DEVELOPMENT OF DESICCANT AIR CONDITIONING SYSTEM USING WAKKANAI SILICEOUS SHALE

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Abstract:
A high efficient air-conditioning system for domestic use is needed to be developed because of the increasing of energy consumption in the residential and commercial sector. In Asia region of high temperature and high humidity, dehumidification requires a large of energy. Desiccant ventilation system is effective to reduce the energy consumption for dehumidification process. We focused Wakkanai siliceous shale (WSS) as a desiccant material, and a desiccant rotor using WSS was developed. The aim of this study is to develop a desiccant ventilation unit using the WSS rotor which can be regenerated by the waste heat from heat pump unit with low-temperature by numerical calculation. As a result, the performance of the dehumidification is predicted by combining a total heat exchange element which has 80 % of latent heat exchange efficiency and 90 % of sensible heat efficiency. This combination is possible to regenerate WSS rotor at low-temperature of 35 – 45^{o}C. In addition, we produced an actual desiccant unit combined with heat pump system, and this system was capable to treat the sensible heat load and latent heat load separately.

Key words: Wakkanai siliceous shale, desiccant ventilation, waste heat utilization

1. INTRODUCTION

In recent years, a high efficient air-conditioning system is needed to be developed because of the increasing of energy consumption in the residential and commercial sector. In Europe, a compact heat pump unit for domestic use which can supply cooling, heating and hot water with waste heat recovering is commonly used. However, compact heat pump is not widely spread in Japan. Moreover, in Asia region where urban population density is high and high temperature and high humidity, dehumidification requires a large energy. Therefore, desiccant ventilation system is expected to reduce the energy consumption. In conventional dew condensing process, the evaporation temperature is usually set at 7ºC, while the evaporation temperature can be increased to 15ºC by treating the sensible heat and latent heat separately. In this way, the energy consumption can be reduced.

To provide original desiccant rotor at a low cost with high performance, we adopted Wakkanai siliceous shale (WSS). The top of Figure 1 shows photographs of WSS and a paper pulp compounded with WSS. WSS is a kind of natural mesoporous material with low prise. It shows high adsorption ability and can be regenerated at a low temperature as 40^{o}C. Because of its low-temperature regeneration, the waste heat released from heat pump is expected to be able to regenerate the WSS desiccant rotor (Nakabayashi et al. 2009).
We developed a paper desiccant honeycomb by adding WSS (Nakabayashi et al. 2011). WSS rotor is shown in Figure1 (left down). Sodium chloride is impregnated into the WSS rotor to increase the amount of dehumidification. Thickness of this rotor is 60 mm, the cell

density is 270 cells per square inch (cpsi). The adsorption isotherm of WSS paper is shown at the lower right of Figure 1. This paper has adsorption characteristics that the adsorption amount increases in a high humidity range (70 %RH), and the maximum water vapor adsorption capacity is 0.4 kg･kg^{-1}.

In this research, we considered the components of the system capable of saving energy while keeping comfortable environment at the same time. Firstly, dehumidifying performance test of the rotor was conducted and a numerical model was created to predict the dehumidification performance and input heat for dehumidification.

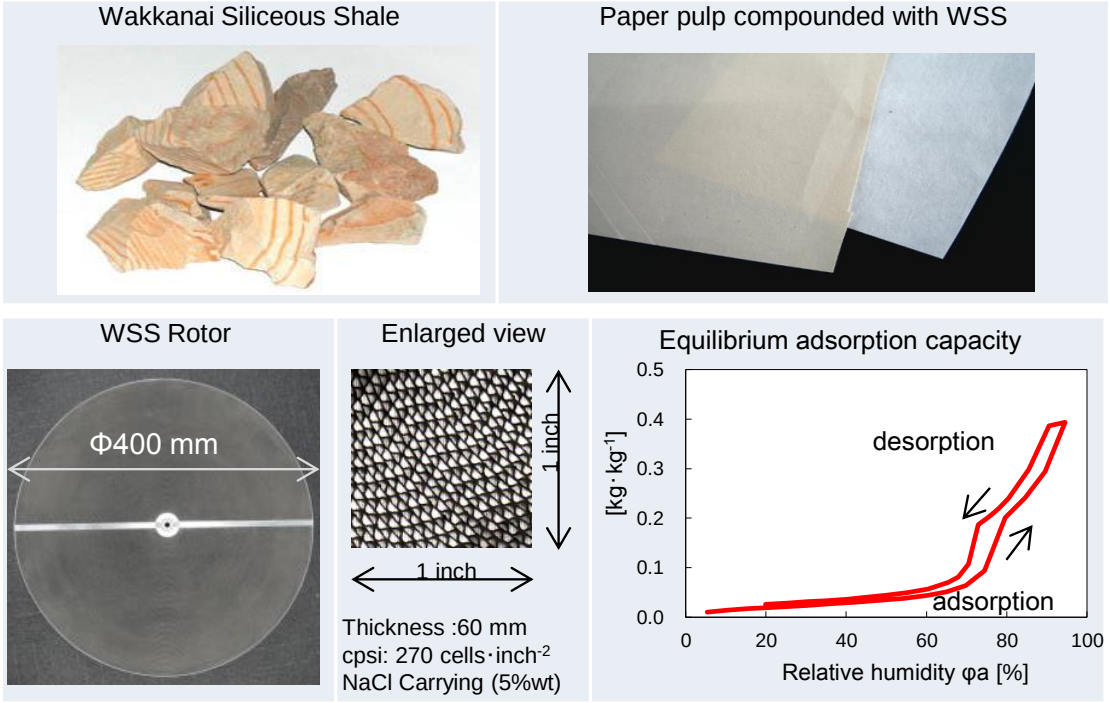


Figure1: Photograph of WSS (left top), Paper Pulp Compounded with WSS (right top), Rotor (left down) and Adsorption Isotherm of WSS Paper (right down)

