PRELIMINARY RETRO-COMMISSIONING STUDY ON OPTIMAL OPERATION FOR THE HEAT SOURCE SYSTEM OF A DISTRICT HEATING/COOLING PLANT

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ABSTRACT

In order to improve the energy performance of a district heating and cooling (DHC) plant, the expected performance of the plant is studied using simulations based on mathematical models. An complete heat source system model, equipped with an embedded module that automatically determines the on/off states of heat source equipment using cooling/heating loads, has been developed and validated using actual performance measurements. The mean error between the simulated and measured total energy consumption was 4.2%. Using the developed model, three proposals for improving the plant operation are simulated in order to determine how much energy can be saved. The simulation result shows that the three proposals, automating primary water flow rate, fully open bypass valve of heat exchanger during no-ice-thermal-discharge period, and increase chilled water supply temperature to 8°C, could reduce plant total energy consumption by 2.1%, 0.7% and 3.3% respectively.

INTRODUCTION

District Heating and Cooling (DHC) systems

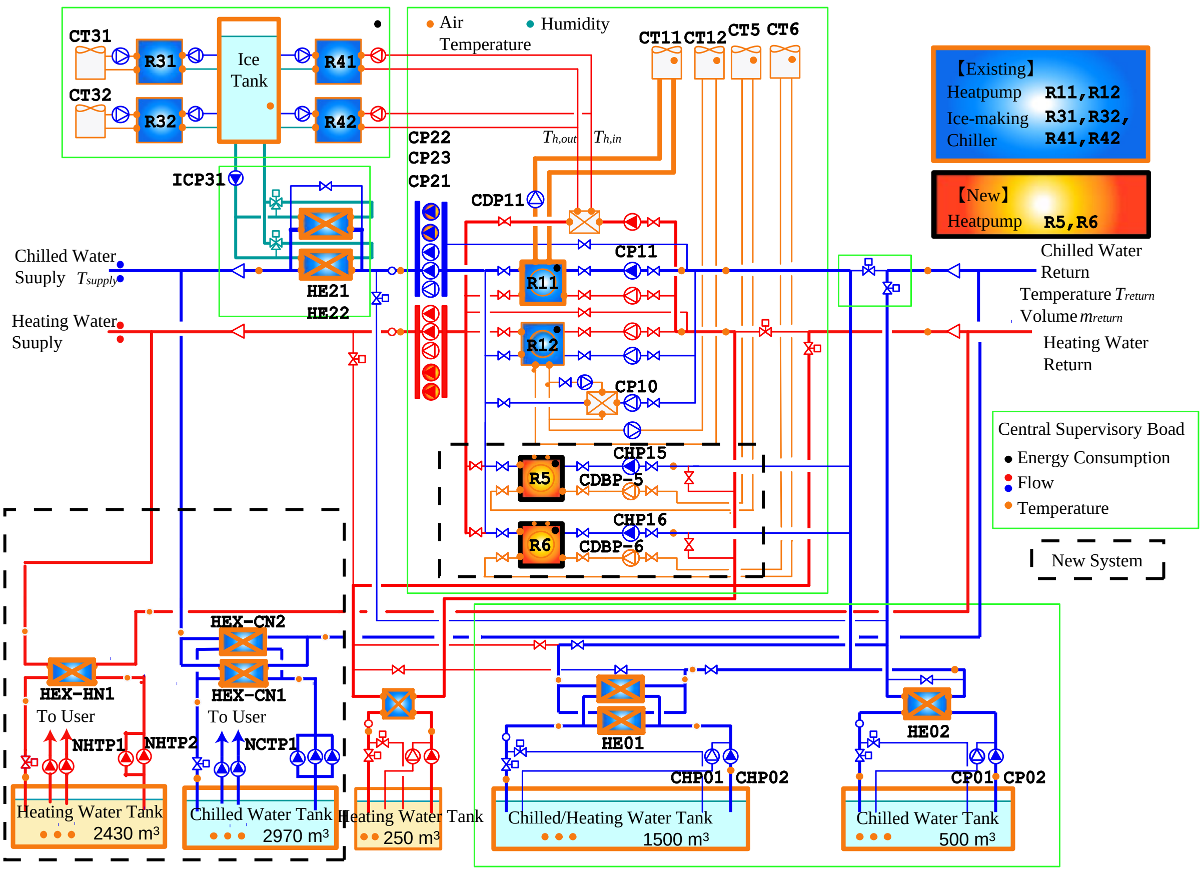

Figure 1: Heat source system diagram

attract attention due to their ability to heat and cool buildings with higher energy efficiency and reduce harmful waste emissions when compared to traditional heating/cooling systems. However, the Coefficient of Performance (COP) of DHC systems has often been reported to be less than one because many DHC systems are not being operated properly. This indicates that it is important to operate DHC systems effectively and efficiently, which requires the development and use of an enhanced operation mode that improves energy efficiency while fulfilling required heating and cooling demands. The purpose of this research is to verify the heat source system performance of an existing DHC system. The authors had previously reported the construction of the system simulation model before 2006 (existing system) and three case studies aimed at optimizing DHC system operation (Shingu et al., 2006) have been reported.

However, in the simulation of the existing system, the on/off states of heat source equipment were determined using measurement data, which are decided according to a certain priority order. So the former model cannot analysis the performance of the plant running at the priority orders different from present one. Therefore, in this paper, the entire simulation model has been improved by embedding a module that is capable of automatically determining the on/off states of various heat source equipment using cooling/heating loads according to a running priority order.

