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Energy Performance Comparison of Two Different Types of Heat Pump-Driven Liquid Desiccant-Assisted Air-Conditioning Systems for an Apartment Building

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Abstract

The purpose of this study is to compare the energy performance of two heat pump-driven liquid desiccant (HPLD) systems that use different mechanisms for solution pre-conditioning. The design and operating logic of the HPLD systems were proposed for an apartment building. The HPLD system comprises a liquid desiccant (LD) unit for dehumidifying the process air, and a heat pump (HP) unit for cooling and heating the LD solution and cooling the process air. In Case A, the temperature and concentration of the solution can be controlled simultaneously in the HPLD system using a single sump, whereas in Case B, they are controlled separately using a dual sump that uses an additional simple device. The load of the target zone and energy performance of the HPLD system were analyzed using TRNSYS 18 and commercial engineering equation solver (EES), respectively. The results showed that the method used in Case B saved 77 % of the operating energy. The mean system COP in Case B was three times higher compared to that in Case A owing to the decrease in the solution cooling and heating loads.

Keywords: Heat pump system; Liquid desiccant system; Energy consumption; Apartment building;

1. Introduction

Recently, a liquid desiccant air conditioning (LDAC) system has emerged as an interesting system among independent handling devices for latent and sensible loads of air conditioning in terms of energy saving potential and improvement in indoor air quality [1,2]. The desiccant solution should be pre-cooled before entering the absorber and preheated before entering the regenerator to widen the difference in vapor pressure between the solution and air for enhanced dehumidification and regeneration effectiveness [3]. This process is the major factor for operating energy consumption in LDAC, and it can be realized by using low-grade heat sources [4] Heat pump systems have drawn attention because it can cover the solution cooling and heating loads simultaneously by the heat absorption process of the evaporator and heat release process of the condenser [5]. Hence, a heat pump-driven liquid desiccant (HPLD) system is considered as an efficient LDAC application owing to its ability of independent temperature and humidity conditioning and energy efficiency [6]. Additionally, a capacity matching of the HPLD system between the solution cooling and heating loads of the LD unit and the evaporator and condenser loads of the HP unit was considered [7].

On the other hand, LDAC applications have the drawbacks of large scale production and high initial cost, which hinder its potential for widespread utilization. Hence, LDAC applications were initially targeted toward commercial buildings and not residential buildings despite the distinct advantages of independent temperature and humidity control. To make LDAC applications attractive for the residential market, the LDAC applications should be improved through products focusing on the simplicity and compactness of the system [8,9]. The

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HPLD system has been recognized as a compact one among the LDAC applications. However, there is a lack of research regarding its applicability in an apartment building [10].

In this research, two HPLD systems that used different mechanisms for pre-conditioning the solution temperature and concentration were designed for an apartment building, and the overall operation strategies were explained. Energy simulation for air conditioning during the summer operation period was performed using the commercial engineering equation solver (EES) program and TRNSYS 18. The applicability of the two HPLD systems in an apartment building was compared based on their performance in terms of energy saving potential and system coefficient of performance (COP).

2. System Overview

The system consists of a heat pump (HP) unit and a liquid desiccant (LD) unit. The LD unit dehumidifies the process air to remove the indoor latent load. The HP unit pre-cools or preheats the desiccant solution and cools the dried process air to remove the indoor sensible load. To cool the solution and air simultaneously, the refrigerant from the HP unit is driven in parallel to two evaporators. Thus, the system can operate as a dehumidifying and cooling device in summer without a separate air handling device.

2.1. Heat pump unit

The heat pump (HP) unit consists of a solution-side evaporator to pre-cool the solution before entering the absorber, an air-side evaporator to cool the dried air returning from the absorber to meet the design target temperature of the supply air, a compressor using a frequency inverter for part-load operation, a solution-side condenser to preheat the solution before entering the regenerator, and an electric expansion valve (EEV) for controlling the refrigerant flow rate. The refrigerant absorbs heat from the heat sources of the evaporators (i.e., cooling capacity) and releases the heat to the heat sink of the condenser (i.e., heating capacity) through the phase change process. Considering the part-load operation mode to cover the variation of cooling loads of the desiccant solution and air, the compressor with inverter control and EEV are used. Then, a required electrical input power of the compressor by the load control can be estimated [11].

