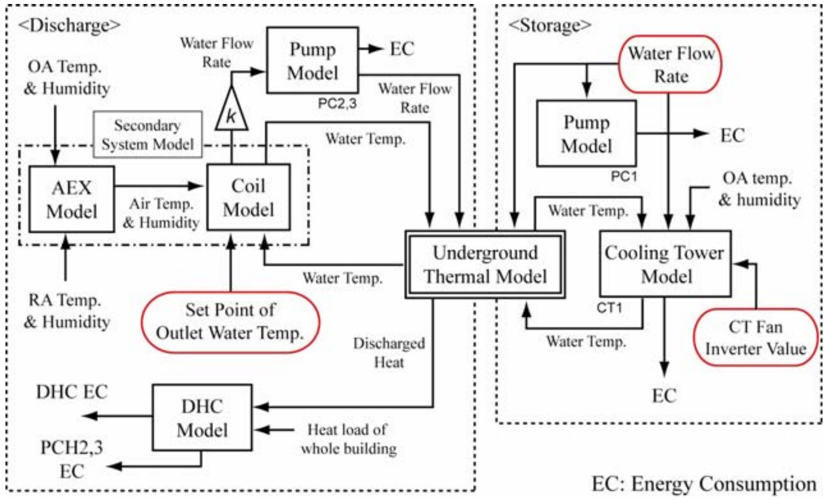

Figure 12 Outline of the Air-conditioning System Model.

Table 4 Specifications of the Components in the HVAC System.

Equipment		Specification
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 m^3/min, Rated energy consumption of fan 6.0 kW, Rated energy consumption of pump 1.6 kW
Pump	PC 1	Rated energy consumption 15 kW, Rotation speed 29.2 rps, Pressure 30 m, Water flow rate 25 kg/s
	PC 2,3	Energy consumption 5.5 kW, Rotation speed 29.0 rps, Pressure 30 m, Water flow rate 6.7 kg/s
Cooling Coil		Rated heat transfer rate 4.2 kW, Water flow rate 0.2 kg/s, Number of rows of tubes 22, Number of tubles per row 4, Length of finned section 480 mm, Height of finned section 1100 mm, Width of finned section 180 mm, Rated return air flow rate 3940 m^3/h, Rated outdoor air flow rate 660 m^3/h
Heat Exchanger		Rated energy consumption 2.8 kW, Air flow rate 3500 m^3/h, Rated exchange efficiency of temperature 77 %, Rated exchange efficiency of enthalpy 60.5 %

The model consists of the underground thermal model, the cooling tower model, the pump models, the heat exchanger model, and the cooling coil model [2][3][4]. Table 4 shows the specifications of the components for which models are developed. Except for the underground thermal model, the models are calibrated by the operational data measured in the actual system. 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 the coefficient k_{cc}. The coefficient k_{cc} is determined from measured operational data, and its value is set to 47.43. The inputs of the air-conditioning system model are the temperature and humidity of the outdoor air and the return air and 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 header $H_{cc,s}$ and the cooling coil for discharge operation.

Application of the Tool to the Actual Building

Table 5 shows changes in the actual operational conditions and performance results for three years starting from the initial operation of February 2005. The tool was applied three times: the first application was for the operation method in the storage period of the 2nd year, the second application was for the method in discharge period of the 2nd year, and the latest application was for the method in storage period of the 3rd year. During the period of storage operation, the water flow rate to the soil $V_{w,s}$ and the inverter value of cooling tower fan I_{ct} are parameters of the operation. The inverter of cooling tower was installed in December 2006 in order to reduce energy consumption for the storage operation. During the period of the discharge operation, the temperature set point of the supply header $\theta_{hcc,s}$ and the set point of the coil outlet water temperature θ_{wco} are parameters.
The efficiencies of the storage operation η_s and the discharge operation η_d and the system COP η_{cop} are defined as follows.

$$\eta_s = \frac{Q_s}{E_s} \tag{10}$$

$$\eta_d = \frac{Q_d}{E_d} \tag{11}$$

$$\eta_{cop} = \frac{Q_d}{E_s + E_d} \tag{12}$$

Actual Performance of 1st Year Operation

During the storage period of the 1st year, $V_{w,s}$ was set to 50 m^3/h. Although there was no inverter of cooling tower, I_{ct} was 100%. The storage period was about three months, and heat amount charged during the period was 213.4 GJ. The charged heat amount estimated in the design period is 256 GJ. Because the designed value was calculated on the assumption that the

Table 5 Changes in Operational Conditions and Actual Performance.

