Oral Presentation | Oral Presentation：8.HVAC system modelling / simulation / evaluation

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Session I-8

Chair:Yuuki Matsunami

3:55 PM - 4:10 PM JST | 6:55 AM - 7:10 AM UTC

[1I08-02] Parametric Study for the Design Optimization of a Double-Coil Air Handling Unit System Using High-Temperature Chilled Water

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Keywords：High temperature chilled water、THIC air-conditioning system、Two-temperature chilled water、Energy Management

The purpose of this examination is to contribute to the establishment of optimal design and control of an air conditioning system using high-temperature chilled water. The air conditioning system in this study is a double-coil air handling unit system that uses 12°C medium-temperature chilled water and 7°C chilled water, and is designed to meet the outdoor air load and room load individually.In order to understand the actual operating characteristics of this system, we analyzed the actual operation data of the implemented system and evaluated the heat extraction rate and heat transfer efficiency (WTF) of 12°C and 7°C chilled water. On a typical day in the height of summer in August, the heat processed by the air conditioning system was about 40% for 12°C chilled water and about 60% for 7°C chilled water. In addition, the WTF was higher for the 7°C system in the summer, and the 12°C system showed higher values in the moderate season. Furthermore, the average temperature difference between the inlet and outlet of the chilled water during the day on a typical day in the summer was 6.5°C for the 7°C system and 5.3°C for the 12°C system.A system simulation model was then developed and its reproducibility was confirmed using actual operation data. Using this model, the energy performance of the double-coil air handling unit system was evaluated by comparing it with a conventional system that uses only 7°C chilled water. In addition, a parameter study was conducted by changing the capacity balance and order of the 12°C chilled water and 7°C chilled water coils, and the appropriate system design of the double-coil air conditioner using high-temperature chilled water was considered.

Case Study for the Design Optimization of a Double-Coil Air Handling Unit System using High-Temperature Chilled Water

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Abstract. This study investigates the optimal design and control of an air-conditioning system utilizing high-temperature chilled water. The system comprises a double-coil air-handling unit that operates with both 12 °C high-temperature chilled water and 7 °C chilled water, designed to independently meet outdoor air and room loads. To assess the system's actual operating characteristics, operational data from the implemented system were analyzed, focusing on the load distribution between the 12 °C and 7 °C chilled waters and overall energy performance. A system simulation model was developed and validated against the operational data. Using this model, a parametric study was conducted by varying the capacity balance and sequence of the 12 °C and 7 °C chilled water coils to determine the optimal system design for the double-coil air-handling unit employing high-temperature chilled water.

1 Introduction

In recent years, improvements in the thermal insulation and sun-shading performance of building envelopes, along with reductions in internal heat generation, have decreased the sensible heat load on office floors. As a result, the treatment of latent heat load has become increasingly important in air-conditioning systems. Under hot and humid weather conditions, the proportion of outdoor air load relative to the total air-conditioning load is expected to rise, necessitating effective handling of outdoor air loads. In unit-type air-handling units (AHUs), cooling and dehumidification are typically performed simultaneously using chilled water. Without a reheating process, humidity control remains inadequate.

As a result, temperature-and-humidity-independent control systems, such as desiccant air conditioners and double-coil air handlers, are receiving increased attention. Among these, double-coil air handlers can separate room loads from outdoor air loads and treat the room loads independently, allowing the use of high-temperature chilled water on the room-load side. This enables energy savings in the heat source system.

This study aims to investigate the operating characteristics of a double-coil AHU system using high-temperature chilled water installed in an office building. Additionally, optimal design and control strategies for the system are examined. The target system employs both 12 °C high-temperature chilled water and 7 °C chilled water, with two chilled-water coils designed to independently treat outdoor air and indoor loads.

To understand the actual operating characteristics of the system, this study evaluates the distribution of the heat load between the two chilled-water coils and the heat transport efficiency using operational data. A simulation model is then developed and its reproducibility is verified using the same data. Furthermore, based on a simulation analysis in which the order of the chilled-water coils is altered in the model, the appropriate design of the double-coil AHU system is examined.

2 Intended system

Figures 1 and 2 show diagrams of the underfloor AHU (UF-AHU) installed on an office floor and the ceiling-mounted AHU (CL-AHU) installed on another floor, respectively. These units handle the interior and outdoor air loads of the office space, while the perimeter load is managed by separately installed fan coil units (FCUs). The UF-AHU conditions the outdoor air using a 7 °C chilled-water coil (pre-coil), which is then mixed with

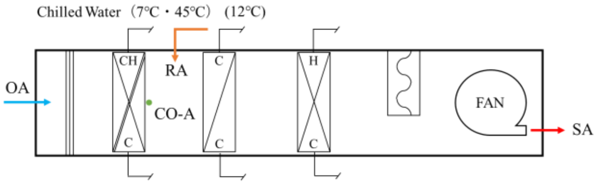

Fig.1. Underfloor AHU (UF-AHU)

