15TH IEA HEAT PUMP CONFERENCE
MAY 26–29, 2026 | HOFBURG
VIENNA 2026
DECARBONISATION THROUGH INNOVATION

# Development of a digital twin for a liquid desiccant system with heat pump integration

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**Abstract**

Efficient humidity management has become a critical demand in modern HVAC systems due to rising outdoor humidity associated with global warming and the rapid growth of moisture-sensitive industries such as semiconductor and battery manufacturing. Liquid desiccant (LD) air-conditioning systems offer a promising solution by directly absorbing water vapor from ventilation air using a hygroscopic solution, thereby handling latent loads without the energy-intensive over-cooling and reheating processes required in conventional systems. As moisture absorption dilutes the solution, stable operation requires simultaneous cooling of the absorber and heating of the regenerator, making heat-pump integration highly effective. However, optimizing LD–heat-pump assemblies at the system level remains challenging because most prior studies address isolated components rather than the coupled thermal interactions that govern real performance. Digital twin technology provides a pathway to overcome this gap by enabling virtual replication of the physical system for supervisory control design, fault detection, and long-term performance evaluation. In this work, we develop a detailed digital twin of an LD air-handling unit equipped with two heat pumps and a cross-flow liquid desiccant contactor. The model is constructed using the Modelica language. Experimental data are obtained from an installed classroom ventilation unit featuring a 7.1 kW chilled-water heat pump and a 4.5 kW hot-water heat pump. The resulting digital twin platform enables fast scenario testing, energy-management optimization, and exploration of advanced control strategies that can further reduce carbon dioxide emissions in buildings and industrial facilities.



*Keywords: Indoor air quality (IAQ); Dehumidification; Liquid desiccant system; Digital twin*

## 1. Introduction

Efficient technologies for controlling indoor humidity are becoming increasingly indispensable. In recent years, the importance of indoor air quality (IAQ) has been widely emphasized. Particular attention has been paid to its role in supporting human wellness, including comfort and health in everyday living and working environments. At the same time, it is increasingly recognized as a key factor in maintaining and improving productivity [1-2]. Among the various factors that define indoor air quality, humidity is a particularly critical parameter especially in countries with humid climates such as Southeast Asia and East Asia. In parallel, the development of electronics, information technology, and semiconductor industries has created more situations in which strict humidity management is required, such as in precision equipment manufacturing [3]. Consequently, humidity-control technologies are in demand in both residential and industrial environments. One technology that can address these requirements is the liquid desiccant system with heat pump integration.

In a liquid desiccant air-conditioning system, a liquid desiccant is employed to dehumidify and humidify air through absorption. The heat pump unit is used to produce cooling water that removes the heat generated during the moisture absorption process. The system is mainly implemented as an outdoor air handling unit for ventilation and is operated to control both indoor humidity and $CO_2$ concentration. One major advantage of this configuration lies in the way the liquid desiccant is regenerated. In this process, the solution is heated so that it releases the absorbed water. This regeneration can be achieved at relatively low temperatures compared with solid desiccant and other systems. As a result, it becomes possible to utilize otherwise unused heat sources, such as exhaust heat from the heat pump unit and waste heat from industrial facilities [4]. Therefore, liquid desiccant systems have attracted attention as an efficient humidity-control system.

A digital twin is a virtual representation of physical equipment and spaces, reproduced in a simulation environment. It is designed to perform real-time simulations with time steps on the order of 1 to 15 minutes. By computing physical quantities that are not directly measured in the real world, the digital twin enables detailed visualization of the system state, which is useful for exploring more efficient operating strategies and for detecting fault conditions. In addition, by incorporating weather data and information on space utilization, the digital twin can predict future operating conditions and support more energy-efficient control of equipment. Because power consumption in HVAC systems is highly sensitive to environmental variations, digital twins are regarded as a key technology in this field, and methods for integrating them into various types of air-conditioning equipment are being actively investigated and evaluated [5].