Period			Operational Conditions				Actual Performance				
			Storage		Discharge		Heat Amount [GJ]		Energy [GJ]	Effi. [-]	System COP
			$V_{w,s}$	I_{ct}	$\theta_{hcc,s}$	θ_{wco}	Storage	Dischage			
Design (OPR)							256	154			
1st year	2/1/05 to 4/30/05	Storage	50	100	-	-	213.4	-	27.0	7.9	-
	7/1/05 to 10/30/05	Discharge	-	-	19	23	-	145.1	20.4	7.1	3.06
	Apply the tool to the system (1)										
2nd year	12/1/05 to 4/30/06	Storage	45	100	-	-	373.6	-	55.5	6.7	-
	Apply the tool to the system (2)										
	7/1/06 to 4/30/06	Discharge	-	-	Not Use	25	-	201.4	13.0	15.5	2.94
	Apply the tool to the system (3)										
3rd year	12/1/06 to 4/30/07	Storage	30	50	-	-	322	-	26.5	12.2	
	6/1/07 to 10/30/07	Discharge	-	-	Not Use	25	-	218.2	15.9	13.7	5.16

Estimated Value by Simulation

Table 6 Analysis of $V_{w,s}$

Vw,s	$50m^3/h$	$30m^3/h$
Heat Amount	411GJ	415 GJ
Energy Consumption	47.4 GJ	36.0 GJ
Efficiency	8.67	11.52

Table 7 Analysis of Discharge Operation.

$\theta hcc,s$	19°C	20°C	Not use the water from DHC			
θwco	23°C	24°C	23°C	24°C	25°C	26°C
Heat Amount	146 GJ	151 GJ	165 GJ	199 GJ	234 GJ	263 GJ
Energy Consumption	9.6GJ	8.6 GJ	12.3 GJ	14.1 GJ	16.7 GJ	17.9 GJ
Efficiency	15.21	17.56	13.41	14.11	14.01	14.69

storage operation began from December, the actual heat amount was less than the amount estimated in the design period. During the discharged period, $\theta_{hcc,s}$ and θ_{wco} were set at 19°C and 23°C, respectively. The actual discharged heat amount was 145.1 GJ, as compared with the assumed discharge heat amount was 154 GJ. Because the storage heat amount was less than the assumed amount, the actual discharge heat amount was also less than the amount estimated in the design period.

Application of the Tool for Storage Operation of 2nd Year

The problem of the operation was the low system COP, whose value was 3.06 in the 1st year. In order to reduce energy consumption for the storage operation by decreasing the water flow rate $V_{w,s}$, the relation between water flow rate and the storage heat amount was analyzed by means of the tool.

The heat amounts and the energy consumptions when $V_{w,s}$ was set at 50 m^3/h and 30 m^3/h were simulated by means of the tool. Table 6 shows the calculation results. The results show that if $V_{w,s}$ is decreased from 50 to 30 m^3/h, the energy consumption of the system is decreased approximately 24%, because the energy consumption of pump is decreased significantly, by 68% (17.0 GJ to 5.4 GJ), although the storage heat amounts are not changed.

Actually, our application of the tool was late, because we needed much time for the development of the tool, and the water flow rate in the storage period of the 2nd year was set to 45 m^3/h. The charged heat amount was 361 GJ, energy consumption was 53 GJ, and the efficiency of storage operation was 6.7, which was worse than the value of the 1st year.

Application of the Tool for Discharge Operation of 2nd Year

In order to increase the discharged heat amount, the set points of $\theta_{hcc,s}$ and θ_{wco} were analyzed by means of the tool. Table 7 shows the analyzed results. The results show that the heat amount is increased if the water from the DHC plant is not used, and when θ_{wco} is set at 26°C the amount is increased approximately 80% as compared with the amount of the operational method at that time. Because the simulation results indicated that the averaged soil temperature was raised if θ_{wco} was set at 26°C, we decided that θ_{wco} should be set at 25°C.

The actual performance result shows that the discharged heat amount was 201.4 GJ and the energy consumption was 13.0 GJ. The efficiency of the discharged operation improved from 7.1 to 15.5.