Furthermore, in 2007 the DHC plant under examination began supplying service to one more terminal user and was tasked with the responsibility of heating/cooling additional floor space, which required the introduction of two new heat pumps and related equipment. Thus, in order to determine the optimal operation methodology for the new system, the new devices were modeled and an entire new system model was developed.

In this research we will preliminarily investigate the commissioning of the DHC plant following its upgrade. This research is still ongoing and in this paper, as a preliminary study, the following three proposals were analyzed by simulation using the developed model: 1) Automatic control of primary chilled water flow rate, 2) Changing the opening of the bypass valve for the ice thermal discharge heat exchanger, and 3) changing the set point temperature of the chilled water supply.

PROFILE OF THE DHC PLANT

The DHC plant examined, which is located in Osaka, Japan, has cooled and heated two buildings

Table 1: Heat Source Equipment

	Heat Source Equipment	Name	Capacity (Chiller: RT, Tank: m^3)	Number
Existing	Two-centrifugal-compressor heat pump	R11	700	1
	Centrifugal heat pump with heat recovery	R12	700	1
	Ice chiller with heat recovery	R41, R42	37	1, 1
	Air source screw ice chiller	R31, R32	75	1, 1
	Chilled water tank	-	500	1
	Chilled-heat water switchover tank	-	1500	1
	Heat water tank	-	250	1
	Ice thermal storage tank	-	150	1
Newly introduced	Three-screw-compressor heat pump	R5, R6	450	1, 1

(one office building and one building with hotel rooms and leisure facilities) since November 1992. In 2007, the DHC plant began servicing another office building. Figure 1 shows the system diagram of the plant. New and original equipment is shown in different colors. Table 1 provides additional information on the primary plant equipment.

The present sequence of operating heat source equipment is R5, R6, R11, and R12. Ice chillers are operated according to heating demand with a normal sequence of R41, R42, R31, and R32.

VERIFICATION AND IMPROVEMENT OF THE COMPLETE SYSTEM MODEL

The complete system model, including the embedded module for determining automatic on/off states, was developed and used to simulate plant performance. Comparisons of the simulated energy consumption to actual measured data show that the simulations used for heat pump R11, ice chiller with heat recovery R41 and R42, and ice chiller R31 and R32 are not sufficiently accurate and need to be improved.

Subsystem of R11

The simulation of heat pump R11 lacked accuracy because the model for primary pump CP11 was unable to simulate water flow rates accurately (Figure 2). As a result, measured flow rates and head pressures of the model for pump CP11 were adjusted.

Since the pipe did not have an installed water flow rate meter, and because attaching sensors to the outside the pipe was considered impractical, water flow rate was estimated using the measured pump power consumption and pressure heads. Then the estimated flow rates were used to compensate the pump model by multiplying a coefficient k to the simulated flow rates.

After pump model was adjusted, the accuracy of the water flow rate simulation improved. Currently, the bypass water flow rate F_b and mixed water temperature T_a can be estimated accurately, which directly influences the calculation accuracy of the cooling amount and R11 power consumption. As shown in Figure 3, the R11 power consumption simulation accuracy showed significant improvement after the pump model was adjusted. However, there are still occasions when the simulated power consumption results differ significantly from measured values. The reasons for the variation are considered to be: 1) The simulation is set up to operate one of the two compressors from 0300 hours but in actual operation, it is manually controlled by the operators and the compressor operation start time does not always occur at 0300 hours. 2) Furthermore, during the thermal discharge period, the actual operating compressor was not always the one stipulated in the predetermined rule. The automatic on/off determination module determines equipment on/off states based on predetermined rules, but actual operation conditions are not always in conformity with those rules. This is why simulation one differs from the measurement. However, this difference was not expected to influence the conclusion as it relates to the purpose of determining the optimal plant operation method. This is because the optimal operation method can be determined by comparing the simulated performance levels of several operation methods to the simulated performance level of present operation method instead of actual measured performance.

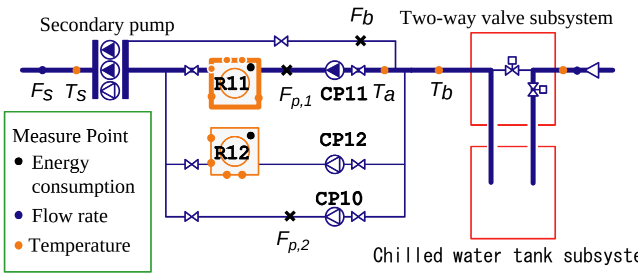

Figure 2: Diagram of R11 subsystem. [subsystem truncated in above figure.]

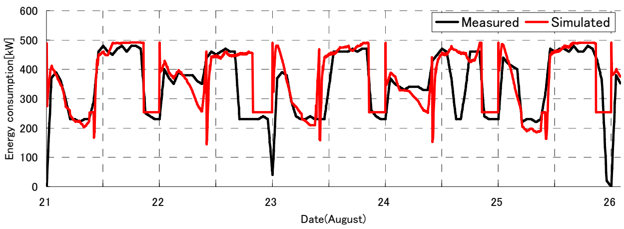

Figure 3: Simulation accuracy for R11 power consumption after pump model adjustment.

R41 and R42 Subsystems

R41 and R42 are two ice chillers with heat recovery systems. However, the primary function of these devices is to heat water by means of their heat recovery function. Ice-making is a secondary operation. In our simulation, the on/off states of R41 and R42 were determined based on brine temperature and the predetermined rule that the chillers are to be stopped when the brine temperature is lower than –5.8°C. However, during actual operations, in addition to brine temperature, the devices are controlled based on heated water temperatures and the status of the hot water tank. This research, however, was restricted to cooling operations and heating systems were not modeled. Therefore, the measured results of the heated water temperatures were added to the simulation inputs and the simulation accuracy of R41 and R42 improved. Power consumption simulations, before and after use of the heated water temperature, are shown in Figure 4 where they are compared with measured values.