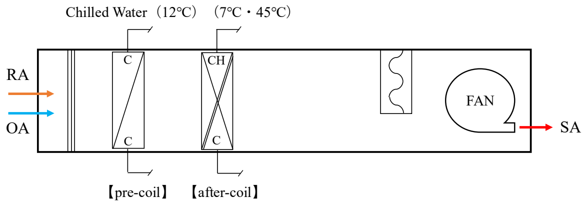

Fig.2. Ceiling AHU (CL-AHU)

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return air and further treated by a 12 °C chilled-water coil (after-coil) before being supplied to the room. In contrast, the CL-AHU mixes the outdoor and return air, pre-cools the mixture with a 12 °C chilled-water coil, and then cools it to the target temperature using a 7 °C chilled-water coil. The supply airflow rate is adjusted by a variable air volume system, and the outdoor air intake is controlled based on the CO_2 concentration of the return air.

T_{OA}: fresh air temperature [°C]
T_{ROOM}: room temperature [°C]
T_{SA}: supply air temperature [°C]
T_{SA_LR}: supply air temperature after load reset control
T_{after_in}: inlet temperature of the after-coil [°C]
T_{pre_out}: outlet temperature of the pre-coil [°C]
T_{after_out}: outlet temperature of the after-coil [°C]
x_{OA}: absolute humidity of the supplied air [g/kg']
x_{ROOM}: absolute humidity of the room [g/kg']
RH: relative humidity [%]
qs: sensible load handled by the air conditioners [kW]
ql: latent load handled by the air conditioners [kW]
ql_coil : latent load handled by the coils [kW]
V_{OA}: fresh air flow rate [m^3/h]
V_{SA}: supply air flow rate [m^3/h]
V_{SA_min}: minimum supply air flow rate (3450 m^3/h)
V_{RA}: return air flow rate [m^3/h]
c: specific heat capacity of air (1000 J/kg･K)
ρ: density of water (1.2 kg/m^3)
r: evaporation latent heat of water (2,500 kJ/kg)
W: water flow rate
LR: load reset control [-]

3 System behavior of double coil AHU

3.1 Operating Conditions of AHU

We evaluated the system behavior and energy performance based on operational data for 2024. The summer period was from April 15, 2024 to November 6, 2024, and the winter period was from January 1, 2024 to April 26, 2024 and from November 5, 2024 to December 31, 2024. The representative days were August 23 (Friday) for the summer period, June 24 (Monday) for the intermediate period, and January 4 (Thursday) for the winter period.

3.1.1 Operating Conditions of UF-AHU

Figure 3 shows the operating conditions of the UF-AHU, and Figure 4 shows the room and outdoor air loads processed by the air handler. Figure 3 shows that in both August and June, mainly 7 °C chilled water is used to process the loads. In addition, in both months, the room temperature during working hours is approximately 26 °C. and the RH values are approximately 58% and 60% in August and June, respectively. Therefore, sufficient humidity control is realized using chilled water at two temperatures, and a good indoor environment is established. Figures 3 and 4 also show that the load distribution between 7 °C and 12 °C chilled water differs from that between outside air and room loads. This is because the outside air is sufficiently cooled with 7 °C chilled water for humidity control, which results in the handling of the room load and a smaller load distribution of 12 °C chilled water. The amount of heat removed by the air handler and supply air flow rate during working hours are stable and that approximately 30% of the processing air-handler heat load is the outside air load. This indicates that the interior space is a stable working environment with no significant changes in the number of people.

3.1.2 Operating Conditions of CF-AHU

Figure 5 shows the heat removed by the CL-AHU. The air-conditioning load is mainly handled by 12 °C chilled water. In addition, the room temperature during the air-conditioning operation is approximately 26 °C in both months. The RH is approximately 62% in August, which is slightly higher than that using the floor-installed UF-AHU, as shown in Figure 3. Using this air-conditioning system, the efficiency of the heat source system may be improved because mainly 12 °C chilled water is used.

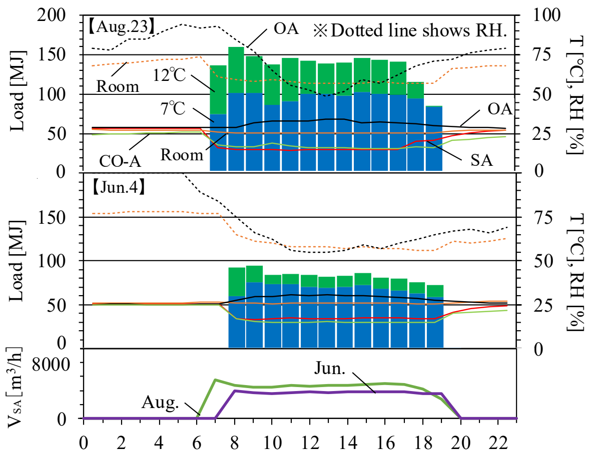

Fig.3. Removed heat and air rate of the UF-AHU

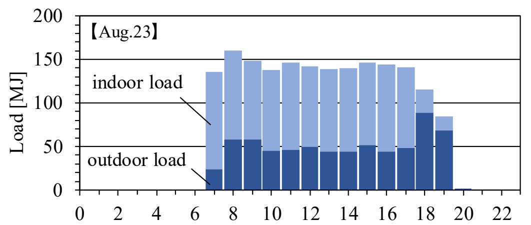

Fig.4. Removed heat of the UF-AHU (Aug.23)

