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STUDY ON OPTIMIZING THE OPERATOIN OF HEAT SOURCE EQUIPMENTS IN AN ACTUAL HEATING/COOLING PLANT USING SIMULATION

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ABSTRACT

In order to determine the optimal combination of the heating source equipments in an existing office building, simulations of six different combination cases were conducted using the newly developed mathematical models of each component. From the simulation results, the optimal combination case can reduce the energy consumption by 19.7%, running cost by 12.8% and carbon-dioxide emissions by 29.6%, compared to the present operational combination.

KEYWORDS

Simulation, Commissioning, Energy Consumption, Running Cost, Carbon-dioxide Emissions

INTRODUCTION

In recent years, environmental issues, including global warming, energy conservation and reducing Carbon-dioxide (CO_2) emissions, are increasingly causing more attentions of all over the world. These issues are important in the field of building equipments industry as well. Buildings occupied by different types of tenant, whose work schedules might differ from each other, are equipped with multiple heat source equipments having different performance to satisfy different requirements, from large heating/cooling loads to small ones.

To satisfy heating/cooling requirements, achieve energy conservation and ensure cost efficiency, it is important to study the combination and operation priority order of the heat source equipments and find out the optimal operation method.

Therefore this research focuses on studying the central heating/cooling plant of an office building located in Osaka Japan to find an optimal operational combination of the heat source equipments. In detail, the following studies are conducted. 1) Develops mathematical models of each equipment in the heat source system using the specification data and refining the model using the data measured by the Building Energy Management System (BEMS). 2) Connects all component models to construct the whole system model of the plant. 3) Uses the system model to simulate the energy consumption, running cost and carbon-dioxide emissions of several different combinations of heat source equipments to find an optimal operational combination.

PROFILE OF THE PLANT

The plant has been in use since December 2004. It consists of two gas-fired absorption chiller/heaters, two air source heat pumps, one centrifugal chiller and one

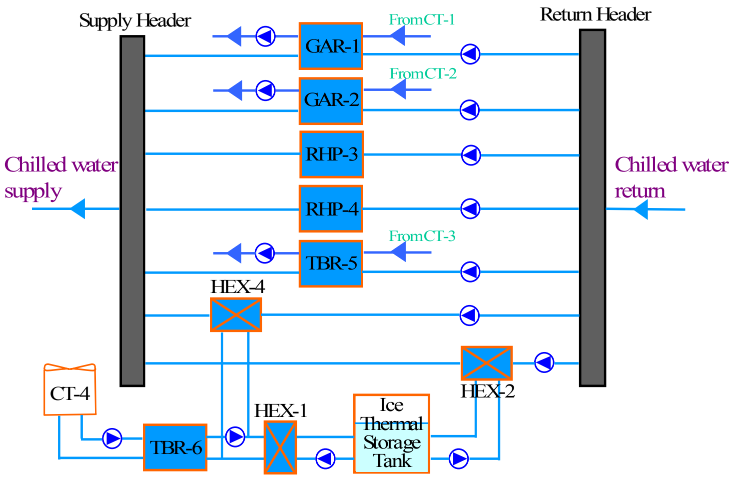

Figure 1 System diagram of the heat source system

Table 1 Heat sources list

Heat source	Name	Capacity[kW]	Number
Gas-fired absorption chiller/heater	GAR-1,2	1759	2
Air source heat pump	RHP-3,4	339	2
Centrifugal chiller	TBR-5	1406	1
Ice-making centrifugal chiller(ice making mode)	TBR-6	703.2	1
Ice-making centrifugal chiller(chilled water		879	

Table 2 Present operation priority

	Operation priority order	
	Daytime	Nighttime
1	thermal discharge	TBR-5
2	GAR-1	RHP-3
3	GAR-2	RHP-4
4	TBR-5	GAR-1
5	RHP-3	GAR-2
6	RHP-4	
7	TBR-6(chilled water mode)	※TBR-6(ice making mode)

centrifugal ice chiller, as shown in Table 1. Figure 1 shows the system diagram of the plant. The features of this plant are that it has multiple heat source equipments, ice thermal storage system, tank, and is required to run 24 hours a day.

