Study on the Feasibility of Heat Pump Desiccant System Combined with Cogeneration System in Heating and Humidification Mode

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Abstract

Recently, in order to reduce energy consumption in the building sector, many air-conditioning systems have been proposed and applied to real buildings. Of particular note, air-conditioning systems that treat sensible and latent loads separately have been assessed as efficient in hot and humid climates. In this study, a highly efficient desiccant system combined with a cogeneration system and a heat pump desiccant system has been developed. The main concept of this system is to utilize the waste heat from the other systems, namely the cogeneration system and the solar heat system, to assist the desiccant material to recover, and to function in concert with the desiccant and heat pump systems. This paper describes a feasibility study into the proposed system, which was conducted based on an annual energy simulation, and assessed the system's performance under various conditions. The case studies were conducted using an energy simulation tool, TRNSYS 16, with comparison to a general ventilation system, a heat recovery ventilator and a heat pump desiccant system, respectively. The sensible and latent loads, indoor air temperature and relative humidity in all cases were calculated using an energy simulation. It was found that the proposed system could save more energy than these other conventional systems through its more efficient use of waste heat.

Keywords: Cogeneration, Heat-pump system, Desiccant system, Energy simulation, Ventilation

Introduction

The energy consumed by heating, ventilating, and air-conditioning (HVAC) systems accounts for about 40% of all energy consumption in the building sector, which equates to 39% of civilian energy consumption [1, 2]. Accordingly, the building sector is obliged to reduce its energy consumption while maintaining control over the humidity in buildings. Moreover, desiccant-based systems represent an environmentally friendly technology by utilizing the capabilities of sorbents to control the moisture content in air [3]. Air-conditioning systems that treat sensible and latent loads separately have already been proposed and applied. Several advanced studies have been conducted on desiccant materials, their progress in system configurations, and hybrid desiccant air-conditioning systems using solar energy and recovery energy [4]. Moreover, a novel self-regenerating heat pump system was recently developed by directly attaching sorbents to the heat exchangers of a new type of heat-pump desiccant (HPD) system [5, 6]. Aynur et al. evaluated the performances of the heating and humidification mode and dehumidification mode in the field [6, 7, 8]. The heat-pump desiccant system was concluded to consume less energy than variable refrigerants. Nevertheless, there remain some concerns about undesirable side effects, such as poor ventilation of the return air and an imbalance in the air volume. There is a high possibility that the compressor of heat pump could automatically shut down when hot and cold air is supplied to a HPD system.

Objectives

In this study, we proposed a highly efficient desiccant system combined with a cogeneration system and a HPD system (CHPD). The main feature of this system is a more efficient design of HPD system through outdoor air supply for both pre-heating and pre-humidification. This study focuses on the CHPD system that is able to satisfy the need for a comfortable indoor environment while saving energy. In order to evaluate the performance of proposed system were conducted using TRNSYS 16.

Methods

1. System description

The operational mechanism is illustrated in Figures 1. The proposed system is combined with a hot-water desiccant system and HPD system. This system will increase its focus on the energy efficiency solutions based on cogeneration and solar water heat. The HPD system, uses a type of refrigerant (R410a), consisted of two heat exchangers with adsorption material, an expansion valve, a four-way value, a hermetic scroll compressor, and ducts attached. A detailed explanation of the HPD system can be found in Reference [7]. The hot-water desiccant system consists of a water pump, a four-way valve, a three-way valve and two heat exchangers also covered with water-vapor-absorbing desiccant materials. In the first cycle (Fig. 1 (a)), hot water (60℃) flows into the left heat exchanger and raises its temperature. Humid warm outdoor air (OA2) is supplied to the evaporator of the HPD system as the supply air. At that stage, the supplied air and dehumidified return air (X) mix. The moisture in the humid air (VII) is absorbed by the evaporator surfaces of the HPD system, and is exhausted outdoors in the exhaust air (VIII). This process is expected to support a poor airflow rate from return air and to prevent automatic shutdown of the heat pump's compressor when cold air is supplied. The pre-heated outdoor air (OA1) is supplied to the condenser of the HPD system as supply air. However, the supplied air (IX) is more humid and hot, it makes a warm and humidity indoor environmental conditions. When the evaporators of the HPD system and hot-water desiccant systems need to be restored, the second cycle (Fig. 1 (b)) comes into effect. The position of each air damper and valves will be change.

2. Analysis conditions and instrumentations

In order to evaluate the various system performance conditions, several case studies were conducted using energy simulations, which were compared with conventional ventilation systems, such as general ventilation, heat recovery ventilator, HPD system and CHPD system. The analysis of simulation case was explained in Table 1. The total sensible and latent loads were simulated between the energy request and the admitted variation of indoor air temperature and relative humidity. Figure 2 is a schematic plan of the office as simulation model. The detailed simulation conditions are shown in Table 2. The office is located in Kumagaya, Japan. This office was installed a mechanical ventilation system to achieve a ventilation rate of 0.5 times per hour. The indoor air temperature and relative humidity were specified based on ASHRAE comfort ranges [3]. Table 3 shows detailed information on the HPD system, total and latent capacities, power consumption, and humidification rate. Besides, this research assumed specific capacities for the hot water desiccant systems that were half those of the HPD system, but the air exchange rate is the same at 57.4 m^3/h. The evaluation of CHPD system was carried out from November to March. During this heating period, the average outdoor air temperature and absolute humidity were found to be 7℃ and 34 g/kg respectively. This was a stand-alone operation by the ventilation and combined heating and cooling systems, but the heating and cooling systems only operated from 9:00 to 18:00 except on weekends. No heating or cooling was provided when the building was vacant.