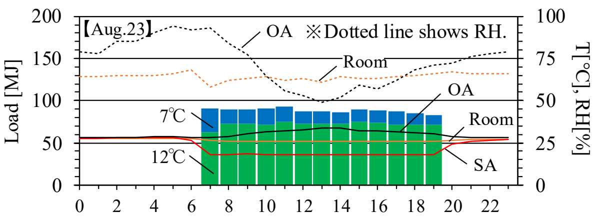

Fig.5. Removed heat of the CL-AHU

3.2 Efficiency of heat source systems

3.2.1 Effects of high-temperature chilled water utilization

Figure 6 shows the changes in the heat load of the heat source system on the representative days in the summer and winter. During the summer, when producing chilled water at 7 °C, a water-source heat pump chiller using groundwater (WHP) is operated first, followed by a turbo refrigerating machine (TR). For 12 °C chilled water production, the TR is operated first, followed by an absorption-type chiller (RA). During periods such as nighttime, when the chilled water load of the building is extremely low, only an air-source heat module chiller (AHP) operates. During winter, the hot water load is handled by operating the WHP, RA, and AHP in sequence.

Figure 7 shows the seasonal performance of the heat source. Note that all the calculated coefficients of performance (COPs) are the primary conversion values obtained using 9.76 MJ/kWh electricity and 45 MJ/m^3 gas. The ratio of heat load processing for the entire building's 7 °C and 12 °C chilled-water systems is approximately equal. This is because 12 °C chilled water is supplied to the FCUs. It is also because, as shown in Figure 5, the CL-AHUs on the office floor use 12°C high-temperature chilled water as much as possible.

The system COP (SCOP) values converted into primary energy in the summer and winter are 1.35 and 0.95, respectively. In addition, the SCOPs of the 7 °C and 12 °C heat source systems in the summer were 1.42 and 1.27, respectively. The slightly lower SCOP of the 12 °C chilled-water system compared to that of the 7 °C chilled-water system is related to the operation of the RA.

Figure 8 shows the relationship between the TR manufacturing heat load and individual equipment COP for cooling water temperatures of 31-33 °C during the summer and categorized by the manufacturing chilled water temperature. The average COPs in this range are approximately 1.9 and 2.2 for the 7 °C and 12 °C systems, respectively. Under similar outdoor conditions, the COP of the 12 °C chilled-water system is higher than that of the 7 °C chilled-water system.

3.2.2 Heat transportation efficiency

Figure 9 shows the chilled-water flow rate and round-trip temperature of the secondary-side header. The average water transportation factors in August are 40.6 and 29.2 for the 7 °C and 12 °C systems, respectively. The average round-trip temperature differences from 7 AM to 7 PM are approximately 6.5 °C and 5.3 °C for the 7 °C and 12 °C systems, respectively. This indicates that the temperature difference in the 7 °C system is larger than that in the 12 °C system, thereby resulting in a higher water transport efficiency.

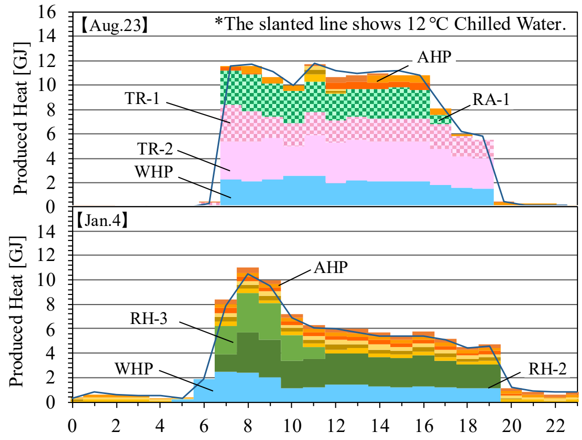

Fig.6. Produced heat by each heat source unit

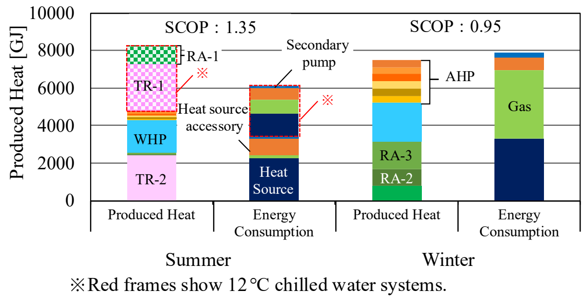

Fig.7. System seasonal performance (SCOP)