The heat source equipments are composed of the most advanced equipments at the time of the completion and adopt the variable control of cooling water flow rate, and on-off control of cooling tower. Ice thermal storage tank adopts dynamic ice system, and it enables efficient operation of the heat source equipments.

During the present operation, equipments are started or stopped according to the operators' empirical judgment based on an operation priority order and hourly heating/cooling load. The present operation priority orders are divided into six types, which are summertime, spring and autumn period, wintertime, daytime, and nighttime. These equipment combinations and priority orders are decided according to heat source equipments' capacities, thermal storage or thermal discharge, magnitude of loads.

The BEMS can memory up to 1600 points of measurements. Using this function, the total electric consumption, water temperature, flow rate, and pressure, etc. are measured once an hour.

CLASSIFICATION OF OPERATION MODE

Operation mode is defined as the combination of running heat source equipments. The purpose of classifying operation mode is to enable the automated determination of the on/off states of equipments and to find the optimal operation priority order by comparing the energy consumptions of running at different priority orders.

Table 2 shows the present operation priority order of summertime. Daytime operation is from 7:00 to 22:00 Nighttime operation is from 22:00 to 7:00. If load exceeds the sum of the capacities of running equipments, one more heat source machine will start. If load decreased to 90% of the capacity of the heat source machine of second top priority level among presently running machines, the machine of top priority level will be stopped.

MODELING OF HEAT SOURCE EQUIPMENTS

The heat source equipment modeling and validation process is as follows. 1) Develop a model using the performance curve obtained from manufacturers. 2) Compare the model-simulated data with the specification data. 3) Use the measured data to refine model by a compensation coefficient, which is the ratio of measure data to simulated data.

Modeling of chiller

This plant has six chillers. The chiller model calculate the energy consumption E using five dimensionless variables, which are load Q_e, outlet temperature of chilled water T_{eo}, inlet temperature of cooling water T_{ci}, flow rate of chilled water M_e, and flow rate of cooling water M_c. Equation 1 shows the chiller model, and Equation 2 to 7 show the definition of each dimensionless variable.

$$r_E = (a_1 r_Q{}^2 + a_2 r_Q + 1)(a_3 r_{T_{eo}}{}^2 + a_4 r_{T_{eo}} + 1)(a_5 r_{T_{ci}}{}^2 + a_6 r_{T_{ci}} + 1)(a_7 r_{M_e}{}^2 + a_8 r_{M_e} + 1)(a_9 r_{M_c}{}^2 + a_{10} r_{M_c} + 1) \quad (1)$$

$$r_E = \frac{E}{E_{rate}} \quad (2)$$

$$r_Q = \frac{Q_e - Q_{e,rate}}{Q_{e,rate}} \quad (3)$$

$$r_{T_{eo}} = \frac{T_{eo} - T_{eo,rate}}{T_{ci,rate} - T_{ei,rate}} \quad (4)$$

$$r_{T_{ci}} = \frac{T_{ci} - T_{ci,rate}}{T_{ci,rate} - T_{ei,rate}} \quad (5)$$

$$r_{M_e} = \frac{M_e - M_{e,rate}}{M_{e,rate}} \quad (6)$$

$$r_{M_c} = \frac{M_c - M_{c,rate}}{M_{c,rate}} \quad (7)$$

Where,

E : Energy consumption, [kW]

Q_e : Load, [kW]

T_{ei} : Inlet temperature of chilled water, [^{o}C]

T_{eo} : Outlet temperature of chilled water, [^{o}C]

T_{ci} : Inlet temperature of cooling water, [^{o}C]

M_e : Flow rate of chilled water, [kg/s]

M_c : Flow rate of cooling water, [kg/s]

$a_1, a_2, a_3, ...a_{10}$: Fitted coefficients

The subscript of *rate* means the rated value of each variable.

Table 3 shows fitted coefficients using specification data. There are five variables in the chiller model, but many chillers' performance curves related to all the five variables are not available. For the unavailable variables, the rated values have to be used. When rated values are used, the term related to that variable will be equal to one so that in Equation 1 unavailable variables will disappear. The coefficients corresponded to the unavailable variables are marked with "-".