2.2. Liquid desiccant unit

The LD unit consists of an absorber (ABS) that operates as a dehumidifying device in summer, and a regenerator (REG) that makes strong solution. The outdoor air and return air are mixed and passed through the absorber to discharge its moisture to the cooled desiccant solution. The temperature of the dried absorber outlet air can be separately controlled by the air-side evaporator of the HP unit. After the dehumidification process, the absorber outlet solution becomes diluted and heated compared to the inlet state because the cooled absorber inlet solution is in contact with the hot and humid air that releases exothermic heat as the solution absorbs the moisture from the air [12]. The scavenging air that adopts the outdoor air, passes through the regenerator and absorbs the moisture from the heated desiccant solution. After the regeneration process, the regenerator outlet solution becomes strong and cool compared to the inlet state; because the heated regenerator inlet solution is in contact with the relatively cold air.

2.3. Heat pump driven liquid desiccant system

In this research, two HPLD systems are discussed: an HPLD system using a single sump i.e., Case A, (Fig. 1), and an HPLD system using dual sump i.e., Case B, (Fig. 2). In Case A, the cold and dilute absorber outlet solution, and the hot and strong regenerator outlet solution are mixed in a single sump. Hence, the concentration and temperature of the outlet solutions can be pre-conditioned simultaneously in the single sump by the mixing process that replaces the function of a conventional heat exchanger between the absorber and regenerator outlet solutions.

In Case B, the outlet solutions from the absorber and regenerator flow toward the respective sump and then circulate to each packed bed. The solution is cooled and diluted in the absorber sump. Whereas, the solution is heated and concentrated in the regenerator sump. An additional condenser is required for releasing the remaining heat to the regenerator outlet air considering the capacity matching between the HP unit (evaporator and condenser loads) and the LD unit (cooling and heating loads of the desiccant solution and air). Moreover, an additional simple device is required for exchanging the solutions between the absorber and regenerator sumps to maintain the solution concentration of each sump constant. Hence, two pipes are placed between the

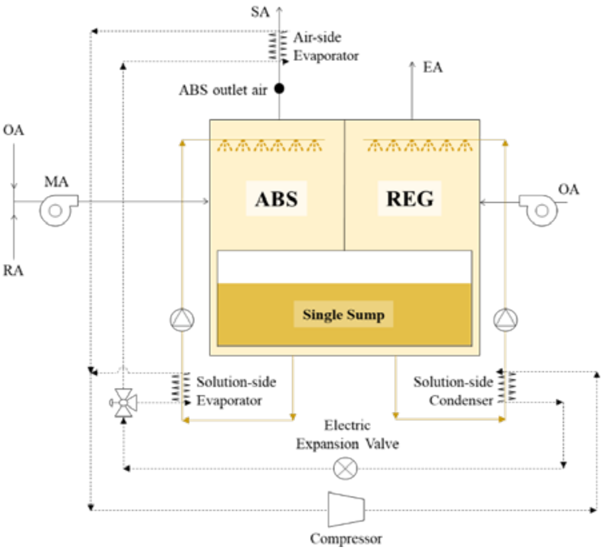


Fig. 1. Schematic diagram of a single sump HPLD system (Case A)

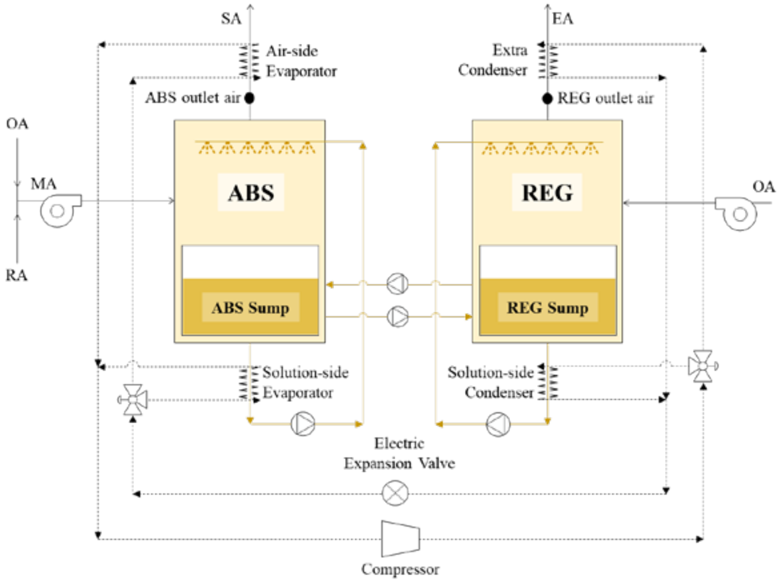


Fig. 2. Schematic diagram of a dual sump HPLD system (Case B)

dual sump for solution exchange after the dehumidification and regeneration process. The temperature and concentration of the outlet solutions can be pre-conditioned separately using the additional device. Then, it can be predicted that the compressor will work in the part-load operation mode deriving the energy consumption saving owing to the decrease in the solution cooling and heating loads.