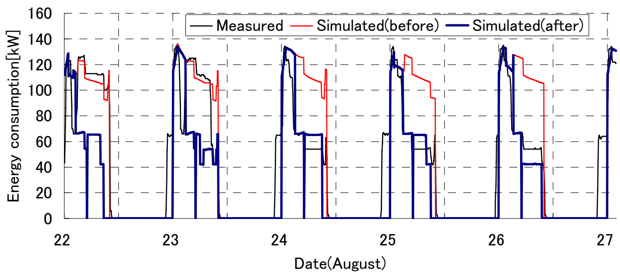

Figure 4: Simulated Power Consumption levels of R41/R42 before and after heated water temperatures were applied to the simulation

Subsystem of R31 and R32

Ice chillers R31 and R32 are only operated at nighttime the price of power is 2/3 cheaper than that of daytime. The chillers use brine to make shaved ice. The increased amounts of ice, also increase brine density and temperatures decrease. When the brine temperature at the inlet chillers decreases to –5.5°C, the ice thermal storage tank is considered to be full and the chillers' state is judged to be off. However, the simulation using this predetermined judgment condition was also unable to match actual performance. As shown in Figure 5, the simulated Ice Packing Factor (IPF) is largely different from the actual IPF. This is because the simulated higher IPF causes lower brine temperatures, which triggers fewer operations of R31 and R32.

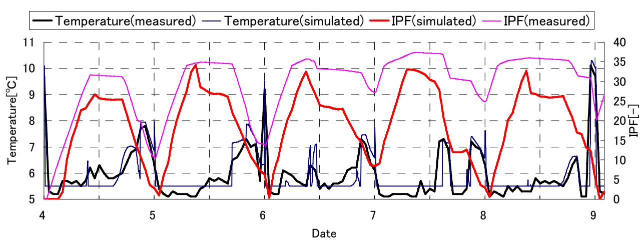

Figure 5: Simulated IPFs and chilled water temperatures compared with measured values.

Under present conditions, operation ice thermal discharge is started when the chilled water supply temperature is higher than the temperature set point of 6.5°C. This control algorithm is also used in simulation, which is the reason why the simulated IPFs are larger than the actual ones. If the simulated chilled water temperatures are lower than 6.5°C, thermal discharge will not start. Thus the simulated IPFs are larger than the actual recorded values. Even if actual power consumption is used to determine the actual on/off state, instead of the automatic judgment function, the IPF simulation accuracy will not be greatly improved. This is believed to be because the simulation time interval (5 minutes) is different to that of measured power consumption (1 hour). The measured 1-hour interval input data is converted to 5-minute intervals using linear interpolation. During the startup and shutdown periods, heat source equipment will not operate for the full hour. Thus, it is necessary to determine and use accurate on/off state time changes instead of using linear interpolation.

Generally, heat source equipment is stopped if the cooling load is lower than a threshold E_{TH}. For an example shown in Table 2, if the cooling loads fulfill the condition of $X_{t-1} > E_{TH} > X_t > X_{t+1} = 0$, the equipment should be stopped between time t-1 and t (Figure 6). This is because X_t is the sum of cooling load between t-1 and t, and the exact stop time can be calculated as shown in Equation (1) and (2).

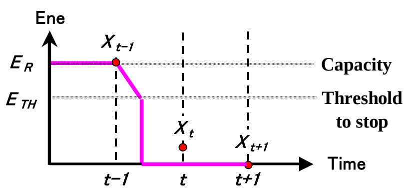

Figure 6: Actual on/off state change at shutdown.

$$\{t-(t-1)\}\,X_t = \frac{(X_{t-1}+E_{TH})t_{off}}{2} \qquad (1)$$

$$t_{off} = \frac{2X_t}{(X_{t-1}+E_{TH})} \qquad (2)$$

Where:

E_{TH} : Threshold, (kW)

t: Time, (hour)

t_{off}: Time of heat source equipment stop, (hour)

X: Cooling load, (kW)

Subscription: t-1, t, t+1: Time step, (hour)

Same as shut down, if the cooling loads fulfill the conditions of $0 = X_{t-1} < X_t < E_{TH} < X_{t+1}$, the equipment should be started between time t and t+1. When considering heat source equipment that is normally run at full capacity immediately after startup, the exact start time can be calculated using Equation (3).

$$t_{on} = \frac{X_t}{E_R} \qquad (3)$$

Where:

E_R : Rated cooling capacity, (kW)

t_{on}: Time of heat source equipment start, (hour)

The above-mentioned method is used to prepare the 5-minute interval input data, after which the simulation is conducted again. As shown in Figure 7, the simulation accuracy is then improved and the simulated power consumptions of R31 and R32 match actual measurements well.

Based on the above-mentioned efforts on improving simulation accuracy, it was determined that the complete system model, with its embedded automatic on/off state determination module, could simulate the plant performance with an average error rate of 4.2% and a Root Mean Square Error (RMSE) of 17.0%, as shown in Figure 8. This is because the present automatic on/off state determination module only includes the algorithm rule for weekday operation. A five-day entire system simulation is shown in Figure 8. However, there are still times when the simulation does not match actual measurements, also as shown in Figure 8, where the periods are marked by blue vertical rectangles. Our investigation confirmed that the reason for the differences was that during these periods, the manual on/off state changes did not match the state change rule. This indicates that the simulation model equipped with the embedded automatic on/off state determination module can still be considered sufficiently accurate if is to be used for studying the optimal operation methods.