2. NUMERICAL CALCULARION FOR THE DESICCANT ROTOR

2.1 Numerical Model for the Desiccant Rotor

To predict the dehumidification amount, a numerical calculation program was created. (Nakabayashi et al. 2009) Figure 2 shows a flat plate model of heat and mass transfer in the individual honeycomb of desiccant rotor. The rotor is divided in the circumferential direction, assuming the heat and mass transfer are provided in the individual honeycomb in same time. This flat plate model is comprised the air layer and the absorbent layer. It is divided into a plurality of sections in a flowing direction, and air temperature and absolute humidity are calculated from one-dimensional advection and heat and mass transfer between air and absorbent layers. The gas is assumed to be an incompressible fluid, air and absorbent are distributed at a uniform temperature. The heat balance equations of air layer and absorbent layer are expressed by structural formulas as shown in equation (1) and (2).

$$c_{Pa}\rho_a\omega\frac{\partial T_a}{\partial\theta}=c_{Pa}\rho_a u_a\frac{\partial T_a}{\partial z}+\frac{\alpha_s}{a_a}\left(T_d-T_a\right) \quad (1)$$

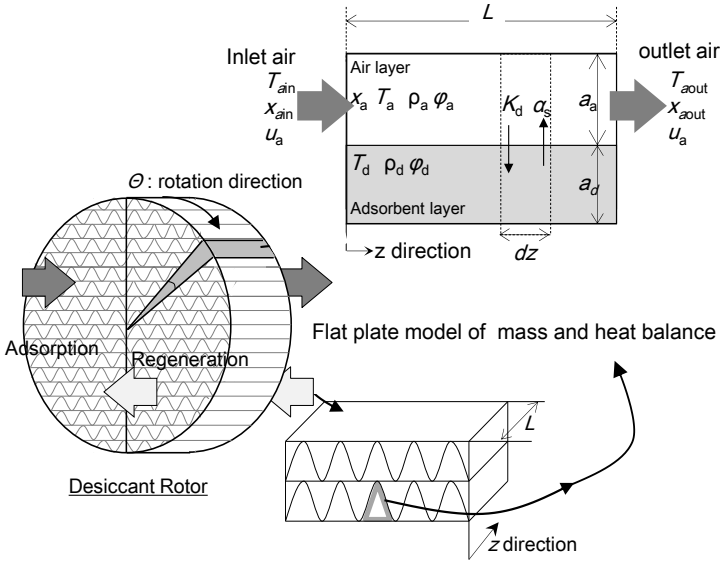
$$c_{Pa}\rho_d\omega\frac{\partial T_d}{\partial\theta}=\rho_d\omega\frac{\partial q}{\partial\theta}Q_h+\frac{\alpha_s}{a_a}\left(T_d-T_a\right) \quad (2)$$

Also, the mass balance equations of air layer and absorbent layer are expressed by structural formulas as shown in equation (3) and (4). In addition, mass-transfer coefficient (K_d) is from the measured value (equation (5)), and heat transfer coefficient (α_S) is convective heat transfer coefficient.

$$\rho_a \omega \frac{\partial x_a}{\partial \theta} = \rho_a u_a \frac{\partial x_a}{\partial z} - \rho_a \omega \frac{a_d}{a_a} \frac{\partial q}{\partial \theta} \tag{3}$$

$$\frac{\partial q}{\partial \theta} = \frac{K_d}{a_d \omega} \left(q_a^* - q_d^* \right) \tag{4}$$

$$K_d = \rho_a u_a a_a (x_a - x_{out}) / \rho_d \Delta y (q_a - q^*) \tag{5}$$



Nomenclature

a	Thickness		m
c_p	Specific heat capacity		kJ·(kg·K)$^{-1}$
ρ	Density		kg · m^{-3}
K_d	Mass-transfer coefficient		m·sec^{-1}
L	Thickness of rotor		m
D	Diameter		m
q	Adsorption amount		g·kg$_{DA}^{-1}$
q^*	Equilibrium amount		g·kg$_{DA}^{-1}$
T	Temperature		°C
t	Time		sec
u	Air flow speed		m·sec^{-1}
x	Absolute humidity		g·kg$_{DA}^{-1}$
z	Direction of rotation axis		m
α_s	Heat transfer coefficient		W·(m^2·K)$^{-1}$
θ	Rotation angle		π
ω	Angular velocity		rad·sec^{-1}