With respect to the analysis of liquid desiccant systems, both numerical simulations and experiments have predominantly focused on the gas–liquid contactor, where the air stream directly interacts with the absorption solution. This component is regarded as critical because it is the location where complex transport phenomena occur [6-8]. By contrast, system-level analyses that encompass peripheral equipment, such as heat pumps, have been reported for hybrid liquid desiccant air-conditioning systems [9-10]. However, these studies are not primarily aimed at real-time analysis, which is a fundamental requirement for digital twin applications. To construct a digital twin suitable for real-time simulation and supervisory control, it is necessary to develop mathematical models of air-conditioning equipment that appropriately balance physical fidelity and computational efficiency, and to determine the key parameters that should be retained in the models. In this study, mathematical models are developed for the liquid desiccant air-conditioning system and its peripheral equipment, and a digital twin is constructed based on an air-conditioning system installed on our university campus. The present analysis focuses on the process side of the liquid desiccant unit only, which directly handles the dehumidification of outdoor air, while the regeneration side is not explicitly modeled. Furthermore, the operating conditions considered in this study are representative of the summer season, during which dehumidification demand is dominant. Under these assumptions, the impact of introducing the liquid desiccant unit on the supply-air conditions, the total power consumption of the air-conditioning system, and the latent load handled by the indoor air-conditioning unit is analyzed.

## 2. Approach

In this study, mathematical models for each piece of equipment are first developed and a simulation platform is implemented to analyze system operation. Mathematical models are formulated for the equipment, their constituent elements, and the conditioned spaces, thereby providing a numerical simulation environment. To enable flexible construction of a digital twin even when the configuration of the target HVAC system is modified in the future, a modular modeling approach is adopted. In this approach, each constituent element is regarded as a module characterized by input and output variables. The objective is to reproduce the internal physical phenomena of each device and to clarify the relationships between manipulated and controlled variables. The models are implemented in the Modelica language, which provides an equation-based, object-oriented, physics-based modeling framework widely used in one-dimensional computer-aided engineering (1DCAE), enabling modular system representation and consistent formulation of coupled thermal and mass-transfer phenomena for digital twin applications.

On our university campus, one classroom is equipped with an air-conditioning system in which a liquid desiccant unit is installed as an outdoor air handling unit. A digital twin of this system is constructed, and the details of the system configuration are presented in Section 3. Using the developed digital-twin-based simulator, the dependence of the liquid desiccant air-conditioning unit performance on its operating variables is analyzed, and the impact of introducing the liquid desiccant unit is evaluated.

## 3. System Description

The air-conditioning system installed on the university campus incorporates a liquid desiccant unit as an outdoor air handling unit. The liquid desiccant unit conditions outdoor air before it is supplied to the room, thereby maintaining indoor humidity and $CO_2$ concentration within specified ranges. A packaged air conditioner installed in the room controls the indoor air temperature. The indoor air pressure is balanced with the ambient atmospheric pressure through mechanical exhaust and natural ventilation. The overall configuration is shown in Fig. 1 [9].

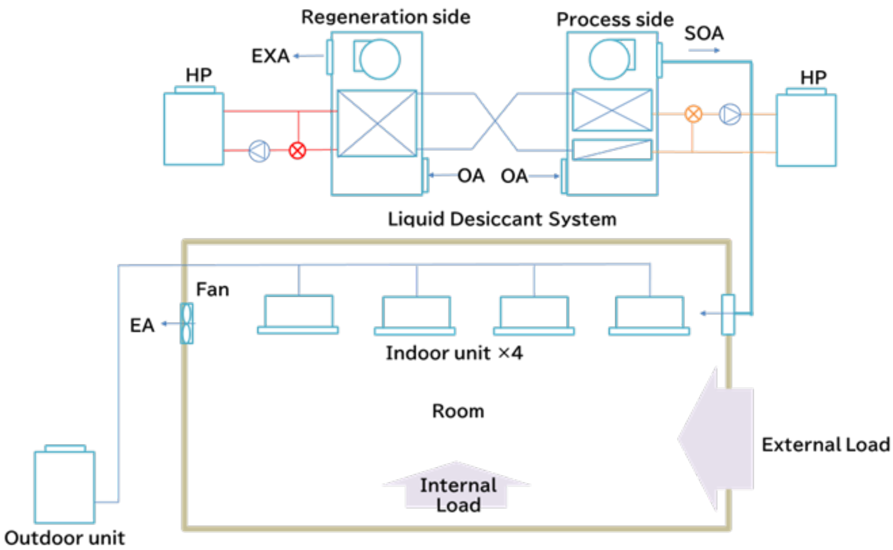


Figure 1. Schematic diagram of the liquid desiccant air conditioning system on the Waseda University campus [11].