Table 2: An example of time series data

Time [h]	· ·	*t-1*	*t*	*t+1*	· ·
Ene [kW	· ·	X_{t-1}	X	X_{t+1}	· ·

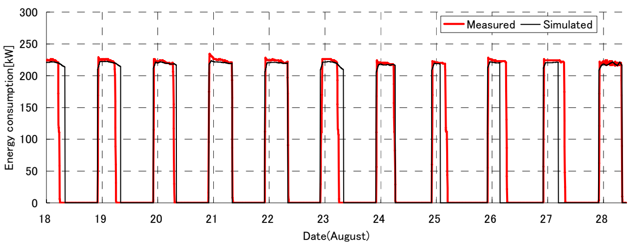

Figure 7: Simulated energy consumption of R31 and R32 compared with actual measurements.

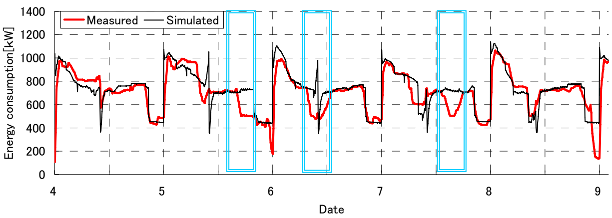

Figure 8: Simulated energy consumption of entire system compared with actual measurements.

RESULTS OF FORMER PROPOSALS USED IN THE NEW SYSTEM

The authors previously presented the following three proposals for improving operations of the DHC plant: 1) Restrict use of the ice-thermal storage system as much as possible because of its low COP (approximately 1.6); 2) Maintain chiller water return temperature at its designed value of 13.5°C by adjusting water flow rate; 3) Set cooling water temperatures at optimal values (Shingu et al., 2006). The new system is being operated according to those proposals. Table 3 shows the new chilled water temperature range compared to the temperature range of the old system. Two new high COP chillers have been introduced as well. As a result, the primary energy COP for the plant has increased from 0.75 (the average value in 2006) to 1.02 (the average value from January to October in 2007), as shown in Figure 9.

MODELING AND VALIDITION FOR THE NEW SYSTEM

Modeling for the newly-introduced equipment included the following three steps: 1) Development of a mathematical model using the specification data; 2) Validation of the model using field measurement data; 3) Fine-tuning the model coefficients if the measured performance was different from the specification performance.

Heat Pumps

Two new type heat pumps, each with identical capacities and COP were introduced. They were named R5 and R6 respectively. This type of chiller consists of three compressors and is capable of automatically determining which compressor to run based on the cooling/heating load. We attempted to apply the mathematical equation used for the existing heat pump to this new device, but the accuracy level was insufficient. Therefore a new method was developed to model the new heat pump.

a) Modeling based on specification curve.

The specification curves are expressed using the relations between the COP and the load factor r_Q, cooling water inlet temperature T_{ci}. The curve shape shows that it is difficult to fit the curves using a 2-power equation, so the 4-power equation, shown in Equation 4, was used. The cooling water inlet temperature T_{ci} and Heating/cooling load Q_e were transformed to non-dimensional variables in order to make the value 1 when the variables are at their nominal rated values.

Furthermore, in order to take into consideration the influence of the chilled water outlet temperature T_{eo}, the chilled water flow rate M_e and the cooling water flow rate M_c, Equation 4 was multiplied by the 2-power term of the non-dimensional T_{eo}, M_e, and M_c. Here the equation (without multiplication to save space) is shown.

The model coefficients are fitted very accurately because the RMSE of r_{cop} is as small as 0.084 (Figure 10).

$$r_{cop} = a_1(a_2 r_{T_{ci}}{}^2 + a_3 r_{T_{ci}} + 1)(a_4 r_Q{}^4 + a_5 r_Q{}^3 + a_6 r_Q{}^2 + a_7 r_Q + 1) + a_8 \quad (4)$$

$$r_{cop} = \frac{COP}{COP_r} \quad (5)$$

$$r_Q = \frac{Q_e - Q_{e,r}}{Q_{e,r}} \quad (6)$$

$$r_{T_{ci}} = \frac{T_{ci} - T_{ci,r}}{T_{ci,r} - T_{ei,r}} \quad (7)$$

Where:

COP: Coefficient of Performance

Q_e : Heat/cooling load, (kW)

T_{ci} : Cooling water inlet temperature, (^oC)

T_{ei} : Chilled water inlet temperature, (^oC)

Subscription *r*: Rated value

b) Model verification and revision

The measured power consumption levels were compared with model-simulated values in order to determine the performance of the heat pump. Our results showed that the measured power consumption levels were larger than the simulated levels, which indicate that the actual heat pump performance is incapable of reaching its specification performance level. In order to more

Table 3: Chilled water supply temperature before and after 2006

	Supply Temperature (℃)			Return Temperature (℃)		
	Standard	Range		Standard	Range	
		Before 2006	After 2007		Before 2006	After 2007
Chilled Water	6	5.0～7.0	4.0～8.0	13	12.0～14.0	11.0～15.0
Heating Water	47	45.0～49	44.0～50.0	40	38.0～42	37.0～43.0