4 Modelling of double-coil air handler

4.1 Intended system

4.1.1 Development of AHU system model

The double-coil AHU system shown in Figure 1 was modeled using the LCEM tool [1]. This is referred to as the base case. Figure 10 shows the system model diagram, Table 1 lists the simulation boundary conditions, and Table 2 lists the cooling coil design capacity. In this model, the air condition at the pre-coil outlet is calculated, and the result is transmitted to the RA mixing object in the after-coil system.The condition of the mixed air is then determined from the ratio of the air rates of the return and pre-coil passing airs. This air becomes the inlet air for the after-coil. After cooling the air in the after-coil, it is supplied at a specified supply air temperature by a fan.

Among the boundary conditions listed in Table 1, the intake outdoor air flow rate is calculated from the opening degree of the outside air motor damper to approximately match the actual measured value of the heat removed by the air handler. Load reset control is also modelled for the supply air temperature control.

The air-conditioning room load, which is a boundary condition for the calculation, is calculated using equations (1) and (2) based on the actual operational data. Considering the actual supply air temperature supplied to the room by the underfloor air conditioning, 2 °C is added to the supply air temperature of the air handler in equation (1).

$$q_s = c\rho V_{SA}[T_{ROOM} - (T_{SA}+2)]/3600 \quad (1)$$
$$q_l = r\rho V_{SA}(x_{ROOM} - x_{SA})/3600 \quad (2)$$

Load reset control for the supply air temperature is applied to the after-coil. This is referred to as single-stage load reset control in this paper. Single-stage load reset control is applied when the supply air rate is smaller than the minimum air rate. The air temperature at the outlet of the after-coil is assumed to be 15 °C in the base case, and the supply air temperature after load reset control (T_{SA_LR}) is calculated using equations (3) and (4).

Furthermore, when this control is applied, the supply air rate is minimum, and the inlet and outlet temperatures of the after-coil are calculated using equations (5) and (6). If the supply air temperature after load reset control satisfies equation (7), the water supply to the pre-coil is stopped, because the air at the outlet of the after-coil requires heating.

$$T_{SA\_LR} = T_{ROOM} - \Delta\theta \quad (3)$$
$$\Delta\theta=(q_s \cdot 3{,}600) / (V_{SA\_min} \cdot c\rho) \quad (4)$$
$$T_{after\_out} = T_{SA\_LR} - 1 \quad (5)$$
$$T_{after\_in} =(T_{pre\_out} \cdot V_{OA}+T_{ROOM} \cdot V_{RA}) / V_{SA} \quad (6)$$
$$T_{after\_in} < T_{after\_out} \quad (7)$$

4.1.2 Validation of model reproducibility

The reproducibility of the simulation model was verified using the actual operational data as the boundary conditions.

The supply air temperature was nearly consistent with the actual measurement, and the temperature-setting value under load reset control was reproduced. However, when the conditions in equation (7) were met, the pre-coil operated in fan mode and the behavior differed from the actual measurements. In this case, the supply air temperature was higher than the set value, and the absolute humidity of the supply air increased. Consequently, e.g., an unhandled latent load of 362.2 kW was generated between August 20 and 23.

Table 1. Boundary conditions

Name	Capacity
Pre-coil	cooling capacity: 35.8 kW air flow rate: 2,250 m^3/h water flow rate: 64 L/min number of rows: 6 T_{pre_out}: 13.5 °C
After-coil	cooling capacity: 19.2 kW air flow rate: 10,300 m^3/h water flow rate: 34 L/min number of rows: 8 T_{pre_out}: 15.0 °C

Table 2. Coil design capacity (the base case)

outdoor conditions	V_{OA}: rated value 2,250 m^3/h
	T_{OA} [°C]:Operating data
	x_{OA} [g/kg']: Operating data
room conditions	q_s: Calculated using equation (1)
	q_l: Calculated using equation (2)
	T_{ROOM} [°C]: Operating data
	x_{ROOM} [g/kg']: Operating data
AHU operating conditions	Operating 0:OFF 1:ON
	Mode 0:Fan 1:Cooling 2:Heating

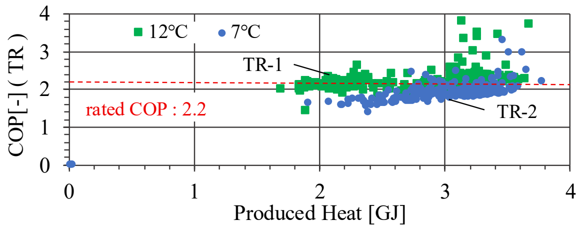

Fig.8. Produced heat and individual equipment COP (TR)