Then the models are validated and refined using measured data. The model-simulated data are compared to measured data to obtain compensation coefficients, which are the ratio of the sum four months energy consumptions of the measured data to that of simulated data. Table 4 shows the Root Mean Square Error divided by measured value (%RMSE) and compensation coefficient of each chiller. The accuracy of RHP-3, RHP-4 and TBR-6 (chilled water mode) is not so good because the running time of these equipments is usually less than one hour and the BEMS measurements for temperature and flow rate are obtained from instant pulse signals, while measurement for energy consumption is the integral value in one hour. Therefore, if running time of a equipments is less than one hour, large error will appear when comparing the measured integral energy consumptions to the consumptions simulated using the instant measured temperatures and flow rates. However, because the running time of these equipments is short, the impact on the total energy consumption is small. The simulated total energy consumptions are quite close to the measured data.

Modeling of cooling towers

The cooling water for GAR-1, GAR-2, TBR-5 and TBR-6 is cooled by CT-1, CT-2, CT-3 and CT-4, respectively. Table 5 shows the specification of the cooling towers. The model developed by Yoshida (1990) and the model explained in SHASE Handbook (1991) are used to simulate the performance of cooling towers.

CT-1, CT-2, CT-3 and CT-4 have 4, 4, 3 and 2 fans, respectively. The running fan number is controlled according to the outlet temperature of the cooling water. The outlet temperature set points are 26°C, 28°C and 30°C. Corresponding to these temperatures, the running fan number is decided automatically.

The running fan number of CT-4 is not controlled. The CT-4 starts and stops all the fans together according to the on/off of the chiller TBR-6.

Then the developed cooling tower models are validated using the method same as that of the chillers. The RMSE of the outlet temperature of CT-1, CT-2, CT-3 and CT-4 are 1.20°C, 1.28°C, 0.99°C and 1.91°C, which show an acceptable accuracy.

Modeling of pump of constant flow rate

There are 12 pumps of constant flow rate in this plant.

a) Modeling using specification data

The 4-degree formula, as shown in Equation 8 to 12, of dimensionless flow rate C_f and the dimensionless pressure head C_h used by HVACSIM+ (Clark 1985) is adopted to model the performance of pumps. The data of rotational speed N, flow rate m_w, pressure head dP and efficiency ε are read from performance curve obtained from the pump manufacturer. Then these data are used to fit the coefficients and $e_0, \ldots, e_4$ using the least mean square method. For pump performance simulation, the input is flow rate m_w, and the output is energy consumption E.

$$C_h = a_0 + a_1 C_f + a_2 C_f^{\,2} + a_3 C_f^{\,3} + a_4 C_f^{\,4} \tag{8}$$

$$\varepsilon = e_0 + e_1 C_f + e_2 C_f^{\,2} + e_3 C_f^{\,3} + e_4 C_f^{\,4} \tag{9}$$

$$C_f = \frac{m_w}{\rho N D^3} \tag{10}$$

$$C_h = \frac{1000 dP}{\rho N^2 D^2} \tag{11}$$

$$E = \frac{m_w dP}{\varepsilon \rho} \tag{12}$$

Where,

C_f : Dimensionless flow rate

Table 3 Fitted coefficients of chillers

	a_1	a_2	a_3	a_4	a_5	a_6	a_7	a_8	a_9	a_{10}
GAR-1,2	0.1026	1.0484	-	-	0.8201	0.4477	-	-	-0.5955	-0.5106
RHP-3,4	0.1296	0.8888	0.0742	0.2104	0.0719	0.3664	-	-	-	-
TBR-5	0.1154	0.9325	-	-	0.1958	0.3527	-	-	-	-
TBR-6(ice making mode)	0.0629	0.8614	0.1187	-0.3954	0.3543	0.5909	-	-	-	-
TBR-6(chilled water mode)	0.0595	0.8891	-	-	0.1918	0.3829	-	-	-	-

Table 4 %RMSE and compensation coefficients of chillers

	%RMSE[%]	Compensation Coefficient
GAR-1	5.6	1.04
GAR-2	9.2	1.08
RHP-3	43.3	1.25
RHP-4	42.4	1.47
TBR-5	13.7	1.16
TBR-6(ice making mode)	14.6	1.15
TBR-6(chilled water mode)	71.6	2.75