3. Evaluation methodology

Modeling of desiccant air-conditioning systems has been studied extensively in previous research during past decades [9, 10]. It is difficult to make a simulation model that can factor in the influence of control strategy parameters, such as the mass of desiccant, the heat exchange surface areas, heat transfer coefficient, and regeneration temperature. Moreover, most of the results involved limited study of rotary desiccant systems, and the HPD system is a new type of heat exchange system with adsorption material attached to the wheel. In this instance, this study decided to adapt a model for the simulation. It is easy to apply parameters for operating data from the enthalpy capacity, sensible capacity, latent capacity and electrical energy consumption provided in the manufacturer's catalogues [11]. Accordingly, this study determined the supply temperature and humidity using the following equations.

$$\dot{Q}_{sen} = \dot{m}_{OA-SA} \cdot c_p \cdot (T_{SA} - T_{OA}) \quad \text{(Eq. 1)}$$

$$\dot{Q}_{lat} = \dot{m}_{OA-SA} \cdot (w_{SA} \cdot h_{gSA} - w_{OA} \cdot h_{gOA}) \quad \text{(Eq. 2)}$$

$$\dot{Q}_{total} = \dot{Q}_{sen} + \dot{Q}_{lat} \quad \text{(Eq. 3)}$$

$$h_g \cong 2500.9 + 1.82 \cdot T \quad \text{(Eq. 4)}$$

where C_p [kJ/kg·K] is the specific heat, h_g [kJ/kg] is the enthalpy of air, h_{gSA} [kJ/kg] is the enthalpy of supply air, h_{gOA} [kJ/kg] is the enthalpy of outdoor air, $\dot{m}_{OA-SA}$ [kg/s] is the air mass flow rate to supply air from outdoor air, $\dot{Q}_{sen}$ [kW] is the capacity of sensible heat, $\dot{Q}_{lat}$ [kW] is the capacity of latent heat, $\dot{Q}_{total}$ [kW] is the capacity of total heat, T_{SA} [℃] is the supply air temperature, T_{OA} [℃] is the outdoor air temperature, w_{SA} [kg /kg] is the humidity ratio of supply air, w_{OA} [kg /kg] is the humidity ratio of outdoor air.

A load demand is calculated based on set-temperature and set-humidity to make a good environmental condition. The supply air and humidity are giving the tendency equation from, load capacity. Here shows a tendency equation at 22℃ ~ 24℃ and 40%RH ~ 50%RH in Figure 3. Tendency of supply air and humidity from the load capacity data is shown here. Moreover, Figure 4 shows a flow chart of the load calculation for the general ventilation system and Figure 5 is for heat recovery ventilator. Figure 6 shows a flow chart of the load calculation to include discrimination for HPD system while Figure 7 is for CHPD system.

Results and discussion

The simulation results for temperature and humidity are shown in Figure 8 and Figure 9 during both stand-alone and heating/cooling-assisted operation from January 9th to January 15^{th}. It is including the weekend of January 14th and 15th. The hot water desiccant assisted HPD system provided the best thermal condition. It was found that, the HPD system maintained an indoor temperature of 12°C ~ 19°C with humidity of 56 g/kg ~ 74 g/kg. The CHPD system maintained an indoor temperature of 15°C ~ 25°C and humidity of 62 g/kg ~ 78 g/kg. The CHPD system can reduce the sensible heat load and latent heat load by about 61% from Case 1. The general ventilation system consumed more energy, at about 1,698 kWh (sensible load: 1,358 kWh, latent load: 339 kWh). The HPD system managed to reduce that by about 47%RH, while the CHPD system achieved about a 61%RH reduction. Moreover, the HPD system by itself consumed about 4,114 kWh, whereas the one integrated into the CHPD system used 1,759 kWh. The demand load, sensible load and latent load is shows in Figure 10. As seen from Figure 11, which shows the electrical energy consumption by the HPD system, it was found that the combination of a hot-water desiccant system, pre-heating and pre-cooling, and HPD system could save energy. Figure 12 shows the comparison of the indoor thermal environment as operating ventilation system

Conclusions

This paper presents the development of a highly efficient desiccant system combined with a cogeneration system and a heat-pump desiccant system (Case 4). A feasibility study of the proposed system was conducted based on an annual energy simulation. Through this case study, the system performance was estimated under various conditions. The calculation results from the simulation are shown as follows:

1. The CHPD system maintained an indoor temperature and relative humidity of 18°C ~ 24 °C and 50%RH ~ 70%RH respectively through ventilation as a stand-alone operation.

2. The general ventilation system consumed more energy at about 1,698 kWh (sensible load: 1,358 kWh, latent load: 339 kWh). The CHPD system had 61%RH less energy consumption (sensible load: 565 kWh, latent load: 105 kWh). The HPD system was about 47%RH less.

3. The electrical consumption of the CHPD’s HPD element (Case 4) was 57%RH less than that of the HPD system by itself (Case 3).

It was determined that the proposed system can achieve better energy savings than the other conventional systems through the efficient use of waste heat. In future, a more detailed simulation including the operations schedule and the system capacity will be estimated by calculating the annual performance factor. Furthermore, an experimental analysis using full-scale equipment will be conducted in a real office building.