Subscript

a	Air,	d	Desiccant
in	Inlet,	out	Outlet
s	Surface,	ad	Adsorption

Figure 2: Numerical Model of Desiccant Rotor

2.2 Reproductibility of Calculation Results

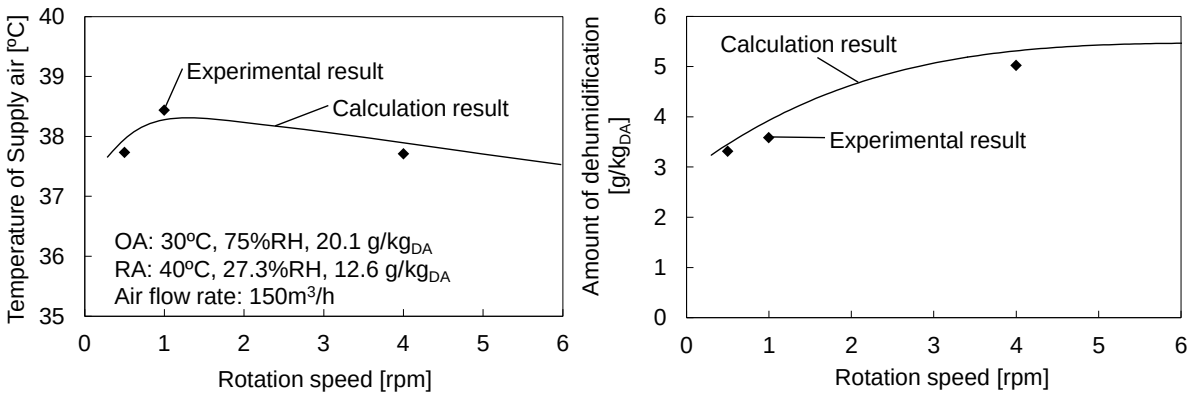
To confirm the reproducibility, we compared with the calculation results and experimental results. Experiment and calculation conditions are shown in Table 1. Return air is 26 °C, 60 %, 12.6 g/kg$_{DA}$, outside air conditon (OA) is 30 °C, 75%, 20.2 g/kg$_{DA}$, and regeneration temperature is 40 °C. To evaluate the change of dehumidification amount, rotation speed is changed from 0.2 to 6 rpm.In this case, we difined a target amount of dehumidification as difference between absolute humidity of outside air and returen air (7.6 g/kg$_{DA}$).

Table1 Experiment and Calculation Conditions

Returen Air condition	Outside Air condition	Regeneration temperatre	Thickness	Diameter	Rotation speed
26°C 60 % (12.6g/kg$_{DA}$)	30°C 75 % (20.2g/kg$_{DA}$)	40°C	60 mm	Φ400 mm	Experiment 0.5, 1.0, 4 rpm Calculation 0.2 – 6 rpm

Comparison between experimental results and calculation results is shown in Figure 3. The solid line shows the calclulation results, and plots show experimental results. We can see the calculated value of dehumidificaiton and temperature of supply air agree with the results of experiment. The amount of dehumidification increases with rising of the rotation speed. The amount of dehumidification reached 5.0 g/kg$_{DA}$ at 4 rpm of rotation speed. However, the maximum amount of dehumidification does not even reach the target value of 7.6 g/kg$_{DA}$, it is

not possible to achieve the target value unless some ingenuity is made. (TOGAWA et al. 2012)



(a)Supply air temperature **(b) Amount of dehumidification**

Figure 3: Comparison between Calculation and Experimental Results.

3. COMBINATION WITH TOTAL HEAT EXCHANGER

3.1 Explanation of the Constitution Elements

To achieve the target amount of dehumidification, we planned a layout of desiccant ventilation unit. The layout of the desiccant unit is shown in Figure 4. This desiccant ventilation unit consists of a static total heat exchange element, a desiccant rotor and a regeneration coil and a precooling coil.

The static total heat exchanger exchanges sensible and latent heat between outside air and return air. The exchanged outside air is cooled to increase relative humidity as 90 % or more by precooling coil without dew condensing. Finally, the precooled air is dehumidified by WSS rotor. Return air is heated by regeneration coil. The WSS rotor shows is excellent water-adsorptive performance in a high-humidity region, and that is why the relative humidity of precooled air should be as high as possible without dew condensing. First of this chapter, to decide the heat exchange efficiency of the element, a performance prediction achieving the target amount of dehumidification was conducted.

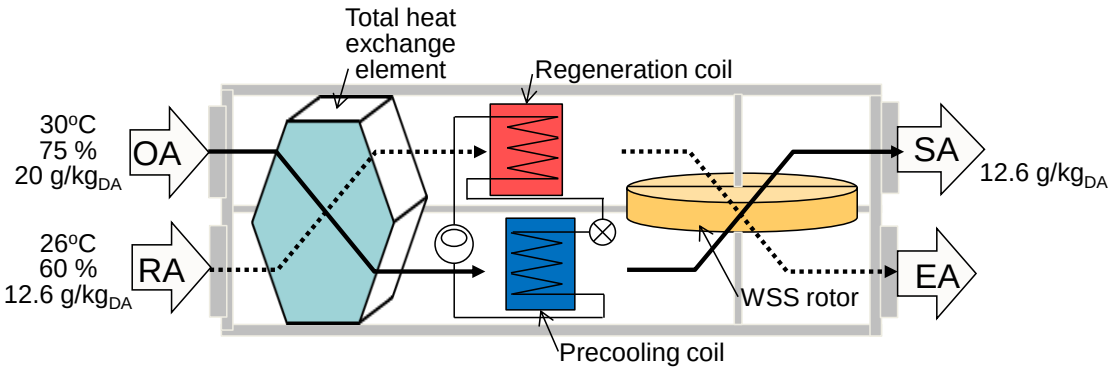


Figure 4 Schematic Diagram of the Desiccant Unit.