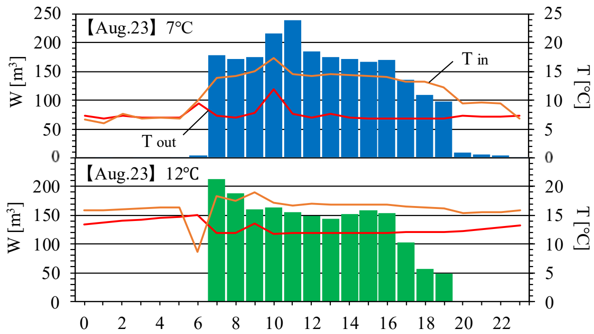

Fig. 9. Flow rates and water temperatures (7 °C・12 °C)

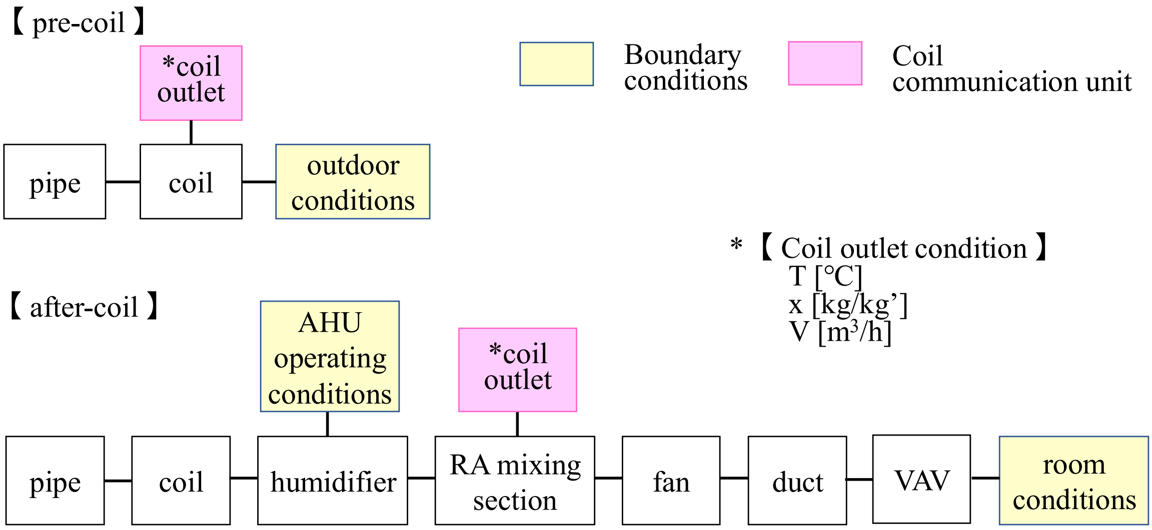

Fig.10. System model diagram using LCEM tool

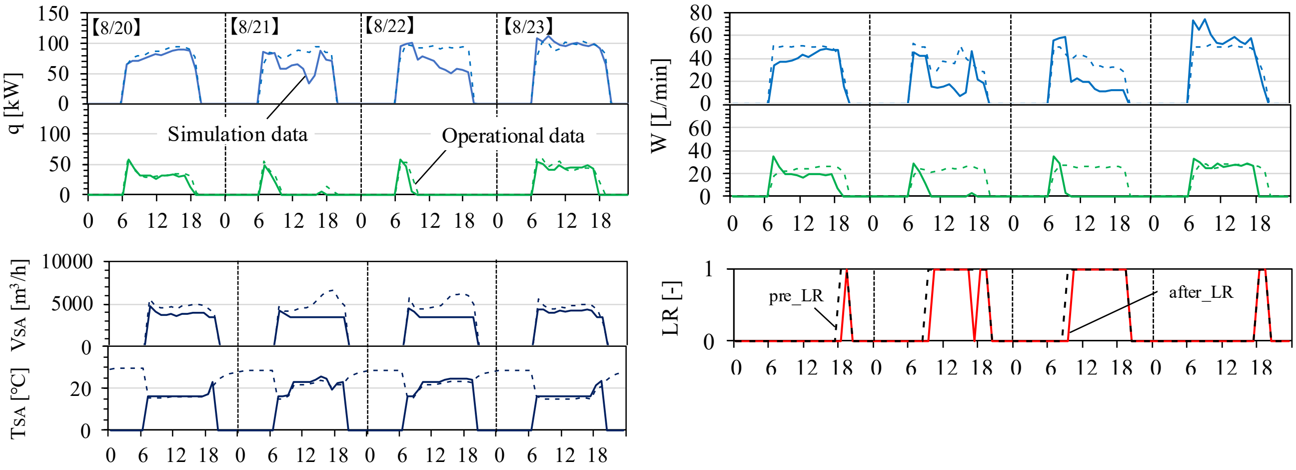

Fig. 11. System behavior using multistage LR model (the base case)