Table 8 Analysis of $V_{w,s}$ and I_{ct} for the Storage Period of 3rd Year.

Vw,s	Ict	Heat Amount		Energy Consumption [GJ]				Efficiency of Storage	System COP
		Storage [GJ]	Discharge [GJ]	Storage			Discharge		
				Pump	CT Fan	CT Pump			
$45m^3/h$	100%	361.5	234.9	12.08	32.94	7.98	16.68	6.82	3.37
	75%	348.7	229.3	13.23	21.16	8.75	16.37	8.08	3.85
	50%	326.0	221.1	14.95	13.30	9.88	16.05	8.55	4.08
	25%	271.6	198.1	17.76	9.06	11.74	14.87	7.04	3.71
$30m^3/h$	100%	362.1	234.2	4.77	33.77	8.18	16.54	7.75	3.70
	75%	349.3	228.8	5.17	21.45	8.87	16.35	9.84	4.41
	50%	326.6	219.6	5.81	13.39	9.95	15.86	11.20	4.88
	25%	273.4	198.1	6.85	9.06	11.74	14.81	9.89	4.66
$20m^3/h$	100%	356.7	231.7	1.82	35.88	8.69	16.56	7.69	3.68
	75%	344.3	225.5	1.96	22.59	9.34	16.15	10.16	4.51
	50%	322.4	218.2	2.17	13.94	10.36	15.85	12.18	5.16
	25%	270.7	196.6	2.49	9.15	11.86	14.83	11.52	5.13
$15m^3/h$	50%	315.5	213.4	1.05	14.42	10.72	15.71	12.05	5.09
$10m^3/h$	50%	299.0	205.5	0.36	15.23	11.31	15.27	11.11	4.87

Application of the Tool for Storage Operation of 3rd Year

In order to improve the efficiency of storage operation, analysis of $V_{w,s}$ and I_{ct} was conducted. Table 8 shows the analysis results. When $V_{w,s}$ is set at 20 m^3/h and I_{ct} is set at 50%, the efficiency and the system COP become the highest among the test cases analyzed in this paper; efficiency improves from 6.82 to 12.18 and system COP improves from 3.37 to 5.16 as compared with the value of the operational method of the 2nd year.

Summary

The present paper develops a tool to test and optimize, by means of simulation, the performance of an HVAC system with underground seasonal thermal storage. The tool consists of physical models of an underground thermal storage system through which heat is exchanged between circulated water in pipes embedded in the building foundation piles and the ground, and the components of an HVAC system.

The tool was applied to an actual building, and the operational methods were changed in view of the results of the application of the tool. Compared with the performance of the system in the 1st year, the efficiency of storage operation is improved from 7.9 to 12.2, and the efficiency of discharge operation is improved from 7.1 to 13.7. If the operational methods proposed by the tool are adapted, the system COP can be improved from 3.06 to 5.16.

Nomenclature

d_{ns} : Distance between piles of north and south direction [m], d_{ew} : Distance between piles of east and west direction [m], r_{po} : Outside diameter of pipe [m], r_{pi} : Inside diameter of pipe [m], α_{pi} : Thermal transfer coefficient of pipe [$W/m^2 \cdot K$], λ_p : Thermal conductivity of pipe [$W/m \cdot K$], $\Theta_{s,i}$: Surface Temperature of Pipe *i* [^oC], $\theta_{w,i,x}$: Water temperature in Pipe *i* [^oC], c_{pw} : Specific heat of water [$J/g \cdot K$], v_w : Water flow rate in pipe [kg/s], v_{ws} : Water flow rate to underground [kg/s], $\theta_{ws,i}$: Inlet water temperature to underground [^oC], $\theta_{ws,o}$:

Outlet water temperature from underground [^{o}C], Q_{soil} : Heat amount of all pipes [J], $V_{w,s}$: Water flow rate for storage operation [kg/s], I_{ct} : Inverter value of cooling tower fan [-], $\theta_{hcc,s}$: Temperature of supply header $H_{cc,s}$ [^{o}C], θ_{wco} : Coil outlet water temperature [^{o}C], E_s : Energy consumption for storage operation [J], Q_s : Charged heat amount [J] E_d : Energy consumption for discharge operation [J], Q_d : Discharged heat amount [J]

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