Table 5 Specification data of cooling towers

	Fan number	Cooling capacity[kW]	Inlet temperature [°C]	Outlet temperature [°C]	Water flow rate[kg/s]	Air flow rate[m3/s]	Power consumption [kW]
CT-1,2	4	3081	37.3	32	138.9	78.50	3.7×4
CT-3	3	1658	37	32	79.22	48.48	3.7×3
CT-4	2	1065	37	32	50.90	31.63	3.7×2

C_h : Dimensionless pressure head

ε : Efficiency

m_w : Flow rate, [kg/s]

ρ : Density, [kg/m^3]

N : Rotational speed, [rps]

D : Diameter of wheel, [m]

dP : Pressure head, [kPa]

E : Energy consumption, [kW]

$a_0,\ldots,a_4$, ﻭ$_4$,…,ﻭ$_0$: Coefficients

b) Compensation with the measured data

The developed models are compensated using the same method as mentioned in the section of chiller. The compensation coefficients of each pump are between 0.881 and 1.18.

Modeling of pump of variable flow rate

The pump of PCD-1 and PCD-2, which are the cooling water pumps for absorption chillers GAR-1 and GAR-2, are variable flow rate pumps, which are controlled by inverter according to the chiller's load information. In detail, the rotational speed and flow rate vary accompanying cooling loads. When cooling load decreases to 50% of the chiller's capacity, the pump reaches its minimum rotational speed. The inverter model is developed based on this logic and connected to the pump model. The validation results are shown in Figure 2. The RMSE of PCD-1 and PCD-2 is 7.69kW (13.98% of rated pump power) and 7.81kW (14.2% of rated pump power) respectively. The simulation error of monthly-integrated value is about 3%, which shows that the model is accurate enough for study the plant performance.

Modeling of ice thermal storage system

a) Modeling for thermal storage

During the thermal storage period, the water for making ice is cooled in ice-making heat exchanger by the brine from chiller TBR-6. The water is super-cooled to –2^{o}C and congeals into sherbet-shaped ice in thermal storage tank. The thermal amount stored in the ice tank is calculated using the flow rate and the temperature difference of the ice making water at the inlet and outlet of the heat exchanger, as shown in Equation 13

$$Q = M_w C_w (T_{in} - T_{out}) \tag{13}$$

Where,

Q : Heat amount, [kW]

M_w : Flow rate, [kg/s]

C_w : Specific heat, [kJ/kgK]

T_{in} : Inlet temperature, [^{o}C]

T_{out} : Outlet temperature, [^{o}C]

b) Model for thermal discharge

During the thermal discharge period, the chilled water is cooled in thermal discharge heat exchanger by the ice-melted water from the ice tank. The thermal discharge model calculates the transferred heat amount using the chilled water flow rate and temperature difference at the inlet and outlet of the heat exchanger. The equation is same as thermal storage model as shown in Equation 13.

c) Model validation

The developed model is validated using the measured thermal storage amount. The results are shown in Figure 3.

Sometime the simulation cannot match the measurement, but generally speaking the model can simulate the performance of the ice tank acceptably.

Modeling of heat exchanger

This plant has four heat exchangers used by the ice thermal storage system. Table 6 shows the specifications of the heat exchangers. For the purpose of simulating the water temperature after flowing through a heat exchanger, heat exchanger model is necessary. The physical model of heat exchanger is used for the simulation. The transferred heat amount of the counter-flow heat changer is calculated using the log-mean temperature difference T_{md} , as shown in

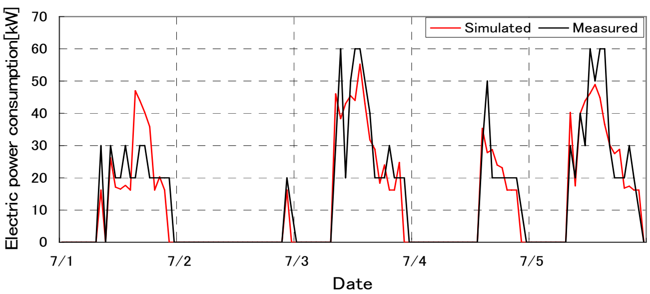

Figure 2 Power consumption of PCD-1

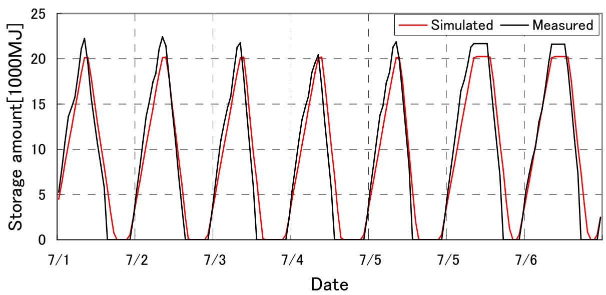

Figure 3 Thermal amount stored in ice tank

Equation 14 to 17.