4.2 Modelling of multistage load reset control

In the single-stage load reset model, the latent heat treatment is insufficient under conditions satisfying equation (7); therefore, load reset control is also applied to the pre-coil. Because load reset control is applied to the pre- and after-coils, this is called multistage load reset control. When equation (7) is satisfied, load reset control is applied to the pre-coil. T_{pre_out} is calculated using equation (8) so that the supply air temperature becomes the reset temperature. In this case, only the pre-coil is used to satisfy the air supply temperature, and the after-coil is bypassed.

$$T_{pre\_out} = [(T_{SA\_LR} - 1)\cdot V_{SA} - T_{ROOM}\cdot V_{RA})]/V_{OA} \tag{8}$$

Figure 11 shows the operational data and simulation results when multistage load reset control is applied to the UF-AHU. The dotted and solid lines represent the measured values and simulation results, respectively. Applying load reset control to the pre-coil causes the pre-coil outlet air temperature to increase. Consequently, the heat-treatment capacity of the 7 °C coil is smaller than the measured value.

Figure 12 shows the relationship between the room load processed by the air conditioner in the base case and the supply air temperature and air rate. It also shows the relationship between the sensible heat load processed by each coil and the coil outlet temperature and water flow rate. When the room sensible heat load falls below approximately 12 kW, load reset control is activated, and the air rate is reduced to the minimum value. The pre-coil outlet air temperature and chilled water flow rate depend on the conditions of the outdoor air and room load.

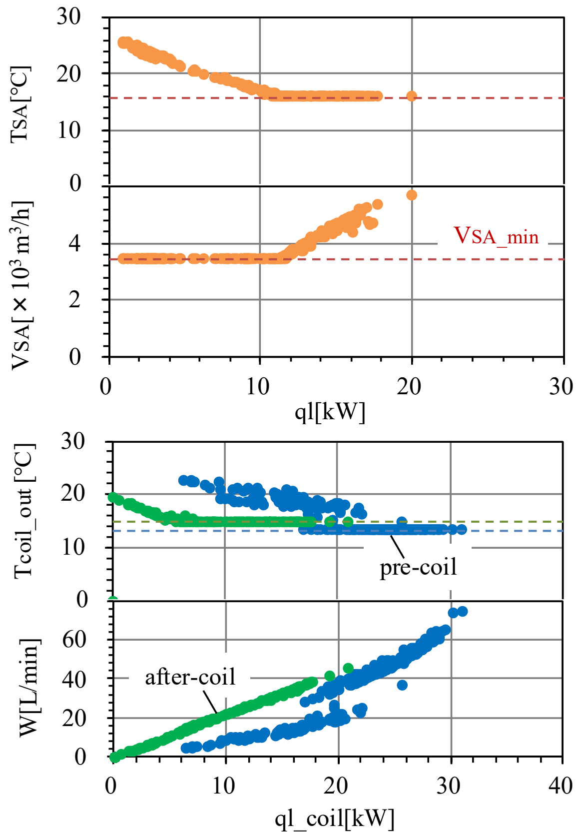

Fig.12. LR control results (base case)

4.3 Appropriate design of double-coil AHU

4.3.1 Simulation conditions

Simulations were conducted to analyze the effects of changing the water temperature supplied to the double coil. The aim was to observe the heat treatment process

Table 3. Coil design capacity (the replacement case)

Name	Capacity
Pre-coil	cooling capacity: 28.4 kW air flow rate: 2,250 m³/h water flow rate: 51 L/min number of rows: 8 T_{pre_out}: 17.0 °C
After-coil	cooling capacity: 37.1 kW air flow rate: 10,300 m³/h water flow rate: 66 L/min number of rows: 4 T_{pre_out}: 15.0 °C

when the water temperature through the pre- and after-coils was changed. Assuming multistage load reset control, the system was compared in base and replacement cases. In the base case, 7 °C and 12 °C chilled waters were supplied to the pre- and post-coils, respectively, similar to an actual air-conditioning system. In the replacement case, the chilled water temperatures for each coil were changed to 12 °C and 7 °C, respectively. Table 3 lists the water coil conditions for the replacement case. The outlet air temperatures for the pre- and after-coils are set at 17 °C and 15 °C, respectively. The outlet air temperature of the pre-coil was designed to be 5 °C higher than the chilled water temperature.