3. Simulation Overview

3.1. Model building

A single-storey household with 4 occupants in an apartment building located in Seoul, South Korea was selected as the model building for the simulation. Table 1 summarizes the detailed information of the model building that was based on passive house design standards. The indoor target condition was set at 26 °C (T_{ra}) with 50 % relative humidity (RH_{ra}) as per the comfort zone of ASHRAE standard 55. TRNSYS 18 was used for obtaining the hourly thermal load profile of the model during the summer operation period (i.e., June–September). Considering the system compactness for the apartment building, the design flow rate of the supply air ($\dot{V}_{sa,design}$) was assumed as 800 m^3/h and the real time flow rate was variable according to the hourly thermal load. The design target condition of the supply air was determined as 15 °C ($T_{sa,design}$) with 10 g/kg humidity ratio ($\omega_{sa,design}$) based on the peak sensible load and peak latent load. The overall energy simulation was conducted using EES.

Table 1. Model building parameters

Parameters	Conducted values
Simulation period	Cooling season (June-September)
Location	Seoul, South Korea
Occupant/Schedule	4 people/ HVAC schedule addressed in ASHRAE standard 90.1
Volume	192 m^3 (10 m (W) × 8 m (D) × 2.4 m (H), a household on one floor in an apartment building)
Window-to-wall ratio	South: 0.44, North: 0.05
U-value	Floor: 0.39 W/m^2·K, Exterior wall: 0.15 W/m^2·K
Internal heat gains	Lights: 2.7 W/m^2, Equipment: 8 W/m^2, People: 70 W/person of sensible heat, 45 W/person latent heat
Indoor air set conditions	26°C dry-bulb temperature, 50% relative humidity
Supply air target conditions	800 m^3/h, 15°C dry-bulb temperature, 10 g/kg humidity ratio

3.2. Liquid desiccant unit model

Lithium chloride (LiCl) was the solution selected in this research. The absorber and regenerator solution were circulated at a constant mass flow rate of ($\dot{m}_s$) 0.26 kg/s to meet the target condition of the supply air in the proposed system. The regenerator inlet air was blown at a constant rate of 1000 m^3/h to achieve a satisfactory regeneration rate and treat the extra condensing load.

The dehumidification effectiveness (ε_{abs}) in the absorber and regeneration effectiveness (ε_{reg}) in the regenerator are defined as the ratio of the humidity ratio change between the packed bed inlet and outlet air to the maximum change with respect to the equilibrium humidity ratio ($\omega_{eq,s}$) of the inlet solution (Eq. (1) and (2)). Additionally, the dehumidification and regeneration effectiveness can be defined as the temperature change in the inlet and outlet air with the inlet solution temperature (Eq. (1) and (2)). The equilibrium humidity ratio ($\omega_{eq,s}$) of the inlet solution can be estimated with its equilibrium vapor pressure (p_s) (Eq. (3)), which is expressed as a second-order polynomial function established by Fumo and Goswami [13].

$$\varepsilon_{abs} = \frac{\omega_{abs,i,a} - \omega_{abs,o,a}}{\omega_{abs,i,a} - \omega_{abs,eq,s}} = \frac{T_{abs,i,a} - T_{abs,o,a}}{T_{abs,i,a} - T_{abs,i,s}} \quad (1)$$

$$\varepsilon_{reg} = \frac{\omega_{reg,o,a} - \omega_{reg,i,a}}{\omega_{reg,eq,s} - \omega_{reg,i,a}} = \frac{T_{reg,o,a} - T_{reg,i,a}}{T_{reg,i,s} - T_{reg,i,a}} \quad (2)$$

$$\omega_{eq,s} = 0.622 \times \frac{p_s}{101.325 - p_s} \quad (3)$$

When the conditions of the packed bed inlet solution ($T_{i,s}$, $\chi_{i,s}$) and air ($T_{i,a}$, $\omega_{i,a}$) are given, the dehumidification effectiveness (ε_{abs}) and regeneration effectiveness (ε_{reg}) can be predicted, and the temperature and humidity ratio of the outlet air ($T_{o,a}$, $\omega_{o,a}$) can be estimated using Eq. (1) and Eq. (2). To predict the dehumidification effectiveness (ε_{abs}), the model derived by Chung and Luo was used [14]. The regeneration effectiveness (ε_{reg}) was fixed at 0.3 because its experimental value in the previous literature slightly fluctuated from 0.3 to 0.4 under variable operating conditions [15].