3.2 Examination of Heat Exchange Efficiency and Amount of Dehumidification

To decide the heat exchange efficiency, performance prediction was conducted. The parameters of the calculation conditions of the dehumidification performance are shown in Table 2. First, the air volume of general house is 300 m^3, a number of 0.5 of ventilation was mandated, and air flow rate is set at 150 m^3h^{-1}. Outside air temperature is 30 ºC, absolute

humidity of outside air are changed as 15 – 23 g/kg$_{DA}$, and return air condition is 26 ºC, 60 %RH (12.6 g/kg$_{DA}$). Sensible heat exchange efficiency (η_S) and latent heat exchange efficiency (η_L) are calculated by equation (6) and (7).

$$\eta_S = (T_{OA} - T_{SA})/(T_{OA} - T_{RA}) \times 100 \quad (6)$$

$$\eta_L = (x_{OA} - x_{SA})/(x_{OA} - x_{RA}) \times 100 \quad (7)$$

A target absolute humidity of supply air of the unit is set at 12.6 g/kg$_{DA}$. Diameter of WSS rotor is Φ500 mm, thickness of rotor is 60 mm, and rotation speed of desiccant rotors is 0.2 rpm. To increase the amount of dehumidification, diameter was increased. Moreover, rotation speed is decided at 0.2 rpm to prolong the processing time of dehumidification. As we saw in the last section, rotation speed should be increased at the high humidity condition of outside air. However, the first process of this layout is heat exchanging by the total heat exchange element, and more than half of the latent heat load is removed by the element. In this case, low-speed rotation is effective to remove the latent heat load (NABESHIMA et al. 2012). The regenerated air temperature is set at 35 - 50 ºC, and the relative humidity of precooled air is set at 90 %.

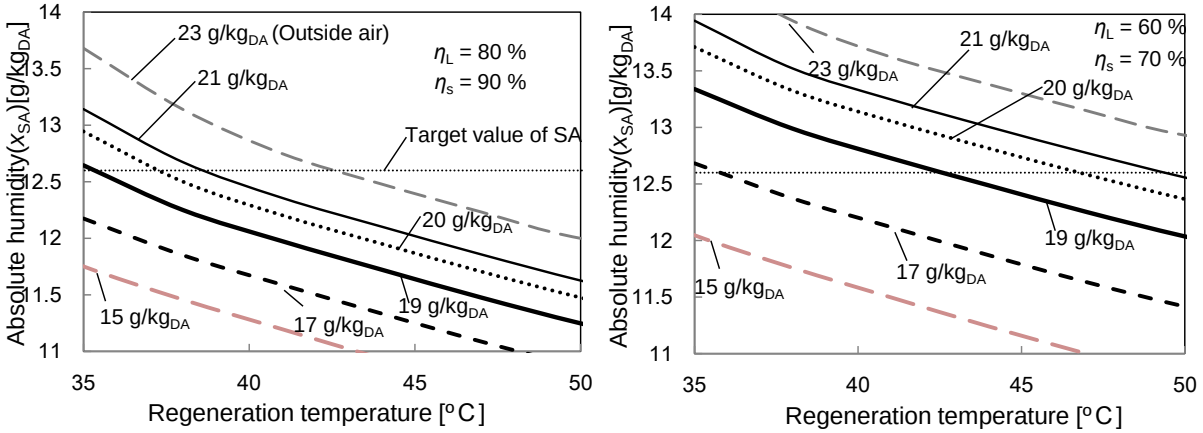
Calculation results are shown in the Figure 5 (a) and (b). Figure 5 shows the relationship between absolute humidity of supply air and regeneration temperature. As shown in Figure 5 (b), when the absolute humidity of outside air is over 21 g/kg$_{DA}$, it cannot reach the target value of supply air (12.6 g/kg$_{DA}$) is not reached. However, as shown in Figure 5(a), the target value of supply air is achieved in all cases. The regeneration temperature is changed from 35 to 45^{o}C. From this result, the target value of the absolute humidity is capable to be achieved at low-temperature regeneration, when the efficiency of total heat exchange element is 80 % of latent heat exchange efficiency and 90 % of sensible heat exchange efficiency.