The liquid desiccant system is divided into two main subsystems: a process side, which adjusts the humidity of the outdoor air, and a regeneration side, which removes water from the absorption solution. On the process side, cooling is required to suppress the temperature rise of the absorption solution caused by moisture absorption. Therefore, a dedicated chilled-water heat pump (7.1 kW) is installed to supply cooling water for temperature control of both the solution and the air stream. On the regeneration side, heating is required to increase the solution temperature for desorption. In practical applications, this thermal demand is expected to be covered by low-grade unused heat sources, such as waste heat from heat pumps or industrial equipment. However, since such heat sources were not available at the university campus where the system was installed, a dedicated hot-water heat pump (4.5 kW) was introduced to supply regeneration heat. It should be noted that the chilled-water and hot-water heat pumps operate independently and are not configured as a single reversible heat pump providing simultaneous heating and cooling. In addition to the liquid desiccant outdoor air handling unit, a packaged air-conditioning unit is installed in the room to control the indoor air temperature. The actual system configuration is shown in Fig. 2.

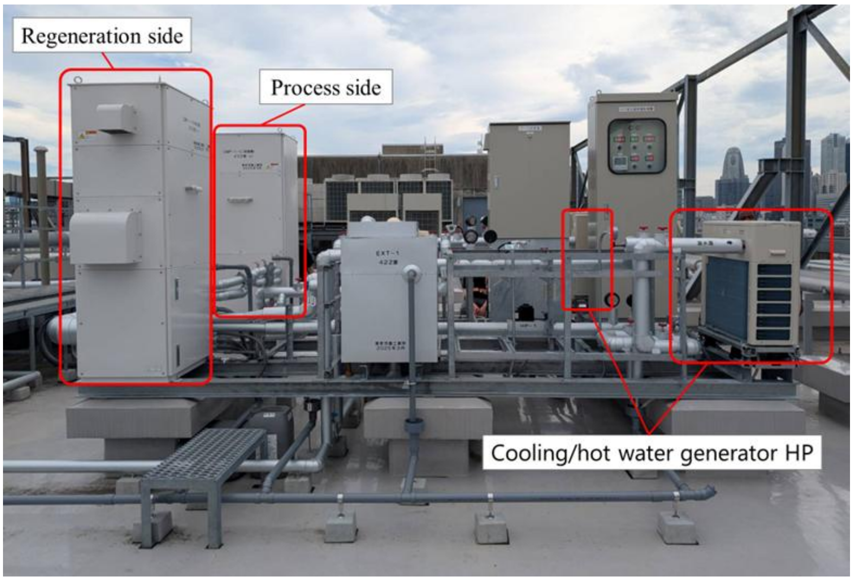


Figure 2. Liquid desiccant system installed in Waseda University campus.

The analysis focused on dehumidification operation, and the following description therefore concerns the flow paths within the process side during dehumidification mode. The process side consists of three main components: an air–solution contactor, an air–water heat-exchange coil, and a solution–water heat exchanger. Air, absorption solution, and cooling water are driven by fans and pumps through these components, where coupled heat and mass transfer takes place. The air stream is first pre-cooled by the cooling water to suppress the temperature rise of the solution in the contactor. It then comes into contact with the solution, and water vapor is absorbed into the solution, driven by the vapor pressure difference between the air and the solution. Before contacting the air, the absorption solution is likewise cooled by the cooling water to reduce its vapor pressure. The solution is stored in a tank, from which a certain fraction is sent to the regeneration side and regenerated solution is returned from the regenerator. The corresponding flow paths are illustrated in Fig. 3. The symbols used in Section 4 are also indicated in the figure.

