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メンブレンを用いた従来空調システムの潜熱処理能力増強に関する研究

Study on strength measure for latent heat handling ability by using membrane unit

正 会 員 ○尹　奎英（名市大）　　学生会員　鈴木笙悟（名市大）
千　成勇（漢陽大）　　趙　慧珍（漢陽大）
鄭　在原（漢陽大）

The sensible heat load of buildings has been decreasing due to improvements in the thermal performance of the building envelope, as well as the increased efficiency of lighting and office automation equipment[1]. As a result, the sensible heat load in the room is expected to decrease, and an air conditioning system based on the conventional dehumidification by supercooled co-ndensation is likely to have insufficient latent heat treatment capacity[2].

Membrane is a system that uses a moisture separation membrane to separate moisture from the treating air. Compared to the desiccant system, which removes moisture by the adsorbent, the membrane system does not require a re-heat source. The system itself is relatively simple, attracting a solution as a next-generation dehumidification system[3].

According to a previous study (Cho et al.)[4], it was shown that the energy efficiency of the membrane dehumidification method was relatively higher than that of the conventional supercooled condensation and re-heating. This technology is still under development, and issues such as energy efficiency and durability improvement remain addressed. However, with a thorough understanding of the technology level at the current stage, step-by-step dissemination of the technology can promote the research and development of the technology and shorten the time to practical application[5].

Therefore, this study investigates the application of this technology as a means of enhancing dehumidification capacity in conjunction with conventional air conditioning systems. In this paper, we reproduced the latent heat treatment capacity failure due to the reduction of internal heat gain, assuming a conventional air conditioning system with supercooled condensation. In addition, the dehumidification capacity of the system was increased by using a membrane in combination with the conventional air handling unit, and the associated energy efficiency and effects on the indoor thermal environment were evaluated.

1. Overview of the target building and air conditioning system

A typical office building assumed a conventional air-conditioning system based on dehumidification by supercooled condensation. The office building was supposed to be rectangular with a standard floor area of 900 m^2 and a long north-south axis. It has a single-core on the west side, and the rentable ratio is assumed to be 64%. The building and heat load parameters are shown in Table 1. The room occupancy schedule was given in accordance with the standard room use conditions[6].

An air conditioning system consisting of an air handling unit (AHU) and FCUs was designed for this target building. The 3.6m wide area around the perimeter was designated as the perimeter zone and the remaining area as the interior zone. The heat load is handled by the FCU in the perimeter zone and the AHU in the interior zone, and the AHU for the interior zone handles the fresh air introduced to the entire floor.

Table-2 shows the equipment list for the air conditioning system (only AHU for the interior zone). The maximum cooling load of the interior zone is 54.8W/m^2, and one unit is assumed to be installed on the floor. The design airflow rate is 4,600 m/h^3, and the diameter of the air supply duct is 50cm (maximum air velocity: approx. 6.5 m/s). The description of FCUs for the perimeter zone is omitted.

Table-1 Building conditions and air conditioning heat load conditions

Building use and structure	Office, SRC
Floor and ceiling height	4m, 2.6m
Exterior skin performance	Exterior wall U-value: 1.23W/m^2K Window glass U-value/SHGC: 2.63W/m^2K, and0.44 (Window area ratio:43 %)
Set room temperature and humidity	26℃/50%.
internal heat generation	Manpower density: 0.15person/m^2 Lighting heat gain intensity: 20W/m^2 Heat gain intensity of OA equipment: 20W/m^2
Amount of outside air introduced	30m^3/h/person
Air Conditioning Operation	Cooling period 6/1-9/30 7 am to 9 pm (Excluding weekends and holidays)

2. Air Conditioning System Operation Control and System Simulation

The model of the target air conditioning system was modeled using TRNSYS Ver18. The simulation period was from June 1st to September 31st, which is the cooling period. AMeDAS standard meteorological data for Tokyo was used as weather data, and simulations were conducted at 1-minute intervals. The airflow rate of the AHU was varied according to the room temperature of the interior zone. The design air supply temperature was set to 15°C, and the reset control was based on the heat load in the interior zone. The FCU for the perimeter zone was set to constant air volume operation.

3. Effect on indoor temperature and humidity due to decrease in internal heat gain

In order to simplify the analysis, the scope of this paper is limited to the interior zone to evaluate the thermal environment. To reproduce the effect of reduced internal heat gain, the intensity of lighting and OA heat gain was reduced from 20 W/m^2 in the reference case to 6 W/m^2 in the simulation. The heat gain intensity of 6W/m^2 assumes that LED lighting is adopted, and office automation equipment is made more efficient. The evaluation of the indoor thermal environment is based on the PMV value (-0.5 to 0.5) and the recommended values of less than 10% of unsatisfied occupant ratio and less than 12g/kg' of indoor absolute humidity[7) Note1)].