Table 2: Calculation Conditions of Desiccant Unit

OA: Outside air, RA: Return air

η_S : sensible heat exchange efficiency, η_L :latent heat exchange efficiency

Air flow rate	Specifications of rotor	Regenerated air temperature	RH of Precooling	Total heat exchanger η_S / η_L	RA condition	Absolute humidity OA(x_{OA})
150 m^{3}h^{-1}	Diameter Φ500 mm Thickness 60 mm Rotation speed 0.2 rpm	35 ºC – 50 ºC	90 %	70 % / 60 % 80 % / 70 % 90 % / 80 %	26 ºC 60 % 12.6 g/kg$_{DA}$	15 g/kg$_{DA}$ 17 g/kg$_{DA}$ 19 g/kg$_{DA}$ 20 g/kg$_{DA}$ 21 g/kg$_{DA}$ 23 g/kg$_{DA}$



(a) η_S = 90 % / η_L = 80 % **(b) η_S = 70 % / η_L = 60 %**

Figure 5: Relationship between x_{SA} and Regeneration Temperature

Figure 6 shows the regeneration temperature change when the absolute humidity of supply air is achieve the target value. We can see the regeneration temperatures to achieve the target amount of supply air. When the absolute humidity of outside air is increased, the regeneration temperature is increased. Moreover, the efficiency of total heat exchange element is higher, the regeneration temperature is decreased. In case of 80 % of latent heat exchange efficiency and 90 % of latent heat exchange efficiency, regeneration temperature is capable to hold at low temperature, the highest value of regeneration is 42 ^{o}C at 23 g/kg$_{DA}$. From this result, it is clear that when the heat exchanger having 80 % of latent heat exchange efficiency is installed, the low-temperature regeneration become possible.

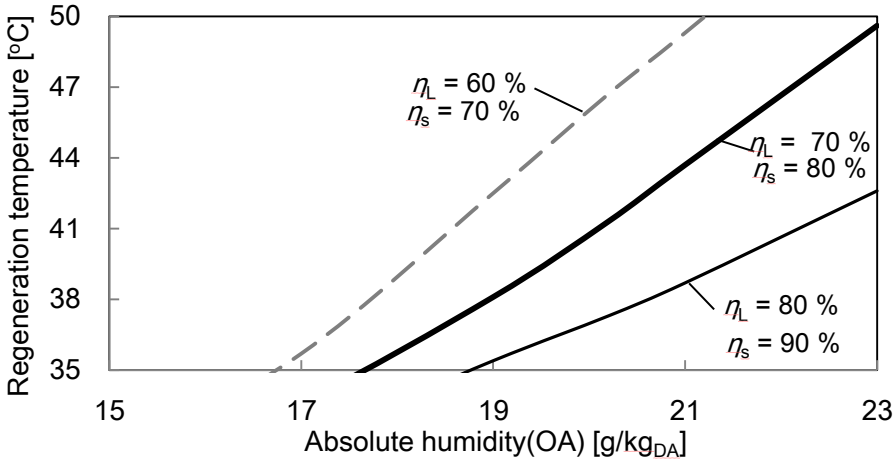


Figure 6: Relationship between Heat Exchange Efficiency of the Element and Regeneration Temperature. (x_{SA} : Absolute Humidity of Supply Air)

3.3 Performance Prediction of the Desiccant Ventilation Unit

In the previous section, the desiccant unit was confirmed to be able to remove the latent heat load at a low temperature which means that the desiccant unit is capable to treat the sensible heat load and latent heat load separately. And in this section, we evaluated the dehumidification performance prediction of the desiccant unit. Calculation conditions are shown in Table 3. Latent heat exchange efficiency is 80 %, and sensible heat exchange efficiency is 90 %. The period of time for the performance prediction is four months from Jun to September. The weather data in Tokyo is from the AMEDAS information.