Acknowledgment

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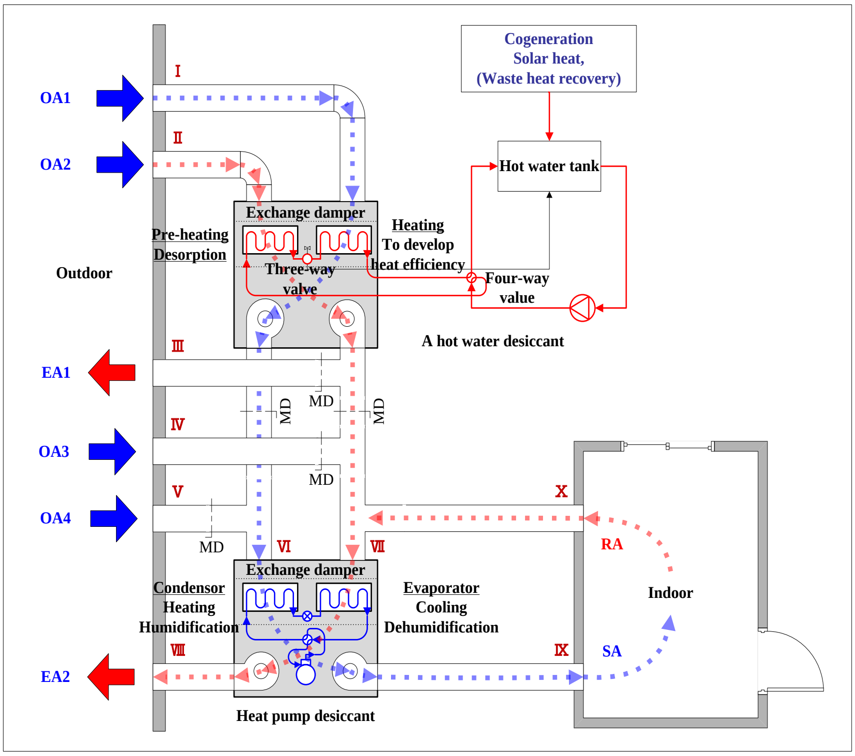
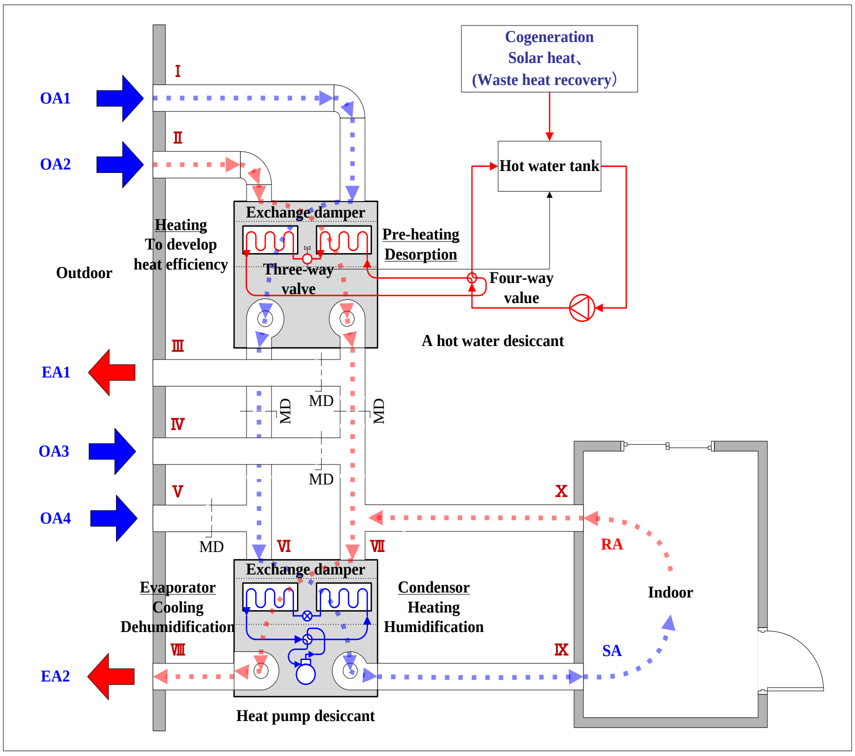