4.3.2 Comparison of system behavior

Figure 13 shows the heat removed by the air handler in the base case from August 26 to 30 as well as the air temperature, absolute humidity, and air flow rate during the heat treatment process. The system behavior on August 29 is similar to than on August 27; therefore, the corresponding results are omitted. Figure 13 shows that the load reset control is switched off from 6:00 to 18:00

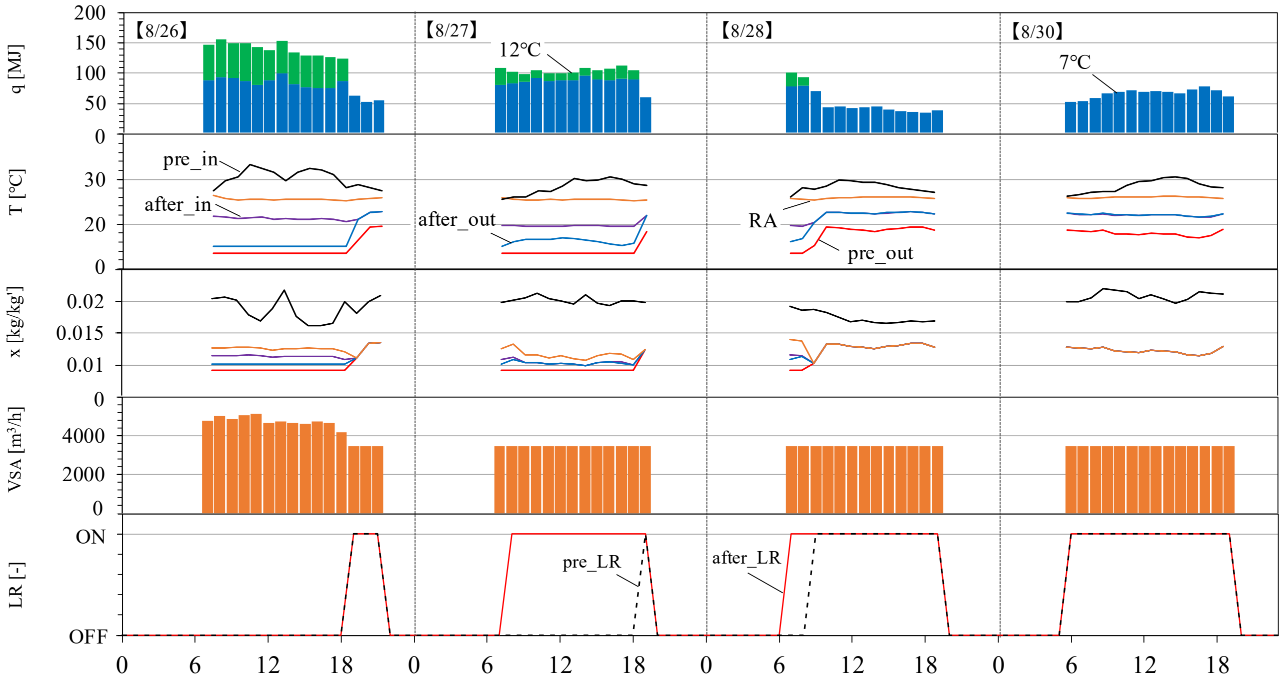

Fig. 13. System behavior with multistage load reset (LR) model (Base case simulation)