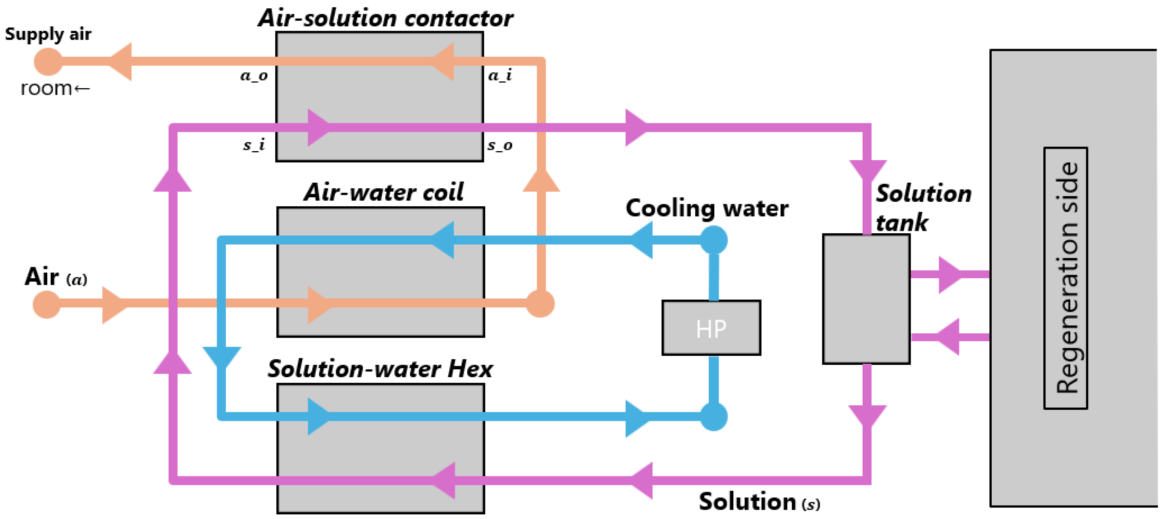


Figure 3. Schematic diagram of flow paths in the process side of the liquid desiccant system.

## 4. Modeling

This section presents the mathematical models developed for each component: the liquid desiccant system (air–solution contactor, air–water heat-exchange coil, and solution–water heat exchanger), the room, and the heat pump (HP). These models are implemented in the Modelica language and executed on the Dymola software platform for numerical simulation.

### *4.1. Liquid desiccant system model*

Mathematical models were developed for each component on the process side. For the elements involving the air stream, in which simultaneous heat and moisture transfer occur (air–solution contactor and air–water heat-exchange coil), a consistent modeling framework was adopted in which the exchanged heat and mass are evaluated using heat transfer and mass transfer coefficients. Counter-flow, two-dimensional exchange between the fluids was assumed. To maintain numerical accuracy, the flow direction was discretized into a series of control volumes, and the dependence of the calculated results on the number of segments was examined. Pressure loss along the path was disregarded.

#### *4.1.1. Air-solution contactor model*

For each control volume, the governing equations were formulated as follows. The overall air and water-vapor mass balances, together with the energy and momentum balances, are given by Eq. (1-4).

$$G_{a\_i} = G_{a\_o} + J_{cond} \tag{1}$$

$$G_{a\_i} x_{a\_i} / (1 + x_{a\_i}) = G_{a\_o} x_{a\_o} / (1 + x_{a\_o}) + J_{cond} \tag{2}$$

$$G_{a\_i} h_{a\_i} = G_{a\_o} h_{a\_o} + Q_s + J_{cond} h_v \tag{3}$$

$$P_{a\_i} = P_{a\_o} \tag{4}$$

On the solution side, the corresponding mass, energy, and momentum balance equations are expressed in Eq. (5-7). Condensed vapor goes into solution.

$$G_{s\_i} = G_{s\_o} - J_{cond} \tag{5}$$

$$G_{s\_i}h_{s\_i} = G_{s\_o}h_{s\_o} - Q_s - J_{cond}(h_v - h_{w,s}) - J_{cond}h_{w,s} \tag{6}$$

$$P_{s\_i} = P_{s\_o} \tag{7}$$

The amount of heat and moisture transfer are determined using the inlet temperature difference and the inlet absolute humidity difference as the respective driving forces like in Eq. (8-9). The absolute humidity at the gas–liquid interface can be calculated using the vapor pressure of water in the solution, which is evaluated using an approximate correlation fitted to experimental data. It was assumed that the entire surface of the contactor was completely wetted by the solution.

$$Q_s = h_h A(T_{a\_i} - T_{s\_i}) \tag{8}$$

$$J_{cond} = h_m A(x_{a\_i} - x_{surf\_i}) \tag{9}$$

*4.1.2. Air-water coil model*

The mathematical model for the air–water heat-exchange coil is essentially identical to that for the air–solution contactor. In the coil, the air and water streams are separated by a metal wall, and any water vapor that condenses from the air stream is discharged from the system as drain water. Unlike in the absorption process in the gas–liquid contactor, the influence of variations in the humidity field at the interface is considered small. Therefore, Lewis's analogy is assumed, and the mass transfer coefficient is derived from the heat transfer coefficient.

*4.1.3. Solution-water heat exchanger model*

In this component, only heat transfer happens. The mass and momentum of each fluid are assumed to remain unchanged, and the exchanged heat is calculated using a heat transfer coefficient. For model simplification, an additional formulation based on temperature effectiveness was also developed.