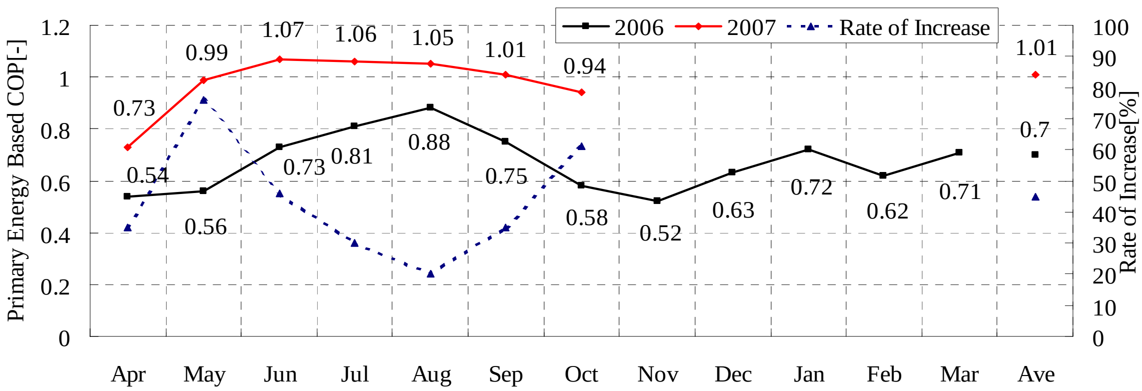

Figure 9: Primary energy based COP before and after 2006

accurately simulate the actual performance of the heat pump, a compensation coefficient *k* was introduced using the ratio between the sum of measured data and the simulated data. The coefficient *k* of heat pump R5 and R6 are 1.12 and 1.02 respectively, which indicates that the actual performance of heat pump R5 was 12% lower than the specification performance. This difference of 12% is so large that it cannot be ignored and needs a separate investigation, which is not included in this research. The simulation results for R5, after adjustment, are shown in Figure 11.

Cooling Tower

The newly-introduced heat pumps R5 and R6 are cooled/heated by two closed water circuit type cooling/heating towers CHT5 and CHT6, which achieve heat transfers by means of fin-coils. The open type cooling tower model was used to simulate the performance of R5 and R6. The simulated data was compared with measured data and it was determined that the simulation accuracy was acceptable.

a) Power consumption of cooling tower fan.

The cooling tower fan is installed on a frequency inverter in order to vary the air flow rate according to rating of the operating compressor number. As a result, the fan power consumption varies according the power frequency. The air flow rate has a linear relation with the power frequency, as shown in Equation 8. Fan power consumption is assumed to be proportional to the 3-power of the air flow rate, as shown in Equation 9.

$$I = \left(\frac{N}{N_r}\right) = \left(\frac{G}{G_r}\right) \quad (8)$$

$$E_{CT} = \left(\frac{G}{G_r}\right)^3 E_r \quad (9)$$

Where:

I: Ratio of power frequency to rated frequency (60 Hz)

N: Rotational speed, (rpm)

G: Fan air volume flow rate, (m^3/h)

E: Fan power consumption, (kW)

Subscriptions:

r: Rated nominal value

CT: Cooling tower

b) Accuracy verification and model compensation.

The fan power consumption levels calculated using Equation 9 were compared with measured

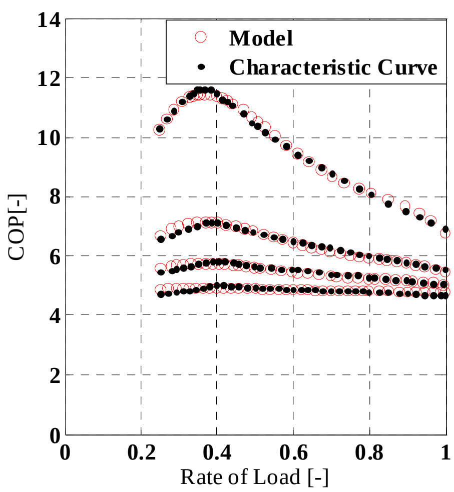

Figure 10: Model accuracy of R5

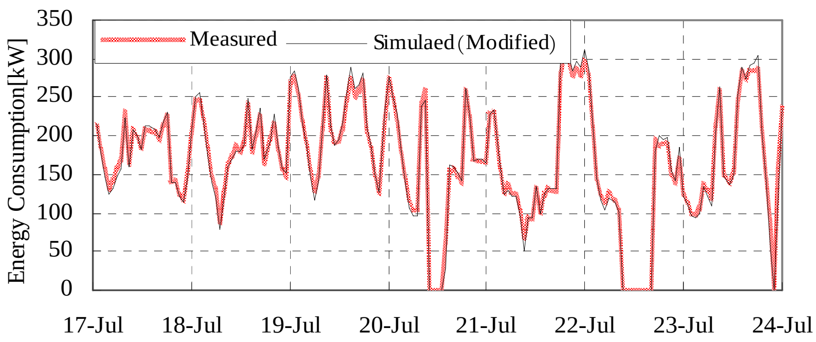

Figure 11: Simulated energy consumptions for R5 after compensation compared to measured values.

values. It was found that error levels increased as the power frequency decreased. This is because inverter efficiency was not being taken in consideration. Therefore, a regression coefficient that considers the influence of inverter efficiency was used to adjust the power consumption level, as shown in Equation 10.

$$E_{CT} = k\left(\frac{G}{G_r}\right)^3 E_r = a\left(b_1 I^2 + b_2 I + b_3\right)\left(\frac{G}{G_r}\right)^3 E_r \quad (10)$$

Where:

a, b_1, b_2, b_3: Fitted coefficients using the measured and calculated power consumption level using Equation 9

After adjustment, the calculation error improved from 9.3% to 1.7%.