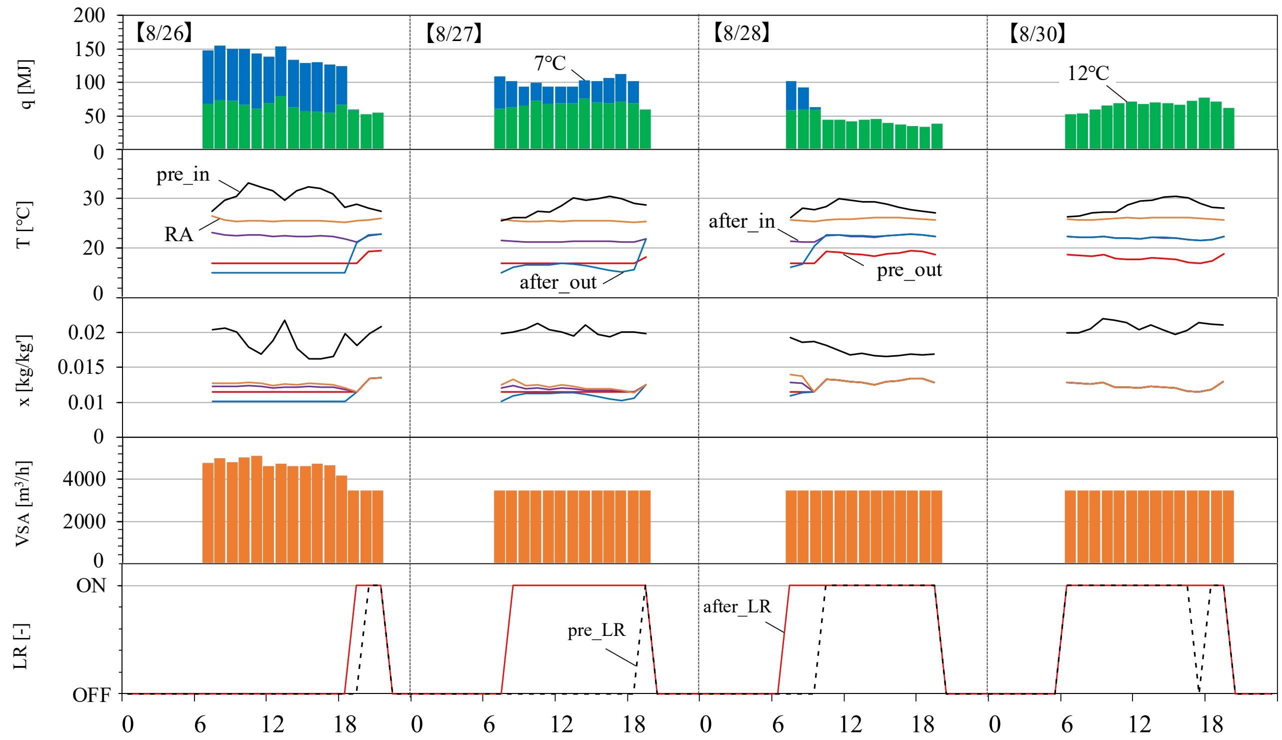

Fig. 14. System behavior with multistage LR model (Coil replacement case simulation)

on August 26. During this time, the heat removed by the pre-coil using 7 °C chilled water accounts for approximately 60% of the heat load of the air handler. During the working hours on August 30, single-stage load reset control is applied at all times. During this time, the supply air flow rate is minimum, and the pre-coil is bypassed. This causes the after-coil inlet temperature to increase. The total unhandled latent load during this period is 439.3 kW. This is caused by the single-stage load reset control, which increases the supply air temperature and results in insufficient humidity control by the chilled-water coil.

Figure 14 shows the heat removed by the air handlers in the replacement case from August 26 to 30 as well as the temperature and absolute humidity during the heat-treatment process. Because the after-coil outlet temperature setting is the same as that in the base case, the outlet air temperature, humidity, and supply air flow rate are the same in both cases.

On August 26, when the load reset control is switched off in the replacement case, the heat removed by the 12 °C and 7 °C chilled waters are similar. Compared to the base case, which is pre-cooled with 7 °C chilled water, the replacement case increases the heat treatment in the after-coil because the pre-coil pre-cooling temperature increases. However, because the supply air temperature is the same, the load treatment ratio of the 12 °C chilled water is increased.

On August 30, when load reset control is applied to both coils, the after-coil is bypassed; therefore, only the pre-coil with 12 °C chilled water is operating. In the morning of August 27, even when load reset control is applied only to the after-coil, mainly the 12 °C chilled water system is in operation. Therefore, in the replacement case, in which the ratio of high-temperature chilled water use is high, the heat source system can operate more efficiently.

Because in the replacement case, the supply air temperature of the air handler is the same as that in the base case, the supply air flow rate does not change. The total unhandled latent heat from August 26 to 30 is 180.6 kW, which is approximately 22% higher than that in the base case (148.2 kW). When the temperature of the chilled water flowing through the pre-coil increases, the dehumidification capacity of the outside air decreases; therefore, the base case is considered more suitable for strictly meeting the indoor humidity requirements. However, when the indoor humidity is within an acceptable range for operation, the replacement case has advantages in terms of energy performance.

5 Conclusions

This study examined the appropriate design and control of air-conditioning systems using high-temperature chilled water. Therefore, the operating characteristics of a double-coil AHU system installed in an office building were studied. Subsequently, using the developed simulation model, an analysis in which the order of the chilled water coils was changed was conducted. The main results were as follows:

・The room temperature and RH achieved using the UF-AHU (pre-coil: 7 °C, after-coil: 12 °C) were approximately 26 °C and 56%, respectively, in August. For the CL-AHU (pre-coil: 12 °C, after-coil: 7 °C), the values were approximately 26 °C and 62%, with RH slightly higher than that of the UF-AHU.

・The average COPs of the TR, under cooling water temperatures of 31–33 °C during summer, were approximately 1.9 and 2.2 for the 7 °C and 12 °C chilled-water systems, respectively. The 12 °C system showed a higher COP than the 7 °C system.

・The base case (pre-coil: 7 °C, after-coil: 12 °C) provided greater dehumidification capacity than the reversed configuration, making it more suitable for meeting strict indoor humidity requirements.

・In the reversed case (pre-coil: 12 °C, after-coil: 7 °C), the indoor humidity environment was not significantly compromised. This configuration is suitable for applications where a broader acceptable indoor humidity range is permitted and energy efficiency is prioritized.

References

Here are some examples:

1. LCEM Tool Ver3.10, Ministry of Land, Infrastructure, Transport and Tourism, Office of the Minister, Public Works Bureau, 2014