$$Q = U_d A T_{md} \tag{14}$$

$$T_{md} = \frac{dT_2 - dT_1}{\ln(dT_2 / dT_1)} \tag{15}$$

$$dT_1 = T_{1,out} - T_{1,in} \tag{16}$$

$$dT_2 = T_{2,out} - T_{2,in} \tag{17}$$

And the heat change of the high and low temperature fluid is expressed in the following equation.

$$Q_1 = c_{p,1} m_1 (T_{1,in} - T_{1,out}) \tag{18}$$

$$Q_2 = c_{p,2} m_2 (T_{2,out} - T_{2,in}) \tag{19}$$

Where,

Q: Transferred heat amount, [kW]

U_d : Heat transfer coefficient, [W/m^{2}K]

A: Heat transfer area, [m^2]

T_{md} : Log-mean temperature difference, [K]

T : Temperature, [^{o}C]

c_p : Specific heat, [kJ/kgK]

Subscriptions:

1: high temperature side

2: low temperature side

in: inlet

out: outlet

If the heat loss is ignored, the heat amounts calculated by Equation 14, 18 and 19 are equal, as shown in Equation 20.

$$Q = Q_1 = Q_2 \tag{20}$$

Through solving the simultaneous equations of 14, 18, 19 and 20, the outlet fluid temperature of both high and low temperature sides can be obtained given the inlet fluid temperatures and flow rates. Therefore the inputs to the model are the inlet temperatures and the flow rates of the both sides fluid, and the outputs are the outlet temperatures of the fluids.

The developed model is validated using the data measured at HEX-2. The RMSE of the outlet temperature of the ice-melted water and chilled water is 0.27^{o}C, 0.38^{o}C, which are quite accurate simulation results.

Modeling of header

Because the outlet temperature set points of different chillers are different, the supply water temperature needs to be calculated by the mixing the outlet water from each chiller. Furthermore, because a bypass route between the return water header and supply water header are installed, a header model is necessary to calculate the mixed water temperature of return water and bypassed supply water, which is used as chiller inlet temperature to simulate chiller performance.

a) Supply header

The model of the supply header is a simple one, which calculates the supply temperature of chilled water by mixing the flows from each chiller, as shown in Equation 21 and 22.

$$M_1 = \sum_{i=1}^{7} m_i \tag{21}$$

$$T_S = \frac{1}{M_1} \sum_{i=1}^{7} t_i m_i \tag{22}$$

Where,

M_1 : Primary chilled water flow rate, [kg/s]

m_i : Chilled water flow rate of chiller *i*, [kg/s]

T_S : Supply temperature of chilled water, [^{o}C]

t_i : Outlet temperature of chiller *i*, [^{o}C]

b) Return header

The model of return header calculates the mixed water temperature of return water and bypassed supply water. The return and supply temperatures and the flow rates of return and bypassed water are the inputs to the models. While the bypassed water flow rate is calculated by subtracting the secondary water flow rate from the primary water flow rate. Equation 23 and 24 show the return header model.

$$T_{R1} = \frac{1}{M_1}\{M_2 T_{R2} + (M_1 - M_2) T_S\} \tag{23}$$

$$T_{R2} = \frac{L}{M_2 C_P} + T_S \tag{24}$$

Where,

M_2 : Secondary chilled water flow rate, [kg/s]

L : Cooling load, [kW]

Table 6 Specification of heat exchangers

	Heat transfer coefficient[W/m2℃]	Flow rate of low temperature side[kg/s]	Flow rate of high temperature side[kg/s]	Heat transfer area[m2]	Transfered heat amount[kW]
HEX-1	2466	64.91	67.18	418.9	703
HEX-2	4792	67.15	47.97	161.1	1406
HEX-3	2721	220.8	202.7	7.800	424
HEX-4	2276	63.53	30.00	168.3	879