*4.1.4. Other component models*

Other components include the solution and water pumps and the regeneration unit. Each pump is configured to deliver a constant flow rate while the system is operating. Therefore, only the rated electrical power consumption of the pumps is taken into account separately. For the regeneration side, electrical energy consumption is estimated under the assumption that it processes only the thermal energy stored in the solution on the process side.

*4.2. Heat pump model*

In the air-conditioning system, the heat pumps are used both as chilled/hot water generators for controlling the temperature of the liquid desiccant solution and as an indoor air-conditioning unit. The heat pump model is based on a refrigeration-cycle formulation using the pinch temperature concept. The refrigeration cycle is determined by specifying the pinch temperature differences between the refrigerant and the secondary fluids in the condenser and the evaporator, together with the refrigerant superheat, subcooling, and the isentropic efficiency of the compressor. In the present analysis, these parameters were assigned using fitted values obtained from a different air-conditioning unit that had been previously investigated by the authors' research team.

*4.3. Room model*

The room model was formulated using a lumped-parameter approach. As thermal loads in the room, sensible heat generation from occupants and latent heat associated with perspiration are considered as internal loads, together with equipment heat gains from devices such as personal computers and light bulbs. In addition, sensible heat transfer through the building envelope from the outdoor environment is treated as an external load. These loads were aggregated into sensible and latent components, respectively, and configured to be supplied as inputs to the room model. To account for the thermal capacitance of indoor furnishings, a constant factor of 10.5, adopted from values reported in the literature, was applied as a multiplier to the specific heat capacity of air[12].

*4.4. Grid-independence analysis*

The dependence of the calculation results on the number of control volumes, N, was examined for both the air–solution contactor and the air–water heat-exchange coil. For each component, the number of control volumes was gradually increased, while the inlet conditions at a representative operating point were applied, and the resulting changes in the computed outlet states were observed. The results for the air–water heat-exchange coil, in particular, are shown in Fig. 4.

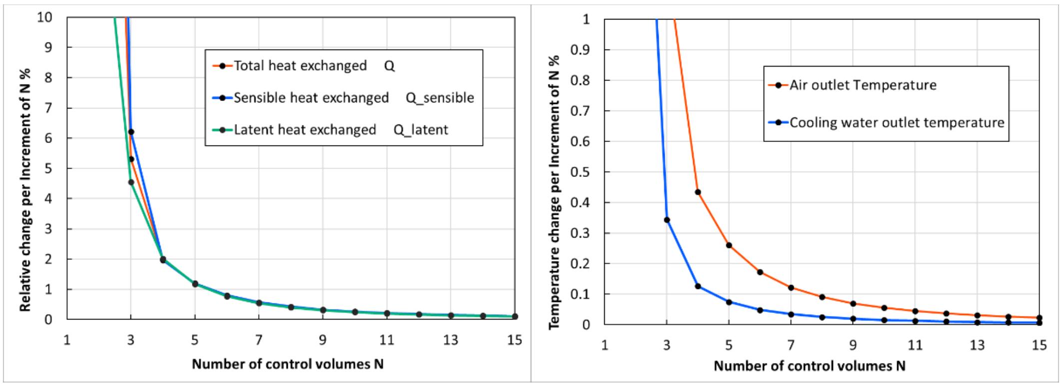


Figure 4. Analysis on the dependence of the calculation results on the number of control volumes.

The variations in heat-transfer rate and outlet temperatures were examined with increasing numbers of control volumes. For N ≥ 12, the relative change in the heat-transfer rate fell below 0.2%. For N ≥ 11, the change in the outlet temperatures was less than 0.05 K. The subsequent calculations were performed with N set to 12 or higher.

*4.5. Numerical Implementation and Simulation Settings*

Although the developed models are capable of transient simulation, the analyses presented in this study focus on steady-state operating conditions of the liquid desiccant unit considered as a standalone subsystem. Steady-state results were obtained after confirming that all state variables within the liquid desiccant unit converged to time-invariant values. For the standalone liquid desiccant unit, the dominant time constants are associated with the thermal capacity of the absorption solution and the heat exchangers, as well as the moisture storage in the solution tank. Based on the system scale and flow rates, these time constants are on the order of several minutes, whereas the characteristic time scale of heat and mass transfer in the gas–liquid contactor is on the order of seconds. Therefore, steady-state analysis is considered appropriate for evaluating the influence of operating parameters on the supply-air conditions and performance of the liquid desiccant unit.