Table 3: Calculation Conditions of Performance Prediction

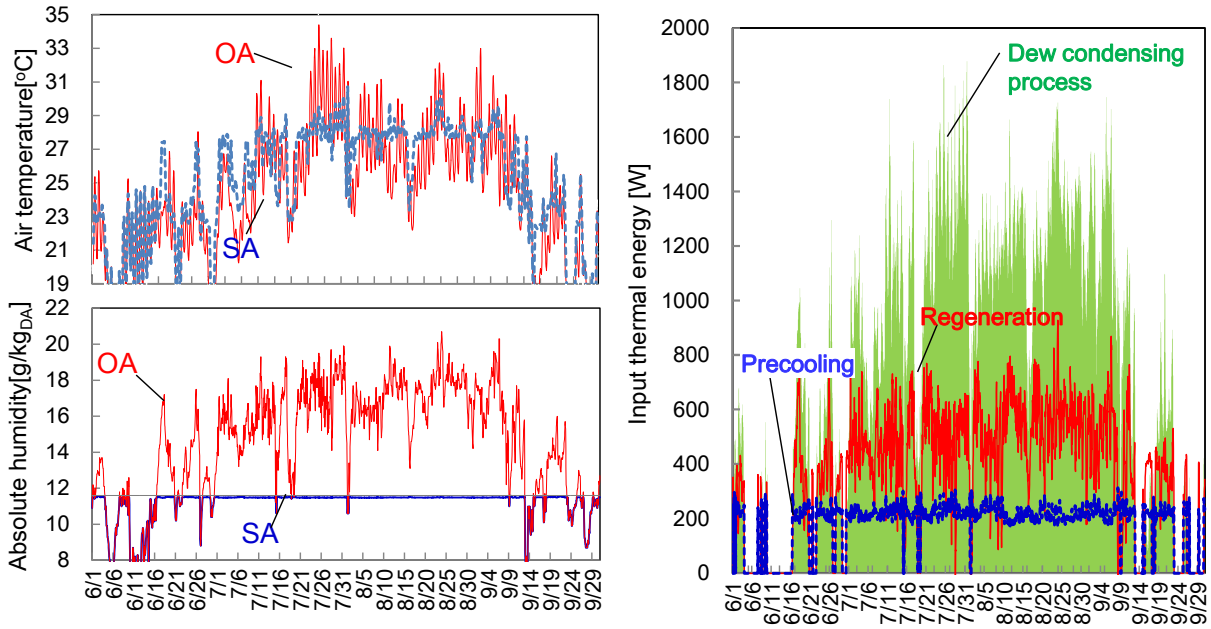
OA: Outside air, RA: Return air

η_S : sensible heat exchange efficiency, η_L :latent heat exchange efficiency

Air flow rate	Specifications of rotor	Total heat exchanger η_S / η_L	RA condition	RH of Precooling	Period of time	Weather data
150 m^{3}h^{-1}	Diameter × Thickness Φ500 × 60 mm Rotation speed 0.2 rpm	90 % / 80 %	26 ºC 60 % 12.6 g/kg$_{DA}$	90 %	June to September	AMEDAS Tokyo (Japan)

Calculation results are shown in Figure 7. Upper figure shows the air temperature changes of outside air (OA) and supply air (SA). The maximum temperature of OA shows 34^{o}C, temperature of SA reaches 31^{o}C. The graph in the center shows the change of absolute humidity of OA and SA. We can see the absolute humidity of SA can be kept at the target value of 12.6 g/kg$_{DA}$. The lower figure shows the Input thermal energy for precooling and regeneration and removed latent heat. Input heat of precooling shows 250 W, and input heat of regeneration changes from 500 to 800 W. Also, the bar shows the input heat of dew

condensing process. These values are enthalpy difference between outside air and dew point of 12.6 g/kg$_{DA}$. The maximum amount of enthalpy difference is 1500 W, and the precooling heat amount is lower than the amount of input heat of dew condensing process. Also, the input cold energy of dehumidification is reduced by 29 – 38 % compared with the dew condensing process. From this result, this layout is effective to reduce the input heat for dehumidification when the waste heat from heat pump is available.



(a) Change of Air Conditions **(b) Input Thermal Energy**

Figure 7: Calculation Results of Performance Prediction.

Table 4 : The Reduction Rate of Amount of Removed Heat for Cooling (The Accumulated Energy)

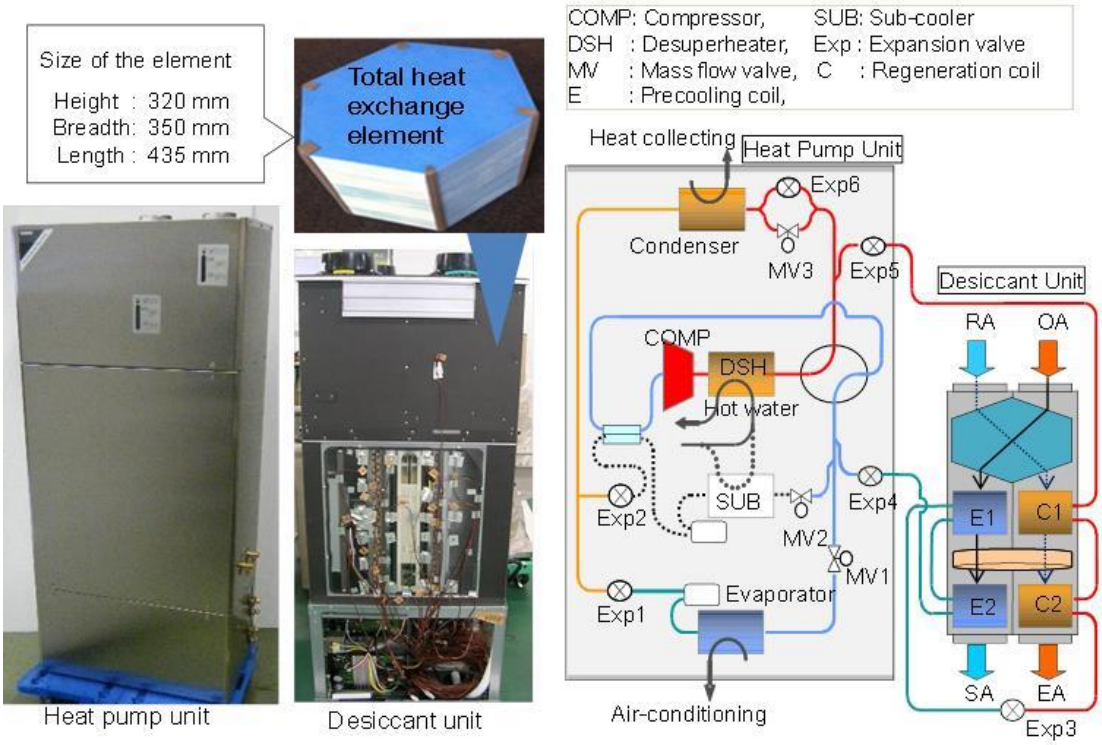
	June	July	August	September
Dew condensing	124 kWh	766 kWh	874 kWh	364 kWh
Precooling	88 kWh	494 kWh	540 kWh	242 kWh
Input energy reduction rate	29 %	36 %	38 %	33 %