T_{R1}: Return temperature of chilled water after bypass, [^{o}C]

T_{R2}: Return temperature of chilled water before bypass, [^{o}C]

C_p: Specific heat, [kJ/kgK]

MODELING AND VALIDATION OF THE WHOLE HEAT SOURCE SYSTEM

The models of the heat source equipments, expressed in the former sections, are connected to construct the whole heat source system model. In addition, the control model, expressed in the section of classifying operation mode, is connected to the whole system model to input the on/off states of heat source equipments.

Modeling and validation of the heat source system

The whole heat source system model consists of the following nine subsystem blocks, absorption chiller/heater GAR-1, absorption chiller/heater GAR-2, air source heat pump RHP-3, air source heat pump RHP-4, heat exchanger for the thermal discharge HEX-2, ice thermal storage system, heat exchanger for thermal storage HEX-4, and header. The whole system model is constructed in the environment of MATLAB® Simulink®, as shown in Figure 4.

The inputs to this model are cooling load, the outdoor air temperature and humidity and secondary chilled water flow rate, and remained thermal storage. The outputs are the total energy consumption, chilled water supply temperature, and thermal amount stored in ice tank.

The outlet temperature set point and chilled water flow rate of each chiller are used constant values, which are the average of the measured data at the range of rated value ± 20%. The flow rates of the pumps in the ice thermal storage are decided using the same method.

The heat source system model is validated by comparing the measured data of total energy consumption to the simulated data. The total energy consumption used for the validation is the sum of the primary energy calculated from simulated electric power and gas consumption. The simulation period is from June 1st to September 30th, 2006, and the time interval is 10 minutes. The result of the validation is shown in Figure 5. The RMSE and %RMSE of the time series data are 170kW and 12.7% and those of daily integral data are 868kW and 3.1% In addition, the average error of daily integral data is 0.14%. This whole system simulation is considerably accurate.