Figure-1 shows the comparison of the sensible heat factor in the room under different internal heat gain intensities. In the case of 20W/m^2, the time above the sensible heat factor 0.75 was 1,136 hours (88.1%) of the total operating time equal to 1,290 hours. In the case of 6W/m^2, it was 375 hours (29.1%) of the whole operating time, indicating a significant decrease in the sensible heat factor.

Figure-2 shows the comparison of indoor temperature and humidity for each case. There is no significant difference in indoor temperature between the two cases. On the other hand, the absolute indoor humidity is above 12g/kg' for 228hours and1,195 hours in the 20W/m^2 and 6W/m^2 cases respectively, indicating that the latent heat load is not being treated properly. Similarly, the relative humidity increased significantly in the 6W/m^2 case, and it was 1,183 hours above 60%. This is because the supply air temperature increased due to the reset control. The supply air volume was reduced by the decrease in sensible heat load, and a lower load has appeared. As shown in Figure-3, the air temperature in the 6W/m^2 case exceeds about 20 °C and the air humidity is as high as 15g/kg'.

Figure-4 shows the PMV comparison for each case. The PMV was estimated by giving the simulation results of room air temperature, humidity, and mean radiant temperature in the interior zone

Table-2 Equipment list of air conditioning system

Name	Specifications	Power
Unit type air conditioner	Number of rows: 7 Number of tubes: 20 Coil area: 0.53m^2 Air volume: 4,600 m^3/h x 615.8 Pa Cooling capacity: 49.3 kW Chilled water volume: 141.3litre/min (7-12°C) Heating capacity: 17.5kW Hot water volume: 50.2litre/min (40-45°C) Fresh air volume: 2,640 m^3/h Humidifying capacity: 16kg/h	SA 2.2kW RA 1.5kW

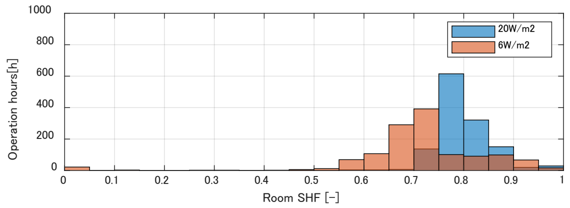

Figure-1 Sensible heat factor in a room with different internal heat gain

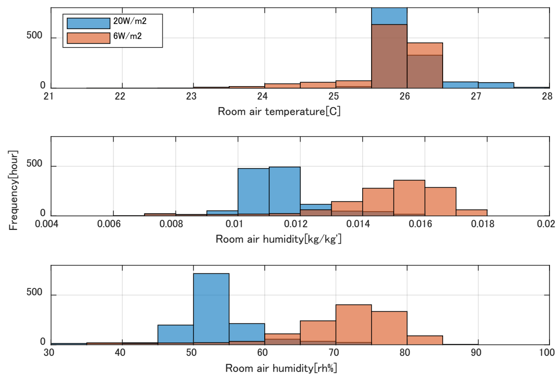

Figure-2 Comparison of indoor temperature and humidity at different internal heat gain

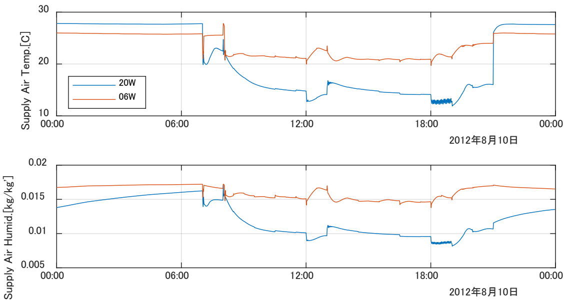

Figure-3 Comparison of supply air temperature and humidity for different internal heat gain

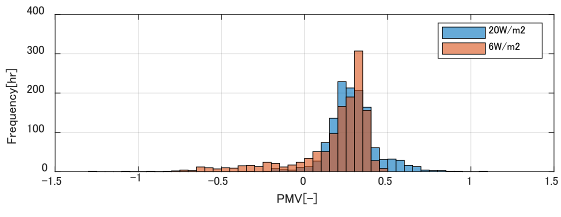

Figure-4 Comparison of PMV at different internal heat gain and the fixed values of air velocity 0.15m/s, Clo value 0.5, and Met value 1.0.