Pump

Seven pumps were newly introduced, all of which are equipped with frequency inverters. The pump model is a mathematical model developed by the authors (Wang et al., 2004). Using this pump model, if two of the four variables for rotational speed, power frequency, flow rate and pressure head are given, the other unknown variables and power consumption levels can be calculated. The newly introduced inverter pumps include primary pumps, secondary pumps and cooling water pumps. Different pump types have different frequency control algorithms, so their known variables are different. Using this pump model, the power consumption of all pump types can be calculated.

Thermal Storage Tank

The building that recently was added to the load of the DHC plant is designed with a large water thermal storage tank installed under its basement and is equipped with a special water system that first uses the chilled/heated water supplied from DHC plant to cool/heat the water in the thermal storage tank by means of heat exchangers. The water in the thermal storage tank is then supplied to the terminal units. A complete water-mix model was used to simulate the temperature profile of the water in the initial, intermediate and final tanks.

The stored thermal amounts are empirically decided by operators using a seasonal schedule and based on the anticipated heating/cooling loads of the next day, which are predicted using weather forecast information in order to minimize the peak demand.

Complete Plant System simulation

The models for the newly introduced equipment were connected to the existing equipment models in order to construct a complete system model, after which the simulation accuracy

Table 4: Comparison of average energy consumption

		Average Energy Consumption				
		Sim [kW]	Measured [kW]	Error [%]	Total Error [%]	%RMSE [%]:
Total		1056.1	1095.0	3.6		8.8
Chiller	Total	750.9	796.9	5.8	4.20	12.0
	R5	173.5	182.4	4.9	0.82	19.2
	R6	168.0	195.2	13.9	2.48	22.9
	R11	290.9	295.1	1.4	0.39	14.4
	R12	90.0	89.6	-0.4	-0.04	46.3
CT	Total	43.7	43.5	-0.5	-0.02	16.0
Pump	Total	261.4	254.6	-2.7	-0.62	9.6
	CP24	20.0	18.3	-8.8	-0.15	33.7
	CP25	45.8	39.8	-14.9	-0.54	28.5
	CHP15	12.6	13.3	4.7	0.06	10.9
	CHP16	13.5	13.8	2.3	0.03	9.9
	CP11	36.3	36.2	-0.4	-0.01	10.1
	CP2	11.5	11.3	-1.0	-0.01	37.8

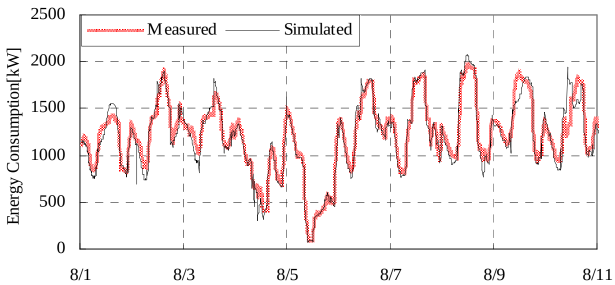

Figure 12: Comparison of simulated and measured total energy consumption of the new system

of the entire system was verified. The simulation period was 88 days from July 5th to September 28th and simulation interval was 10 minutes. The actual on/off statuses of the heat source equipment and the actual chilled water flow rates to each heat pump were used because they were manually set, which makes it difficult for them to be simulated. The average simulation error is 3.6% and RMSE was 8.8%. The comparison to total plant power consumption is shown in Figure 12. The average simulated and measured power consumption levels of the primary pieces of equipment are shown in Table 4. The “Total Error” listed in Table 4 contains the ratios of the differences between the simulated and measured total power consumption levels divided by the measured total power consumption levels. The simulation accuracy of the equipment consuming large ratios of power has a significant influence on total accuracy. The largest simulation error was detected at heat pump R6. This was determined to be caused by the fact that the designed chilled water outlet temperature set point was used during the simulation while the actual chilled water temperature of R6 was unable to maintain the set point temperature of 5^{o}C.

Because the total simulation error was as small as 3.6%, and because the large simulation error of R6 was considered due to an equipment fault, the simulation model was determined to be sufficiently accurate and use of the complete system model was deemed acceptable when analyzing proposals for improving DHC plant operations. The performance levels of the improvement proposals were evaluated by comparing the simulated performance of the proposals to the simulated results of present operations.

OPTIMAL OPRERATION FOR NEW SYSTEM

The following three proposals were analyzed using the previously mentioned models: 1) Automatically control the water flow rate of the heat pumps. 2) Increase the opening of the valve that bypasses the ice thermal heat exchanger during non-thermal-discharge periods. 3) Increase the chilled water supply temperature set point. The focus of 1) and 2) was to determine if the effects of improper system operation had been corrected. The third analysis focused on optimizing the operation of the entire system.

Automatically control water flow rate of heat pumps

Currently, the water flow rate is control manually. The chilled water pumps CHP15, CHP16 (which send water to heat pump R5 and R6 respectively) are equipped with inverters while the pumps of CP11 and CP12 (which send water to heat pump R11 and R12) are constant rotation pumps. As a result, the operator calculates the ratio of the water sent to R5 and R6 using Equation 11 and inputs the value through the plant management work station to control the flow rate.