(a) first cycle



(b) second cycle

Figure 1 Operation diagram of the CHPD system

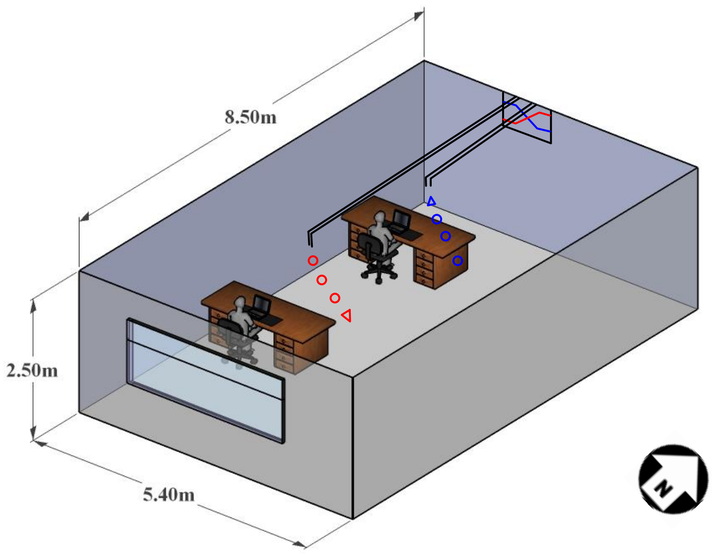


Figure 2 Schematic plan of the office

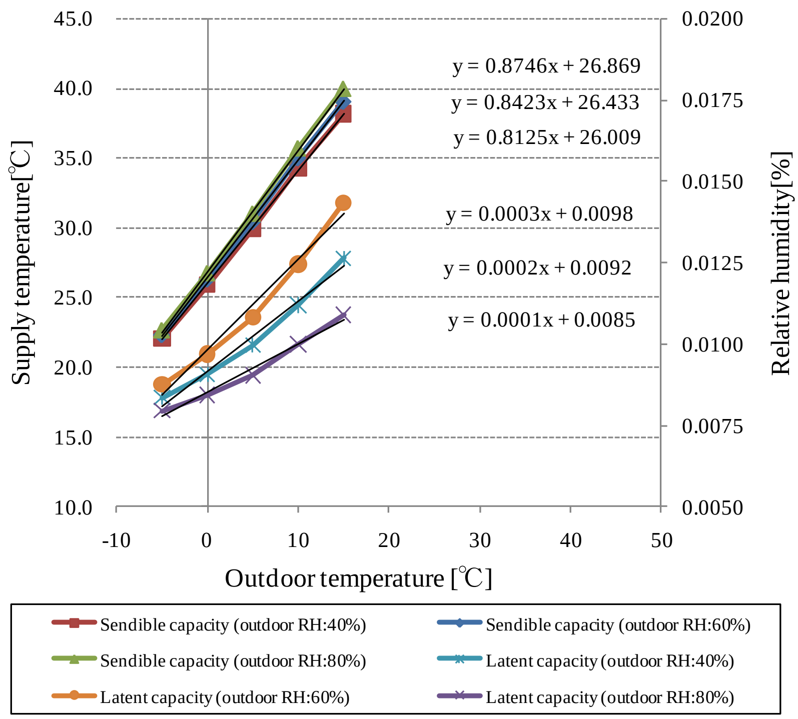


Figure 3 Tendency of supply air and humidity from the load capacity