Due to the above reasons, the 6W/m^2 case showed a significant inadequacy of the latent heat treatment in the room, and the number of hours when the PMV was 0.3~0.35 in the 20W/m^2 case became large against that of the 6W/m case. In the 6W/m^2 case, there were times when PMV was negative. The PMV was outside the ± 0.5 range for 55 hours and 94 hours in the 20W/m^2 and 6W/m^2 cases respectively. The percentage of unsatisfied occupant ratio was 6.4% and 6.5 % respectively, and both cases satisfied the recommended value of less than 10%.

4. Evaluation of Dehumidification Performance Improvement Using a Membrane Mechanism

Figure-5 shows a conceptual diagram of membrane installation in a conventional AHU. A membrane device has three units consisting of a series of 5 hollow fiber membrane modules (see Figure-6) connected in series (treatment air volume: 600 m^3/h x 1500 Pa) and a vacuum pump of which vacuum side pressure is 1kPa. The membrane equipment is installed on the discharge side of the blower in the air supply duct to remove moisture from the conditioned air. In this study, the above 8 devices were connected in parallel (Unit's size: 0.32m x 0.6m x 0.32m), and the design air supply humidity after membrane treatment was set at 8g/kg'. The design dehumidification volume of the air supply was 27.6 kg, and the vacuum pump capacity was set by 20kW. With the introduction of the membrane equipment, the required static pressure of the blower became 2,115.6 Pa, and a 5.5 kW turbo blower replaced it. The simulation considering the mass-transfer process of the membrane[4] was conducted coupling with the energy simulation by using TRNSYS.

Figure-7 compares the air supply humidity before and after the membrane equipment was installed. The simulation was also performed where the vacuum pump was downsized to 5 kW. With the introduction of the membrane equipment, the air supply humidity remained at the set value of 8g/kg' in the case of a 20kW vacuum pump. In the case of the 5kW vacuum pump, the air supply humidity was not dehumidified to the set value but was left to run its course. As shown in Figure-8, it can be confirmed that the room humidity is below 12g/kg' for most of the time with the introduction of the membrane equipment (20kW vacuum pump case). It can also be seen that in the case of the 5 kW vacuum pump. However, the dehumidification performance was not sufficient, a significant reduction in humidity was achieved compared to when the membrane equipment was not introduced.

Figure-9 shows the comparison of indoor temperature and humidity before and after the installation of the membrane. The horizontal axis is the indoor operative temperature, and the figure shows the range[Note2)] that is acceptable for 80% of the occupants during cooling (Clo value 0.5). The percentage of time that the temperature was within the acceptable range was 7.2% and 98.8% for the case without a membrane and with a 20kW pump, respectively, indicating that the introduction of a membrane kept the thermal environment within the acceptable range most of the time. In the case of the 5 kW vacuum pump, 62.5% of the time was within the acceptable range, which was about a percentage 55 point improvement over the time without a membrane.

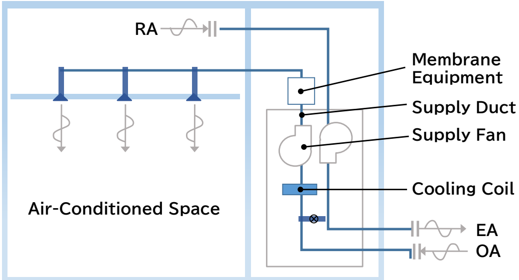

Figure-5 Introduction of membranes in conventional air conditioning systems

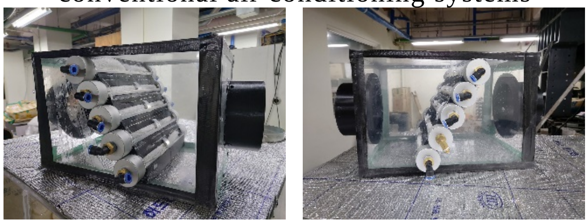

Figure-6 Photograph of an open-type Unit with membrane module[Note3)]