Due to limitations imposed by measurement expenses, water flow rate meters were not installed in the circuits of R11 and R12. For that reason, the water flow rates were estimated using electric power consumption measurements and a mathematical pump model (Wang et al., 2004).

$$p = \frac{F_t - F_{11} - F_{12}}{F_t} \tag{11}$$

Where:

F_{11}: Water flow rate of heat pump R11, assumed to be constant because pump rotation is constant

F_{12}: Water flow rate of heat pump R12, assumed to be constant because pump rotation is constant

F_t: Total primary water flow rate, controlled to be same as the secondary water flow rate

p: Ratio of water flow rate of heat pump R5 and R6 to total flow rate

This control objective was used to make primary water flow rate the same as the secondary water flow rate by adjusting the inverter output of pumps R5 and R6 in a way that allowed primary pump energy to be saved. However, because the required control values were manually input by operators, it is difficult to input the values frequently and the level of input mistakes is considerable. Therefore, performance was simulated as if the flow rates were controlled automatically in order to determine how much difference would occur between manual control and automatic control. The simulation result shows that the sum of energy saving during the simulation period was 2.9% of the total plant energy consumption. The daily energy saving rates are shown in Figure 13. The maximum energy reduction rate was 8.1%. However, there were days when energy consumption actually increased. The reason for this was examined and it was determined to be because the operator input control values that were different from the control algorithm, i.e. improper inputs.

However, even if the water flow rate is controlled automatically, the outlet water temperature from R12 is often higher than the set point. This is because the chiller water pump for R12 is a constant ration pump and the water flow rate is higher than the rated value. Because of this, another simulation was conducted to determine how much energy could be saved if the water flow rate of R12 was reduced to its rated value. The result shows that the energy saving ratio reaches to 2.1%, which is 0.8% lower than the former simulation. However, the hours when the outlet water temperature was overset point decreased significantly from 78.3 hours to 4.0 hours when compared to the former result (Table 5).

Regarding cost, case 1 was found to be capable of reducing total operation cost by 3.3% and power demand (the maximum power) by 11.0%. It was also determined that case 2 was capable of reducing total running cost by 2.4% and power demand by 3.4%.

Reduce water flow resistance by increasing valve opening

The heat exchanger for ice thermal discharge was bypassed during the no-thermal-discharge period. However, the value opening at the bypass pipe is set at maximum of 55%. The operator

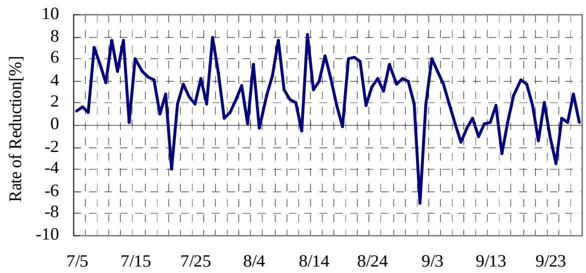

Figure 13: Daily energy saving rate by automatic control for primary water flow rate

Table 5: Energy saving rates by automatically controlling primary water flow rate

		Total hours of chilled water outlet temperature exceeding set point [h]	Rate of Reduction [%]												
				Chiller						Pump				
			Total	Total (Chiller)	R11	R12	R5	R6	Total (CT)	Total (Pump)	CP11	CP12	CHP15	CHP16
Measured Data Input		3.8												
Automated Control (AC)	Case 1 (AC)	78.3	2.9	2.0	-9.0	-17.0	15.7	17.5	3.3	5.2	0.2	0.5	28.6	30.9
	Case 2 (AC+R12)	4.0	2.1	1.3	-9.2	-1.4	10.1	12.1	1.6	4.3	0.2	7.0	24.9	27.5

explanation for this 55% opening were: 1) even if the valve opening is set at 55%, the water flow rate is not much different than when it is open 100%; 2) in order to prevent freezing in the heat exchanger at the beginning of a thermal discharge, maintaining the opening at 55% reduces the time required to close the bypass circuit if required. However the water flow resistance of 55% opening is higher than that of 100% which requires the pumps to consume more energy. An experiment was conducted to measure the resistant coefficients at valve openings of 55% and 100% respectively. The coefficient was then used to simulate the performance when 100% open. During the 88 days simulation period, the results showed that 0.7% of the plant total energy could be saved and total pump energy savings could amount to 2.6%. The total operating cost and power demand could be reduced by 0.3% and 0.5% respectively. Figure 14 shows the daily average energy saving rates for secondary pumps CP24 and CP25. The daily average energy saving is 7.0% and maximum rate is 11.4%. Even though the energy savings are not excessive in contrast to the total plant energy consumption, it is clear that the bypass valve opening should be set at 100% during the no-thermal-discharge period.

Change to Chilled Water Supply Temperature

The chilled water temperatures at the outlet of heat pumps was set lower than required. If the supply temperature could be set slightly higher, the return temperature would increase and the water temperature difference between the return water and thermal tank water would be larger. This would allow more stored thermal energy to be used and water thermal storage system would become more efficient. For our preliminary discussions, the cooling loads were assumed to be same as the present values. The differences between the supply water temperature and all the other temperature set points were also set to the same values as the present differences. The performance levels of the supply water temperature set point were changed in increments of 0.2°C from the present set point of 6°C to 8°C and the results were simulated. The set point values are shown in Table 6. The simulation results are shown in Figure 15. It was determined that if the supply temperature set point could be changed from the present 6°C to 8°C, 3.3% of the plant's total energy consumption and 4.6% of the total energy consumed by heat pumps could be saved. However, the plant's terminal users require 6°C chilled water and thus it is not possible to supply 8°C chilled water during the peak cooling load period. The possibility of changing the supply water set point according to the magnitude of cooling load will be a topic for future research.