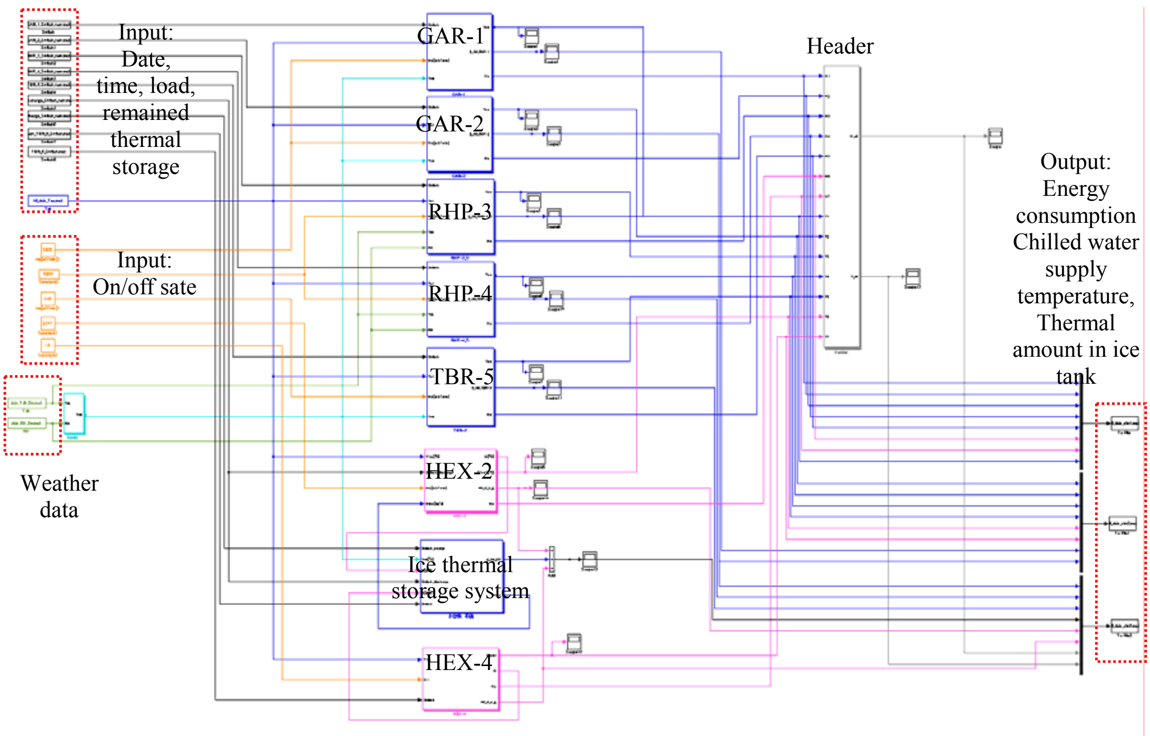

Figure 4 The whole heat source system model

Validation of the control model

At the former whole system validation, the on/off states of heat source equipments used the measure data. Here the on/off states of heat source equipments are decided automatically according to operational priority order and cooling load. The RMSE and %RMSE are 337kW and 25.2% for the time series data and 1439kW and 4.48% for the daily integral data. In addition, the average error of daily integral data is 1.9%.

In summary, the whole heat source system model with automated on/off control is accurate enough to study the performance of the plant.

STUDY OF THE OPTIMAL OPERATION OF THE HEAT SOURCE SYSTEM

The optimal operation of the plant is studies through simulating the plant performance at different operation priority orders using the former mentioned model. Table 7 shows the operation priority orders of the six cases for studying. Case 1 to 3 and Case 5 to 6 are the cases for studying the daytime and nighttime operation, respectively. Case 4 is used to study performance of not using the ice thermal storage system.

In addition to the energy consumption, the running cost and the carbon-dioxide emissions are also calculated to evaluate these operation orders. The running costs are calculated according to the price system of electric power company and gas company in Osaka Japan, as shown in Equation 25 and 26. The carbon-dioxide emissions are calculated based on the emissions per unit energy, as shown in Equation 27 and 28. Table 8 shows the meanings and values of all the variables in these equations.

$$C_e = P_{be}(1.85 - \cos\varphi)E_c + P_{ae}E_t - P_s E_s \quad (25)$$

$$C_g = C_b + P_{bg}G_r + P_{ag}G_t \quad (26)$$

$$CD_e = CD_d E_d + CD_n E_n \quad (27)$$

$$CD_g = CD_a G_t \quad (28)$$

The simulation results are shown in Table 9, which are the ratios of the four months integral simulated data to measured data of the primary energy consumption, running cost and carbon-dioxide emissions. The ratio less than one shows that the case is more effective than present operation. Therefore, the most effective operation priority order is that in the case 1 for the daytime operation, and case 5 for the nighttime operation. This optimal operation order can reduce the energy consumption by 19.7%, running cost by 12.8% and carbon-dioxide emissions by 29.6%, compared to