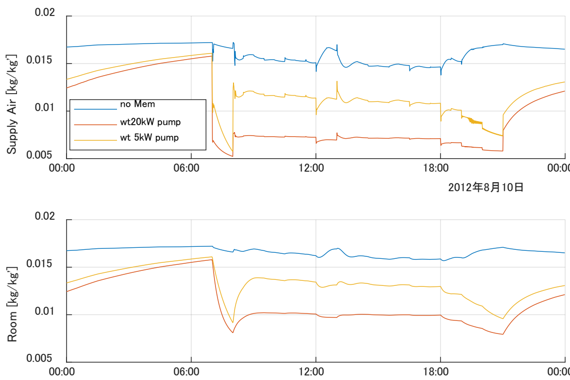

Figure-7 Air supply humidity before and after installation of membrane equipment

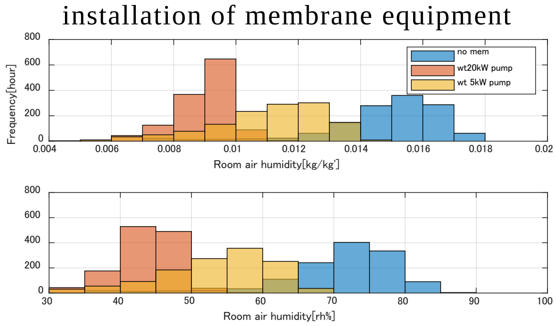

Figure-8 Indoor humidity before and after installation of membrane mechanism

5. Energy Performance of an Air-Conditioning System Using a Membrane System

Table-3 shows the comparison of the system energy performance before and after the installation of the membrane equipment. The total coil heat output for the period was 6.66 MWh for the case with an internal heat gain intensity of 6W/m^2, and the untreated heat was 11.15 MWh. The fan power was also 1.4 MWh for the estimation period. In contrast, the coil heat output of the membrane system (vacuum pump 20 kW case) was 5.98 MWh, and the untreated heat was almost eliminated. The amount of water removed by the membrane equipment was 18,644 kg, and the specific energy consumption which is the vacuum pump power per the removed water volume was 0.5 kWh/kg. In the case of the 5 kW vacuum pump, the untreated heat was 3.26 MWh. This is equivalent to the amount of unprocessed heat rate in the case with 20 W/m^2 of internal heat gain, and the specific energy consumption is 0.43 kWh/kg, which is smaller than the 20 kW case.

6. Summary

Assuming a typical office building, we reproduced the latent heat treatment failure of a conventional air conditioning system due to the reduction of internal heat gain intensity. In addition, the introduction of membrane equipment was assumed, and the effect of its introduction was compared and evaluated in terms of the thermal environment and the energy efficiency. The effect of the introduction of the membrane equipment on eliminating the latent heat treatment failure was confirmed, and the system’s energy efficiency was quantitatively shown. It can be said that it is desirable to consider the introduction of a membrane system based on an appropriate understanding of the trade-off between the installation scale, the effect of eliminating latent heat treatment failure, and the increase in energy consumption.

Note1: Acceptable thermal environmental ranges for general comfort as indicated by ASHRAE Standard.
Note2: Acceptable range of operative temperature and absolute humidity at 80% occupant satisfied

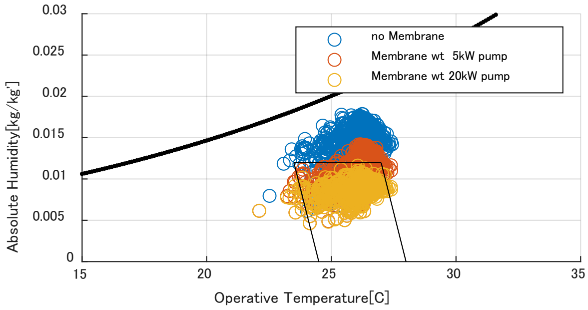

Figure-9 Comparison of indoor temperature and humidity before and after membrane installation

rate (10% global and 10% local of unsatisfied occupant rate) at 0.5 Clo.
Note3: Membrane type: Dense membrane, Membrane selectivity: 1000, Size of membrane module: 360 mm (Length) × 55 mm (Outer diameter), Size of hollow fiber: 300 mm (Effective length) × 400 um (Inner diameter), Number of hollow fibers per a membrane module: 3,800, Effectiveness area: 1.3m^2

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Table-3 Comparison of system energy performance with and without installation of membrane system

	20W/m^2	6W/m^2		
	No membrane		Membrane (20kW)	Membrane (5kW)
Treated heat rate [MWh]	24.69	6.66	5.98	6.01
Untreated heat rate [MWh] excluding overtreated heat	2.95	11.15	0.07	3.26
Water volume removed[kg] (Equivalent latent heat rate MWh)	-	-	18,644 (12.95MWh)	11,400 (7.92MWh)
Supply fan power [MWh]	1.67	1.40	4.22	4.22
Vacuum pump power [MWh]	-	-	11.15	5.91
Specific energy consumption [kWh/kg]			0.50	0.43