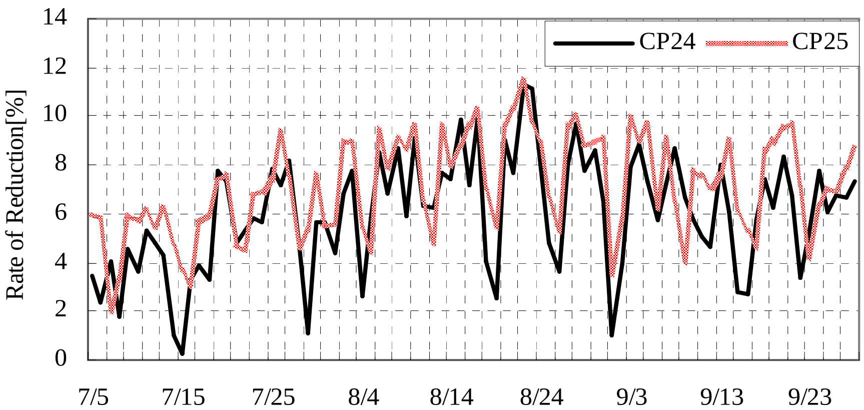

Figure 14: Daily Average Energy Saving Rates of Pumps CP24 and CP25 When Bypass Valve Opening Changed from 55% to 100%

Regarding cost, if the chilled water supply temperature can be maintained at 8°C, total operating expenses can be reduced by 3.1% and power demand can be reduced by 0.02%.

Furthermore, the DHC plant's performance based on combined adoption of all three proposals was simulated as well. The total summed energy saving was 5.8% cost saving was 5.7%, and power demand saving was 5.5%.

The control schemes for the three proposals are: 1) Control the water flow rate of R5 and R6 automatically according to equation 11; 2) Fully open the bypass valve during the no-ice-thermal –discharge period; 3) Set the chillers' chilled water temperature set point at 8°C.

CONCLUSIONS

A preliminary retro-commission study on the heat source system of a DHC plant was conducted to detect faults and determine more optimal operation procedures by means of a simulation. A simulation model for the heat source system was developed and used to check the performance results of three optimal operation proposals.

1) A complete heat source system model with an embedded function capable of automatically determining the on/off states of various equipment was developed. The simulation's accuracy was evaluated and improved by comparing the simulation data results with actual measured data. It was found that if the procedures used in an actual operation were different from established operational procedures, significant differences between the simulation and actual measurement would occur. However, it was also found that if the unusual operations were excluded, the simulated performance agreed with actual performance levels sufficiently to state that the simulation model is accurate enough to be used when analyzing improvement proposals.

Table 6: Temperature set points for simulation of chilled water supply temperature change

	Set Point Value [℃]						
	Supply	Outlet Chilled Water Temperature				Discharge	
		R5	R6	R11	R12	Chilled Water Tank	Chilled/Heating Water Tank
Standard [℃]	6	5	5	5	5.5	9	9
Case Study [℃]	θ_S	$\theta_S - 1$	$\theta_S - 1$	$\theta_S - 1$	$\theta_S - 0.5$	$\theta_S + 3$	$\theta_S + 3$

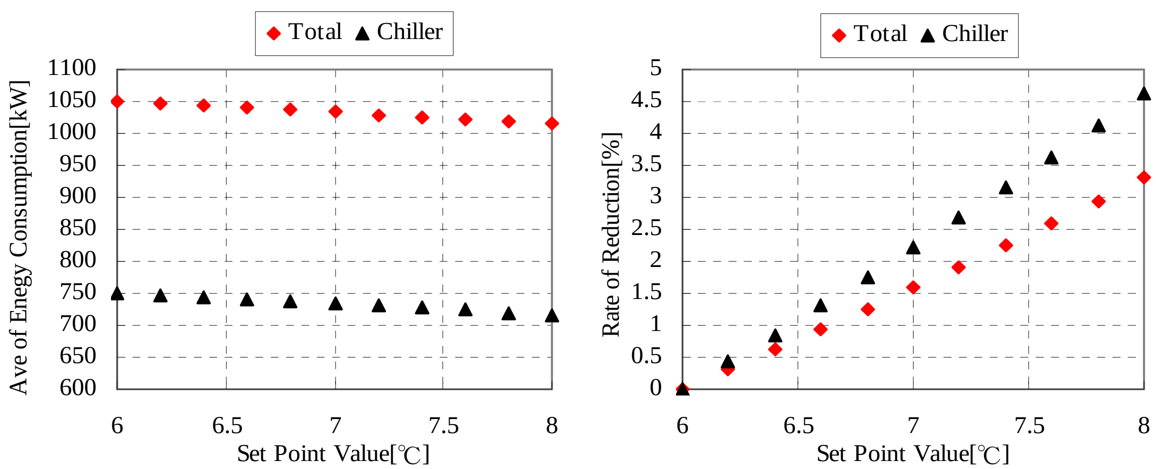

Figure 15. Energy Savings by Changing the Chilled Water Supply Temperature Set Point

2) The performance of the entire heat source system, including the newly-installed heat source equipment was simulated. The average simulated error of the plants total energy consumption was 3.6% of its measured value, and its RMSE was 8.8%. This indicates that the simulation is sufficiently accurate to be used when analyzing optimal operation methods.

3) The simulation for automatic control of the chilled water flow rates to the heat source equipment indicated that 2.9% of the plant's total energy consumption could potentially be saved. If the water flow rate to heat pump R12 was reduced to its rated value, the energy saving rate reaches 2.1% and the hours when the outlet water temperature is overset can be reduced from 78.3 h to 4.0 h.

4) The performance of the system when changing the opening of the valve that bypasses the ice thermal discharge heat exchanger when ice thermal discharge is not being conducted was simulated. If the valve opening could be changed from its present 55% to 100%, the total energy consumption would decrease by 0.7% and 7% of the energy expended by the secondary pump could be saved.

5) The simulation performed to determine the effect of changing the chilled water supply temperature from present 6^{o}C to 8^{o}C shows that 3.3% of the total plant energy consumed can be saved and that energy expended on heat source equipment can be reduced by 4.6%.

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