Pressure losses in the air and liquid flow paths are disregarded in the present model. This simplification is adopted because pressure losses have a negligible influence on heat and mass transfer characteristics and, consequently, on the supply-air temperature and humidity, which are the primary focus of this study. The main impact of pressure losses is on the electrical power consumption of fans and pumps.

The simulations were conducted using Dymola, a simulation environment for Modelica-based models. Transient simulations were performed with a time step set to 0.1 s or smaller. The DASSL algorithm was employed as the numerical solver.

Thermophysical properties of moist air, liquid water, and refrigerants were calculated using the property functions embedded in Modelica and Dymola. These property models are implemented based on established reference data, including ASHRAE standards for moist air, IAPWS-IF97 formulations for water and steam, and NIST reference data for refrigerants.

## 5. Results and Discussion

### *5.1. Steady-state performance analysis of liquid desiccant unit*

A steady-state analysis was conducted for the process side of the liquid desiccant system alone to investigate the effect of cooling-water flow rate on the supply-air conditions and to examine the distribution of latent heat removal capacity between the air–water heat-exchange coil and the air–solution contactor. The cooling-water flow rate was swept across the feasible operating range of the pump. For the analysis, the inlet air conditions were set to represent daytime summer conditions in Tokyo, and a representative operating point was chosen based on actual operation: a cooling-water mass flow rate of 0.167 kg/s. The inlet states of each substance and used coefficients are summarized in Table 1. The heat-transfer coefficient and the mass-transfer coefficient for water vapor were assigned using fixed values obtained from actual equipment tests and relevant literature [13]. It was assumed that the analysis was conducted under standard atmospheric pressure.

Table 1. Inlet conditions of air, cooling water, and solution.

| Substance | Parameter | Value | Unit |
|---|---|---|---|
| Air | Temperature | 33.0 | ℃ |
| | Relative humidity | 60.0 | % |
| | Absolute humidity | 0.0191 | kg/kg (D.A.) |
| | Mass flow rate | 0.126 | kg/s |
| Cooling water | Temperature | 12.0 | ℃ |
| | Mass flow rate | 0.10 – 0.30 | kg/s |
| Solution | Temperature | 20.3 | ℃ |
| | Mass flow rate | 0.10 | kg/s |
| Air-solution contactor | Heat transfer coefficient | 26.39 | W/ ($m^2$.K) |
| | Mass transfer coefficient | 0.02086 | kg/ ($m^2$.s) |

The calculation results are presented in Fig. 5. and 6.

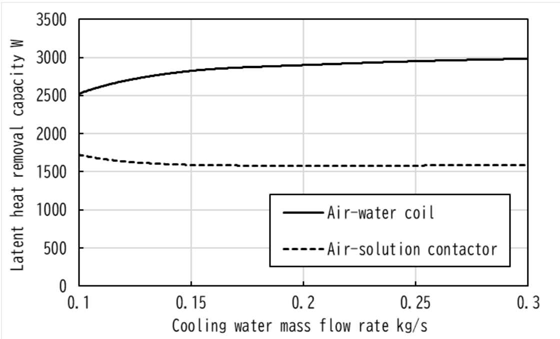


Figure 5. Comparison of latent heat removal capacity between air-water coil and air-solution contactor.

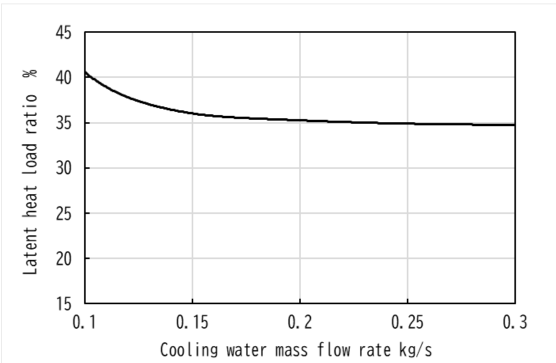


Figure 6. Ratio of latent heat load removed by dehumidification in the air–solution contactor.

The ratio of latent heat removal capacity between the air–water heat-exchange coil and the air–solution contactor was examined as a function of cooling-water flow rate. As the cooling-water flow rate increases, the cooling capacity of the coil increases, resulting in a greater amount of latent heat removed at the coil. However, it is observed that the increased latent heat removal at the coil reduces the absolute humidity of the air entering the air–solution contactor, which serves as the driving force for moisture absorption, consequently leading to a reduction in the dehumidification capacity of the contactor. These results suggest that an excessive increase in cooling-water flow rate raises the ratio of latent heat load handled by the air–water heat-exchange coil, and that careful adjustment of the cooling-water flow rate is therefore necessary to achieve the desired balance of latent heat removal between the two components.