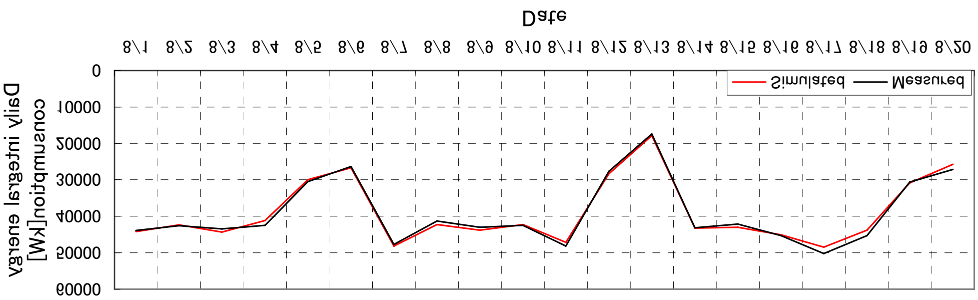

Figure 5 Daily integral primary energy of whole plant

Table 7 Priority orders for case study

		Standard order	Case(1)	Case(2)	Case(3)	Case(4)	Case(5)	Case(6)
Daytime	1	Thermal discharge	Thermal discharge	Thermal discharge	Thermal discharge	GAR-1	Standard order	
	2	GAR-1	TBR-5	RHP-3	GAR-1	GAR-2		
	3	GAR-2	GAR-1	RHP-4	GAR-2	TBR-5		
	4	TBR-5	GAR-2	GAR-1	RHP-3	RHP-3		
	5	RHP-3	RHP-3	GAR-2	RHP-4	RHP-4		
	6	RHP-4	RHP-4	TBR-5	TBR-5	TBR-6(chilled water mode)		
	7	TBR-6(chilled water mode)	TBR-6(chilled water mode)	TBR-6(chilled water mode)	TBR-6(chilled water mode)			
Nighttime	1	TBR-5	Standard order			TBR-5	RHP-3	GAR-1
	2	RHP-3				RHP-3	RHP-4	GAR-2
	3	RHP-4				RHP-4	TBR-5	RHP-3
	4	GAR-1				GAR-1	GAR-1	RHP-4
	5	GAR-2				GAR-2	GAR-2	TBR-5
		※TBR-6(ice making mode)					※TBR-6(ice making mode)	※TBR-6(ice making mode)

the present operational combination.

CONCLUSIONS

In this research, the optimal combination of the heat source equipments is studied as a part of commissioning work for the heat source system in an existing office building. The main conclusions are summarized in the followings.

1) An automated operation mode determination method is developed according to a preliminary decided operation priority order and time series cooling load.

2) Regression or physical models are developed using the performance curve or physical properties of heat source equipments and validated using the measured data.

3) The whole heat source system model is constructed by connecting each equipment model and the control model. The average energy consumption simulation error is 1.9% and the %RMSE is 4.48%. Therefore, the whole system model is sufficiently accurate for studying the optimal operation of the plant.

4) Six cases of different operation priority orders are studied using the developed model. The optimal operation priority order can reduce the primary energy consumption by 19.7%, running cost by 12.8% and carbon-dioxide emissions by 29.6%, compared to the present operation.

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Table 8 Description of the variables in Equation 25 to 28

Symbol	Meaning	Unit	Value
C_e	Electricity rate	yen	–
P_{be}	Basic piece rate	yen/kW	1648.5
$\cos\phi$	Power factor	–	1.00
E_c	Part of air-conditioning power, affecting contract demand	kW	–
P_{ae}	Specific piece rate	yen/kWh	10.02(summertime) 9.10(other seasons)
E_t	Power consumption	kWh	–
P_s	Thermal storage discount piece rate	yen/kWh	4.9
E_s	Power consumption for thermal storage	kWh	–
C_g	Gas rate	yen	–
C_b	Flat basic rate	yen	75600

symbol	Meaning	yen	Value
P_{bg}	Basic rate for gas flow	yen/(m3/h)	945(summertime) 2258(wintertime)
G_r	Rated gas consumption of heat source equipments	m3/h	216
P_{ag}	Piece rate	yen/m3	41.92(summertime) 46.03(wintertime)
G_t	Gas consumption	m3	–
CD_e	CO_2 emissions for electric power	kg	–
CD_d	Unit emission for daytime power	kg/kWh	0.293
E_d	Daytime power consumtion	kWh	–
CD_n	Unit emission for nighttime power	kg/kWh	0.266
E_n	Nighttime power consumption	kWh	–
CD_g	CO_2 emissions for gas	kg	–
CD_a	Unit emission for gas	kg/m3	2.29

Table 9 Case study results of primary energy, running cost and CO_2 emissions

		Standard order	Case(1)	Case(2)	Case(3)	Case(4)	Case(5)	Case(6)
Primary energy	Electric power	0.958	1.399	1.139	0.948	0.615	0.874	0.777
	Gas	1.062	0.470	0.732	1.070	1.291	1.060	1.269
	Sum	1.018	0.857	0.901	1.019	1.010	0.983	1.065
Running cost	Electric power	0.904	1.237	1.079	0.933	0.718	0.841	0.767
	Gas	1.054	0.536	0.765	1.061	1.255	1.053	1.236
	Sum	0.980	0.884	0.921	0.997	0.988	0.948	1.003
CO_2 emission	Electric power	0.957	1.425	1.149	0.947	0.627	0.876	0.782
	Gas	1.062	0.470	0.732	1.070	1.291	1.060	1.269
	Sum	1.030	0.760	0.859	1.032	1.089	1.004	1.121