4. PERFORMANCE TEST OF AN INTEGRATED SYSTEM

4.1 Outline of the Integrated System

Lastly, we developed an integrated desiccant unit combined with heat pump system. Photograph and schematic diagram of the integrated system is shown in Figure 8. This integrated system consists of ground source heat pump unit and desiccant ventilation unit. The heat pump can supply cooling, heating and hot water. The desiccant unit is used to remove the latent heat. Total volume of this unit is less than 1.0 m^3. According to the schematic diagram of the developed integrated system, in which the refrigerant circuit is installed. The high pressure refrigerant coming out of the compressor condenses in the desuperheater and hot water with a temperature of 63-65ºC can be obtained. After that part of the refrigerant condenses in the condenser, continues to flow through the evaporator and gets back to the compressor. The other refrigerant is used to regenerate the desiccant rotor (through expansion valve 5, C1, C2) using the residual condensation heat and changes to be low pressure liquid by getting through expansion valve 3 and then evaporates to precool the air in the desiccant unit.

The cooling and dehumidification experiment of the integrated system is conducted to evaluate the separating treatment of the sensible and latent heat. In this case, latent heat load from outdoor is removed by desiccant unit, and sensible heat is removed by heat pump unit. At first, the air flow rate is set at 150 m^3h^{-1}, outside air condition is 32.6 ºC, 68 %RH (20.7 g/kg_{DA}), and return air condition is 26 ºC, 64 %RH (13.3 g/kg_{DA}). A target amount of dehumidification is set as 7.6 g/kg_{DA}. Also, rotation speed of desiccant rotors is 0.2 rpm. The regenerated air temperature is controlled at 38 ºC, and the precooled air temperature is controlled at 22.0 ºC. The set cooling temperature is 15 ºC. Inlet water temperature of heat source (primary side) was controlled at a constant temperature of 20 ºC. The opening degree of the expansion valve 1 was controlled by a controller to make sure the maximum cooling.



(a) Photograph of the Unit **(b) Schematic Diagram of the Integrated Unit**

Figure 8: Integrated Heat Pump Unit.

4.2 Experimental Results of Air-Conditioning and Dehumidification

Figure 9 is a psychrometric chart of the dehumidification process in the desiccant unit. At first, outside air is exchanged the sensible and latent heat with return air by the total heat exchange element. Absolute humidity of outside air is reduced to 15.2 g/kg_{DA}, also, the latent heat exchange efficiency shows 73.4 %, and sensible heat exchange efficiency shows 76 %. In this case of rotor being regenerated at 38 ºC, the dew condensing caused by precooling is prevented, although relative humidity of No.2 is increased in the precooling process. The supply air condition is 27.0 ºC, 58 %RH, and absolute humidity as 12.6 g/kg_{DA}. The amount of supply air can be achieved the target value of supply air. From these results, it can be understood that the ventilation unit has a competent performance as a desiccant ventilation unit without dew condensing. It is also confirmed that the desiccant unit is capable to treat the sensible heat load and latent heat load separately.

The performances of the cooling and dehumidifying operation are shown in Table 5. The upper part of Table 5(a), the heat amount of cooling is 2.76 kW, and energy consumption is 1.19 kW. Table 5(b) shows the performance of the desiccant unit, within which the removed heat amount of desiccant unit is 1.25 kW. COP is shown in Table 5(c). Here, a COP* is defined as the performance coefficient of the cooling and dehumidification operation. Q_C is

the cooling load, Q_D is a removed heat of desiccant unit, and W is an electricity consumption of compression machine. COP* shows the total performance of the heat pump and desiccant unit. Cooling COP is 2.31, and COP* is 3.36.

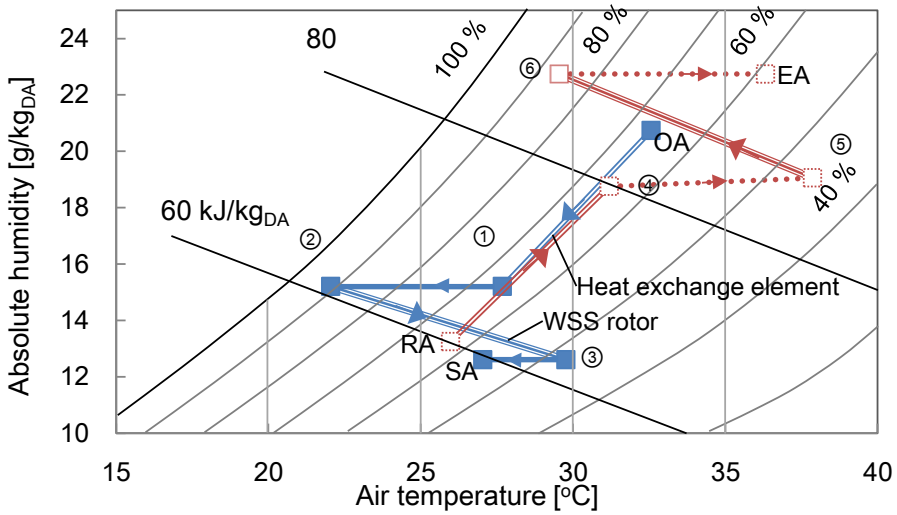


Figure 9: Changes of air temperature and absolute humidity

Table 5 Result of a Performance Evaluation

(a) Cooling performance of heat pump

Heat pump unit	Supply	Return	Water flow rate	Heat amount
Heat collecting	19.4 °C	17.6 °C	23.8 ℓ/min	2.87 kW
Cooling (Q_C)	14.6 °C	17.1 °C	15.5 ℓ/min	2.76 kW
Electricity consumption(W)	-	-	-	1.19 kW