*5.2. Power consumption calculation for the liquid desiccant system*

In this section, a comparative analysis is conducted between two air-conditioning system configurations by incorporating a room model. The first configuration consists of an indoor air-conditioning unit combined with a mechanical ventilation fan, while the second configuration consists of an indoor air-conditioning unit combined with a liquid desiccant outdoor air handling unit.

At the present stage, the room model, the indoor air-conditioning unit, and a module that supplies air with prescribed thermodynamic conditions to the room have been implemented. Based on the steady-state analysis results of the liquid desiccant unit, the supply air to the room is assumed to have constant temperature and humidity conditions corresponding to the processed air from the liquid desiccant unit. Under this assumption, a fixed amount of conditioned air is supplied to the room. Simultaneously, the electrical power consumption of the heat pump chiller supplying chilled water to the liquid desiccant processing unit, as well as that of the associated auxiliary equipment, is evaluated. The ventilation air flow rate was set to be identical for both system configurations. The rated power consumption and operating parameters of each piece of equipment were obtained from the corresponding manufacturer catalog data and used in the analysis. The thermal energy required for solution regeneration was assumed to be supplied by renewable or otherwise unused heat sources, such as waste heat. Accordingly, the energy consumption associated with the regeneration heat source was not included in the present power consumption evaluation.

The indoor air temperature is controlled by the indoor air-conditioning unit using a PI control scheme, maintaining the room temperature at a setpoint of 26.0 °C. The indoor humidity is not actively controlled and evolves according to the system operation and moisture balance in the room. The initial conditions of the outdoor air and the room were selected to represent typical summer conditions in Tokyo, based on meteorological data provided by the Japan Meteorological Agency. The metabolic heat and moisture loads generated by occupants were assumed to correspond to resting conditions and were adopted from values reported in the literature [14]. The number of occupants in the room was set to two-thirds of the nominal room capacity. Results are shown in Fig. 7–8. In the figures, the legend entry "AC+LD" denotes the configuration consisting of an indoor air-conditioning unit combined with a liquid desiccant outdoor air handling unit, while "AC" denotes the configuration consisting of an indoor air-conditioning unit combined with a mechanical ventilation fan supplying the same ventilation flow rate as the outdoor air handling unit.

Table 2. Parameters for power consumption calculation of the LD processing unit

| Subject | Part | Value | Unit |
|---|---|---|---|
| Outdoor air | Temperature | 35.0 | ℃ |
| | Relative humidity | 60.0 | % |
| Supply air from LD | Temperature | 23.1 | ℃ |
| | Relative humidity | 44.9 | % |
| | Volume flow rate | 400 | $m^3/h$ |
| Liquid desiccant unit | Pump rated power consumption | 0.70 | kW |
| | Fan rated power consumption | 0.60 | kW |
| | Cooling water mass flow rate | 0.25 | kg/s |
| | Solution mass flow rate | 0.13 | kg/s |
| Room | Volume | 100 | $m^3$ |
| | Number of people inside | 25 | people |
| | Sensible heat load per person | 75 | W |
| | Latent heat load per person | $1.8\times 10^{-5}$ | kg/s |
| | External heat load | 5120 | W |

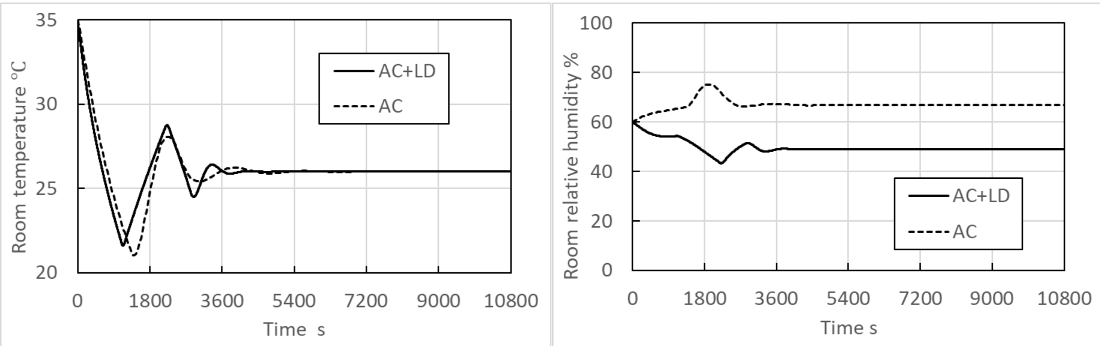


Figure 7. Temporal variations in room temperature and humidity.