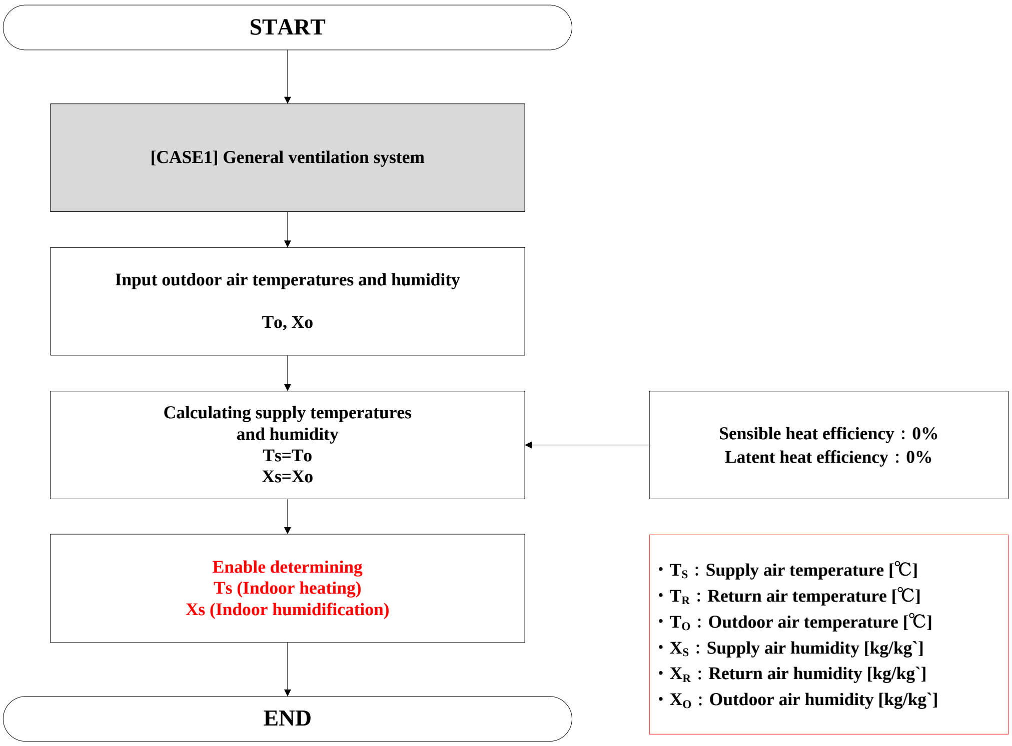


Figure 4 Flow chart of load calculation of the general ventilation system

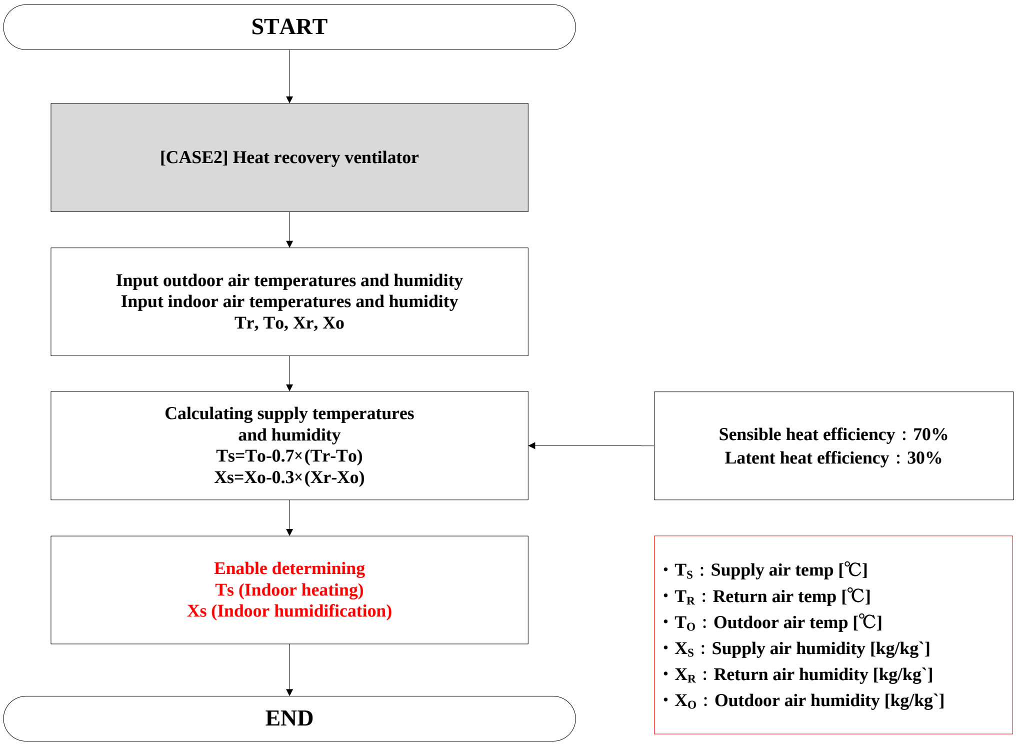


Figure 5 Flow chart of load calculation of the heat recovery ventilator