(b) Performance of desiccant unit

Desiccant unit	Outside air	Return air	Supply air	
Removed heat amount(Q_D)	32.6 °C 20.74 g/kg$_{DA}$	26.0°C 13.3 g/kg$_{DA}$	27.0 °C 12.6 g/kg$_{DA}$	1.25 kW

(c) COP of the integrated unit

COP		[-]
Cooling COP	COP = Q_C / W	2.31
COP*:Cooling and Desiccant	COP* = (Q_C+Q_D) / W	3.36

5. CONCLUSIONS

To develop a desiccant ventilation system that is effective to perform the separately treatment of sensible heat and latent heat, we conducted a performance prediction to examine the dehumidification performance of the desiccant ventilation unit using WSS rotors.

1. Calculation program of the WSS rotor was created, and the agreement of the program was confirmed. However, the maximum amount of dehumidification did not even reach the target value of 7.6 g/kg$_{DA}$, it is not possible to achieve the target value unless some ingenuity is made.

2. We planned a layout that consists of a static total heat exchange element, desiccant rotor and regeneration coil and precooling coil. Moreover, to decide the required efficiency of the total heat exchange element, we conducted a system simulation of the desiccant system. As a result, when the heat exchanger with 80 % of latent heat exchange efficiency is installed, the target dehumidification amount can be achieved with low-temperature regeneration. In addition, this layout can reduce the input cold energy by 29 – 38 % compared with the dew condensing process.

3. We developed an integrated desiccant unit combined with heat pump system. This integrated system consists of a ground source heat pump unit and a desiccant ventilation unit. The heat pump can supply cooling, heating and hot water. The desiccant unit is used to remove the latent heat. Total volume of this unit is less than 1.0 m^3.
The performance of the desiccant unit can achieve the target value of absolute humidity of 12.6 g/kg_{DA}. It is also confirmed that the desiccant unit is capable to treat the sensible heat load and latent heat load separately.
In addition, cooling and dehumidification performance test was conducted. As a result, total performance of the system was 3.95, and COP* was 3.36.

In conclusion, the developed integrated unit was capable to perform a cooling and dehumidification operation by separating sensible heat treatment and latent heat treatment without dew condensing. When the latent heat load is increased too much, the low-temperature regeneration of this desiccant system can be achieved.

6. REFERENCES

A. Kodama, T. Hirayama, M. Goto, T. Hirose and R. E. Critoph, (2001). “The use of psychrometric charts for the optimization of a thermal swing desiccant wheel” Applied Thermal Engineering, JSRAE volume 21 No.16 page1657

H. Inaba, T. Kida, A. Horibe and M. Kaneda, (2002) “Sorption Characteristics of Honeycomb-Type Sorption Element Composed of Organic Sorbent”, JSME International Journal Series B 45 No1 page183

J. Togawa , A. Kurokawa, S. Nakabayashi, K. Nagano, S. Yoshida, (2009) “Development of Desiccant System using Wakkanai Siliceous Shale—1st Report: Development and Evaluation of a Paper-Base for the Desiccant Rotor—” JSRAE volume 26 No. 4 page 543

J. Togawa , Y. Nabeshima, K. Nagano, (2012) “Development of Desiccant System using Wakkanai Siliceous Shale —3rd Report: Evaluation of the Dehumidification Performance of Desiccant Rotor—” JSRAE volume 29 No.1 page163

S. Nakabayashi, K. Nagano, M. Nakamura, J. Togawa , A. Kurokawa, (2009) “Development of Desiccant System using Wakkanai Siliceous Shale—2nd Report: Development, Evaluation and Numerical Simulation of Rotor—” JSRAE volume 26 No.4 page 551

S. Nakabayashi, K. Nagano, M. Nakamura, J. Togawa , A. Kurokawa, (2009) “Evaluation of the Water Adsorption Performance of Wakkanai Siliceous Shale Supported by Chlorides” KAGAKU KOGAKU RONBUNSHU Volume 35 No.6 page 602

S. Nakabayashi, K. Nagano, M. Nakamura, J. Togawa , A. Kurokawa, (2011) “Improvement of water vapor adsorption ability of natural mesoporous material by impregnating with chloride salts for development of a new desiccant filter” Adsorption Volume 17 page 602

Y. Nabeshima, K. Nagano, M. Nakamura, J. Togawa, (2012) “Development of Desiccant System using Wakkanai Siliceous Shale, —4th Report: Optimization of the Operational Conditions for the Desiccant System—”: JSRAE volume 29 No.1 page171

Y. Nabeshima, K. Nagano, J. Togawa, (2012) “Development of Desiccant System using Wakkanai Siliceous Shale—5th Report: Examination for the Form of Desiccant Rotor—”, JSRAE volume 29 No.1 page 179