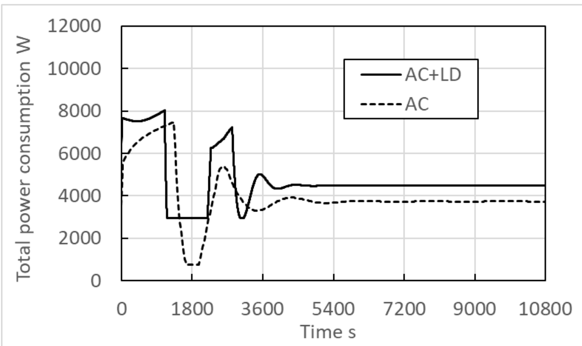


Figure 8. Temporal variation of the total system power consumption.

As can be observed from the results, the internal states of the room and the air-conditioning equipment are successfully visualized. This capability represents an essential first step toward the construction of a digital twin. As shown in Fig. 7, the use of the liquid desiccant system maintains the room relative humidity at a lower level compared with the AC-only configuration. However, the total power consumption of the system becomes higher when the liquid desiccant outdoor air handling unit is introduced. One of the contributing factors is the relatively large power consumption of auxiliary equipment, such as pumps. This result suggests that, when introducing a liquid desiccant outdoor air handling unit, it is necessary to carefully examine whether the target air-conditioned space is suitable from the viewpoint of overall energy consumption.

## 6. Conclusion

A mathematical model was developed to represent an air-conditioning system that incorporates a liquid desiccant unit as an outdoor-air handling component, and this model was implemented in the Modelica language. With the aim of enabling future development into a digital twin simulator, the system was organized on a component- and module-wise basis, with consistent input and output variables defined for each element. A standalone steady-state simulation was then conducted for the process side of the liquid desiccant unit to investigate how the cooling-water and absorbent-solution flow rates influence the supply-air conditions.

The standalone analysis demonstrated that increasing the cooling-water flow rate enhances the latent heat removal capacity of the air–water heat-exchange coil, while simultaneously reducing the dehumidification capacity of the air–solution contactor due to a decrease in the absolute humidity of the pre-cooled air. The integrated system analysis incorporating a room model confirmed that the liquid desiccant system effectively reduces the indoor relative humidity compared with a conventional ventilation configuration, though at the cost of increased total power consumption owing to the auxiliary equipment.

The project will be further advanced by carrying out the following tasks. First, simulations of the liquid desiccant unit including control will be performed, in which the cooling-water flow rate (valve opening) is treated as the manipulated variable and the supply-air humidity as the controlled variable. Second, the liquid desiccant processing unit, the room model, and the heat pump (indoor air-conditioning unit) will be connected, and integrated simulations will be conducted. In parallel, the power consumption of two system configurations will be compared: an indoor air-conditioning unit with natural ventilation, and an indoor air-conditioning unit combined with a liquid desiccant outdoor-air handling unit. Through these studies, methods for achieving energy-efficient air-conditioning operation while maintaining indoor air quality will be explored. Furthermore, by advancing digital twin technology, we aim to enable the proposal of optimal HVAC system configurations tailored to individual buildings.

## Nomenclature

| Symbol | Definition | Unit |
|---|---|---|
| $G$ | Mass flow rate | kg/s |
| $x$ | Absolute humidity | kg/kg (D.A.) |
| $h$ | Specific enthalpy | J/kg |
| $P$ | Pressure | Pa |
| $Q$ | Heat flow rate | W |
| $J$ | Mass transfer rate | kg/s |
| $h_h$ | Heat transfer coefficient | W/(m$^2$・K) |
| $h_m$ | Mass transfer coefficient | kg/(m$^2$・s) |
| N | Number of control volumes | - |
| COP | Coefficient of performance | - |
| Subscripts | Definition | - |
| $a$ | Air side | |
| $s$ | Solution side | |
| $v$ | Vapor in the air | |
| $w,s$ | Water absorbed in solution | |
| $i$ | Inlet condition | |
| $o$ | Outlet condition | |
| $cond$ | Condensed | |
| $surf$ | Surface of solution | |

**Acknowledgements**

This work was supported by the Cabinet Office, Government of Japan, Cross-ministerial Strategic Innovation Promotion Program (SIP), “Building Smart Energy Management System”.

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