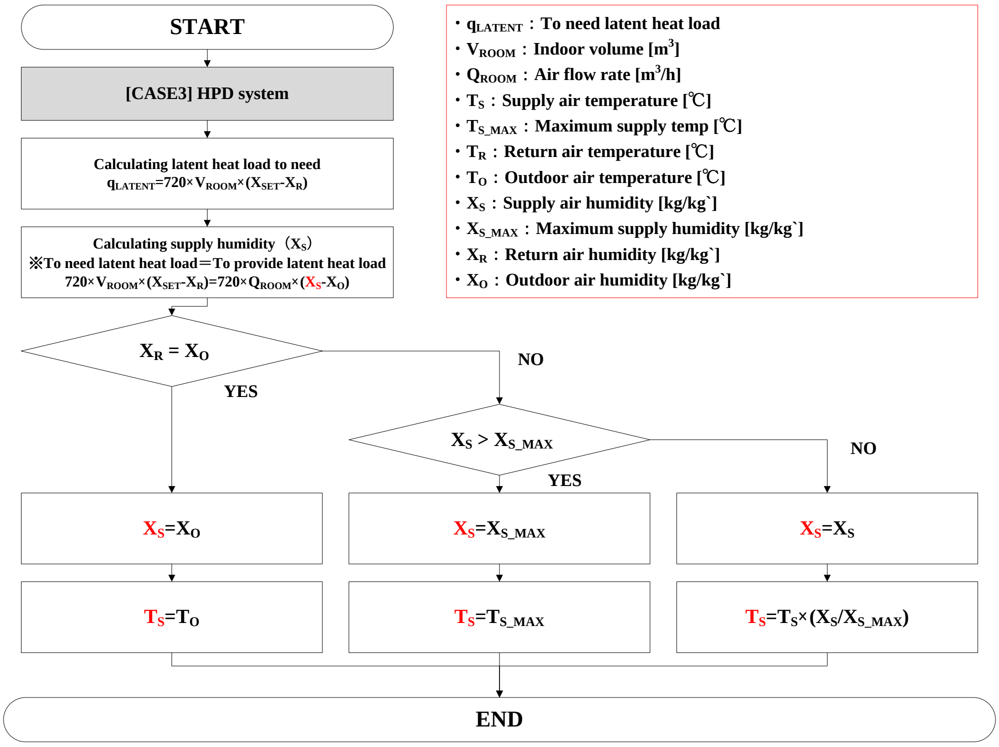


Figure 6 Flow chart of load calculation of the HPD system

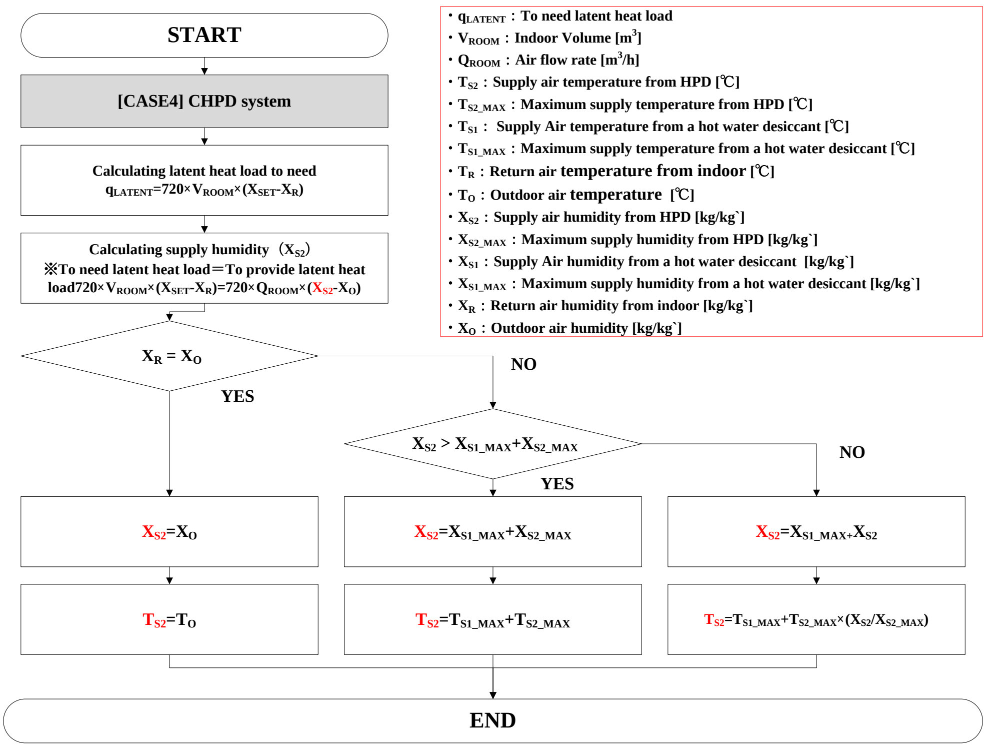
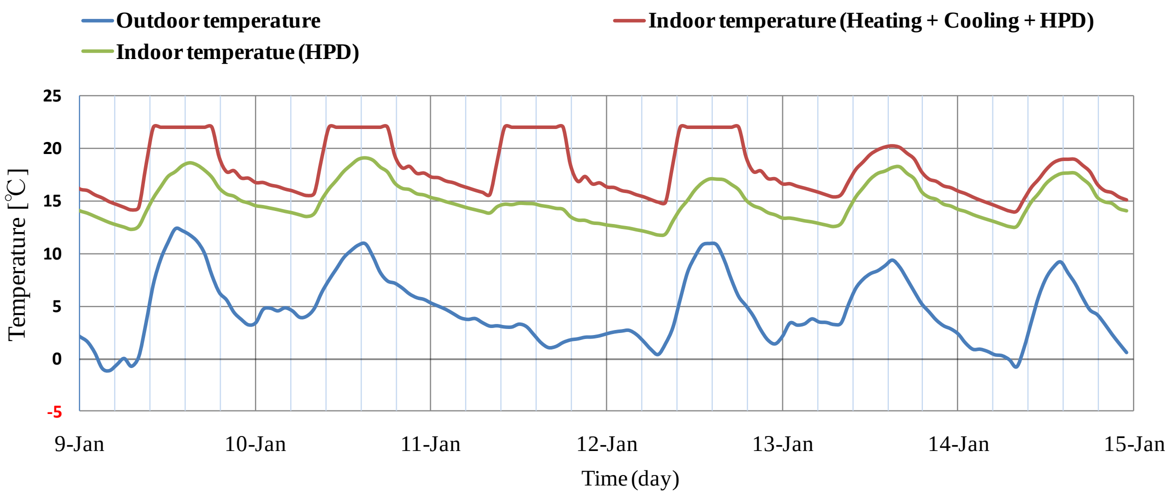
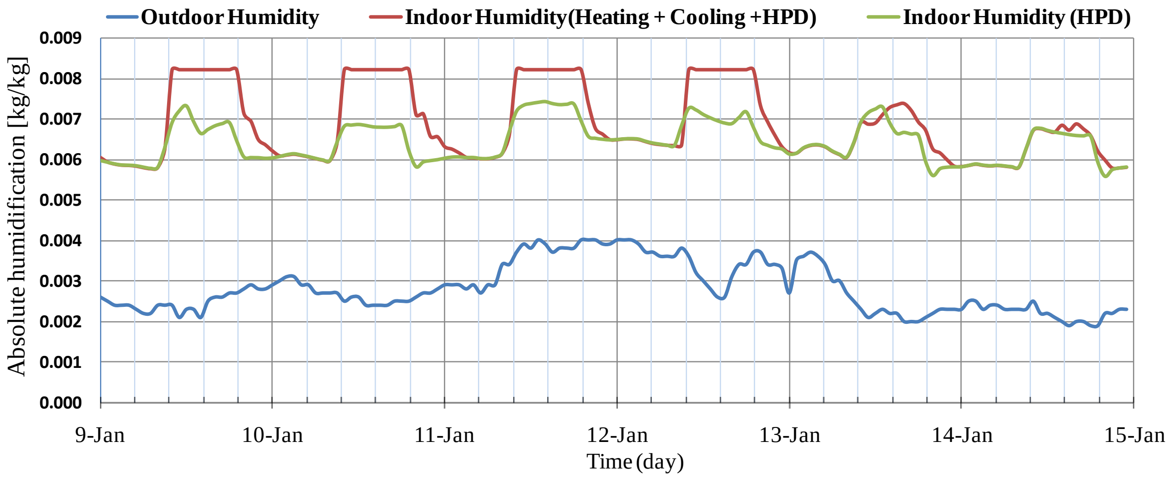


Figure 7 Flow chart of load calculation of the CHPD system

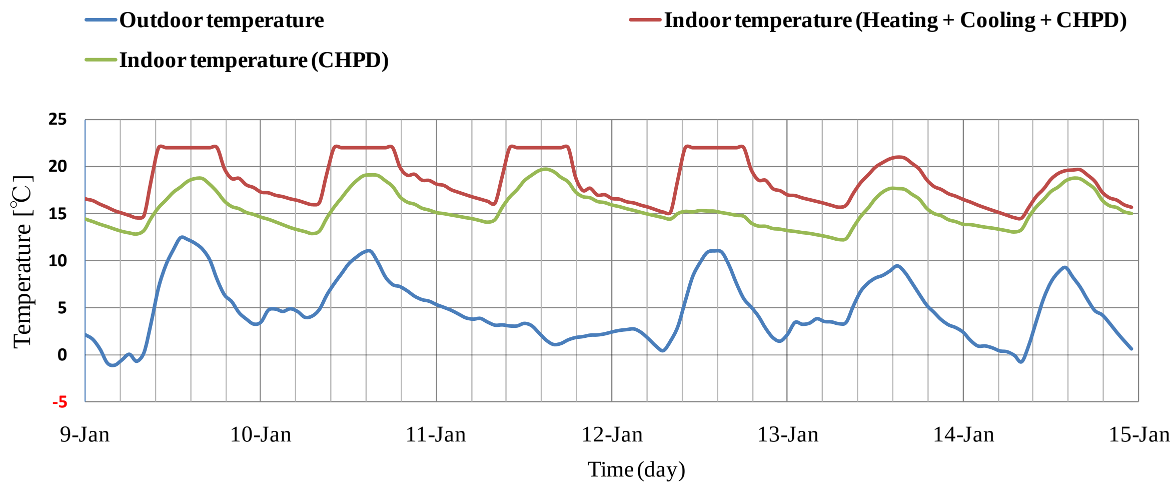


(a) Temperature distributions of indoor

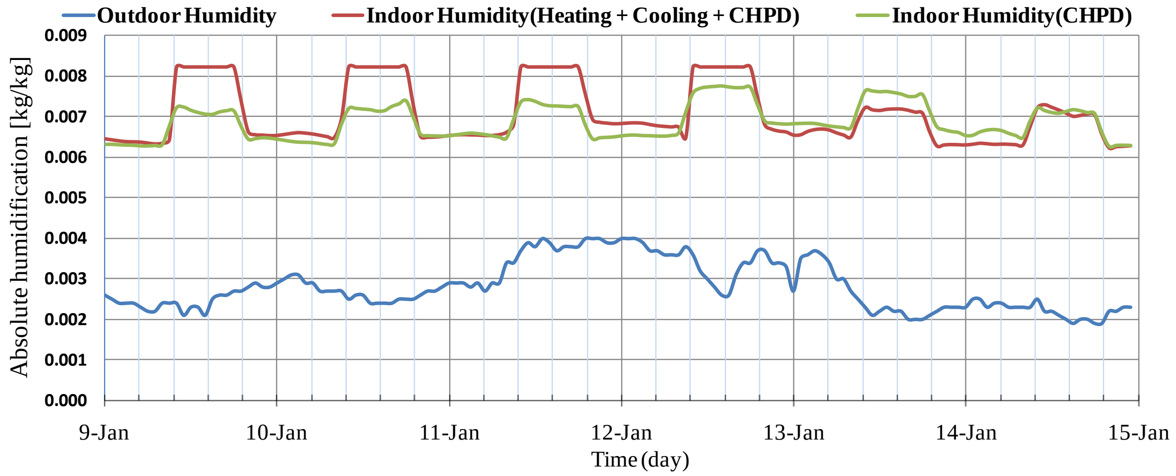


(b) Humidity distributions of indoor

Figure 8 The simulation results of Case 3



(a) Temperature distributions of indoor



(b) Humidity distributions of indoor

Figure 9 The simulation results of Case 4

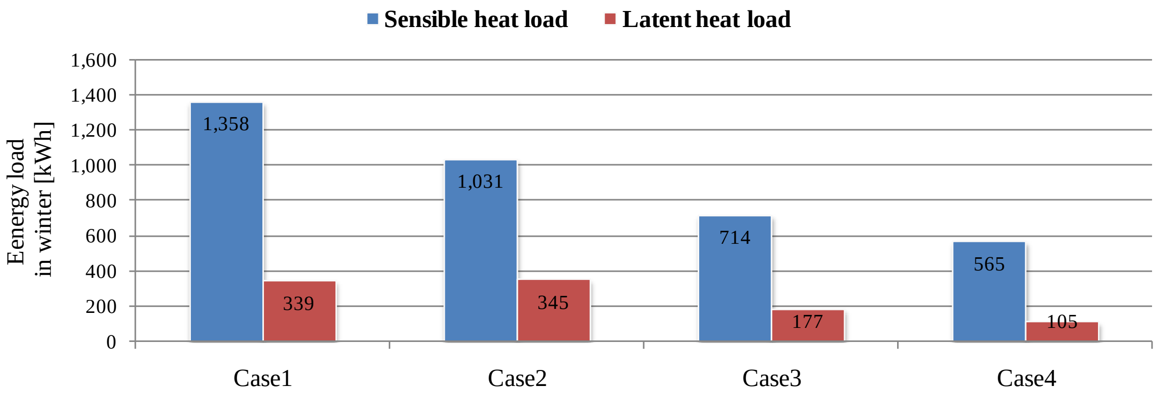


Figure 10 Case study : To request of energy load for target environment in winter

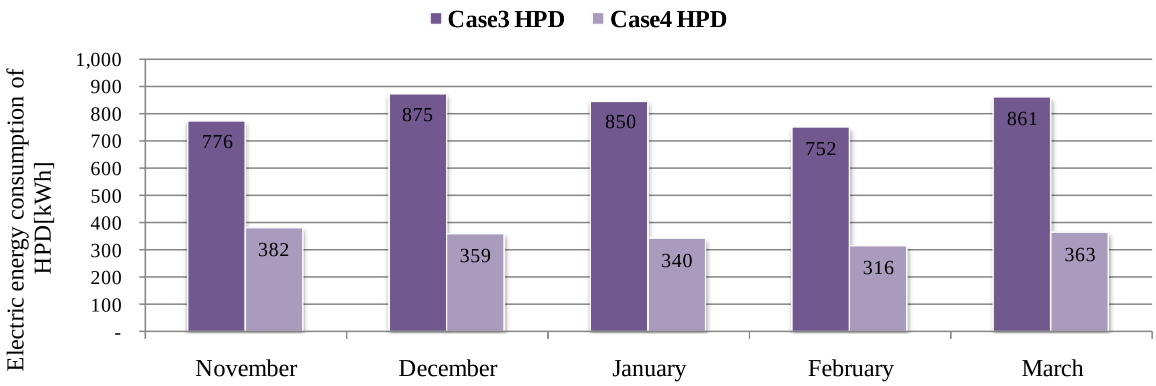


Figure 11 Compare to electric energy consumption of HPD system

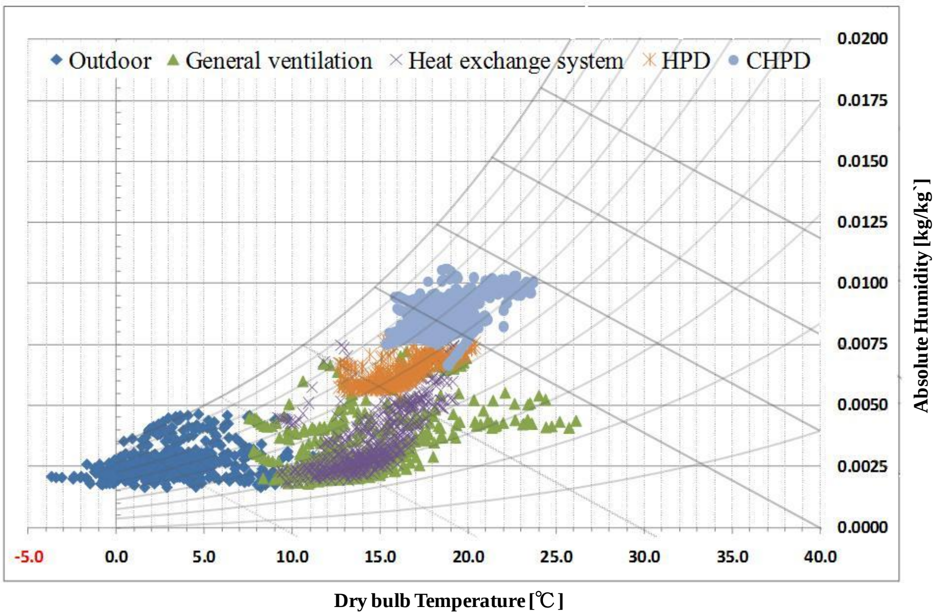


Figure 12 Comparison of the indoor thermal environment by Cases

(The indoor temperature and humidity show the provided indoor conditions by ventilation on January)

Table 1 Analysis of Cases

	Analysis of system	Reference
Case1	General ventilation system	Sensible efficiency：0% Latent efficiency：0%
Case2	Heat recovery ventilator	Sensible efficiency：70% Latent efficiency：30%
Case3	HPD system	Operation data from manufacture's catalogues
Case4	CHPD system	The capacity of Hot water desiccant System is 50% of HPD system and Here, HPD system is also used operation data from manufacture's catalogues.

Table 2 Simulation condition model and building details

Target model	Depth 5.4m × Width 8.5m × Height 2.5m
Control condition And operation schedule	Operation schedule：08~18hours Set temperature, relative humidity：22 [℃], 50[%] From November to March
Weather data	AMeDAS* weather data, Kumagaya city, Japan
Heat transfer coefficient	Wall：0.49 [$W/m^2 \cdot K$], Floor：2.00 [$W/m^2 \cdot K$], Window：5.69 [$W/m^2 \cdot K$], Window frame：8.17 [$W/m^2 \cdot K$]
Air change rate	0.5 times per hour (57.4 [m^3/h])
Internal load	Two computers (0.42 [kW/h]), Two persons (0.3 [kW/h])

* AMeDAS : Automated Meteorological Data Acquisition System, developed by the Japan Meteorological Agency

Table 3 Specification of the heat pump desiccant system

Amount of airflow		500 [m^3/h]
COP		3.78 [kg/h], 4.13
Heating and Humidification	Total heat capacity / Latent heat capacity	7.1/ 4.4 [kW]
	Power consumption	1700 [W]
Indoor conditions		22[℃], 50[%]
Outdoor conditions		0[℃], 50[%]