The dehumidified moisture rate ($\dot{m}_{dehum}$) from the humid air in the absorber can be calculated by multiplying the mass flow rate of the inlet air and the humidity ratio change before and after passing through the absorber. Similarly, the regeneration rate ($\dot{m}_{regen}$) in the regenerator can be calculated by multiplying the mass flow rate of the scavenging air and its humidity ratio change. The conditions of the absorber and regenerator outlet solution can be estimated based on the mass balance and energy balance.

Considering the compactness of the system, the volume of the dual sump was set at 20 L and that of the single sump at 40 L. It was assumed that the initial conditions of the desiccant solution in the sump can be affected by the indoor set conditions. Therefore, the initial temperature ($T_{sump,old}$) and concentration ($\chi_{sump,old}$) of the solution in the sump was set to 26 °C and 30 %, and initial solution weight of the sump was set to 26 kg ($W_{sump,old}$) based on the LiCl density. It was assumed that the real-time state of the single sump can be constant at 26 °C with 30 % concentration and 26 kg of weight because the opposite outlet solutions were mixed in the single sump. On the other hand, the conditions such as temperature, concentration, and weight of the dual sump changes whenever the system operates. Hence, the conditions were calculated with transient analysis using EES. Additionally, the appropriate rate for the solution exchange was estimated to be 4 L/h, which is sufficient to cover both the dehumidification and regeneration rate.

3.3. Heat pump unit model

The refrigerant type selected in this research was R410A. To analyze each component of the heat pump (HP) unit, several assumptions were made as follows: 1) the evaporating temperature (T_{evap}) and condensing temperature (T_{cond}) of the refrigerant are constant during its phase change. 2) The effectiveness of the heat exchange of the evaporator (ε_{evap}) and condenser (ε_{cond}) is constant at 0.7, and the isentropic efficiency (η_{comp}) of the compressor is 0.75. 3) The superheating and subcooling degree can be neglected.

The evaporating and condensing temperature were determined as 12 °C and 55 °C based on several requirements as follows: 1) the estimated temperature and humidity ratio of the supply air should approach the target conditions of 15 °C and 10 g/kg. 2) The estimated temperature of the solution before entering the regenerator should be above 40 °C. 3) The required input power of the compressor should be below 4 horsepower for an apartment building. 4) After compression, the refrigerant should completely release its heat to the heat sink to avoid extra condensing load.

The thermodynamic properties of R410A of each state in the HP cycle were acquired by intrinsic function of the EES based on the determination of the evaporating and condensing temperature. The actual compressor outlet enthalpy ($h_{act,comp,o}$) can be calculated by Eq. (4), with the compressor isentropic efficiency (η_{comp}).

$$h_{act,comp,o} = \frac{h_{ideal,comp,o} - h_{evap,o}}{\eta_{comp}} + h_{evap,o} \tag{4}$$

Figure 3 shows the calculation process of the cooling capacity of the evaporator, required input power of the compressor, and the heating capacity of the condenser. The actual cooling capacity of the solution-side evaporator ($\dot{Q}_{act,evap,s}$) and air-side evaporator ($\dot{Q}_{act,evap,a}$) can be calculated using Eq. (5) and (6), based on the evaporating temperature (T_{evap}) and heat exchange effectiveness (ε_{evap}). Then, the actual cooled temperature of the absorber inlet solution ($T_{abs,i,s}$) and supply air (T_{sa}) can be obtained by Eq. (7) and (8).

$$\dot{Q}_{act,evap,s} = \varepsilon_{evap}\left(\dot{m}C_p\right)_{abs,i,s}\left(T_{abs,sump,old} - T_{evap}\right) \tag{5}$$

$$\dot{Q}_{act,evap,a} = \varepsilon_{evap}\left(\dot{m}C_p\right)_{abs,o,a}\left(T_{abs,o,a} - T_{evap}\right) \tag{6}$$

$$\dot{Q}_{act,evap,s} = \left(\dot{m}C_p\right)_{abs,i,s}\left(T_{abs,sump,old} - T_{abs,i,s}\right) \tag{7}$$

$$\dot{Q}_{act,evap,a} = \left(\dot{m}C_p\right)_{abs,o,a}\left(T_{abs,o,a} - T_{sa}\right) \tag{8}$$

The total cooling capacity ($\dot{Q}_{act,evap,tot}$) of the evaporator can be expressed as the sum of the actual cooling capacity of the solution-side evaporator and air-side evaporator. The refrigerant flow rate ($\dot{m}_{ref}$) required to treat the cooling capacity can be calculated by Eq. (9), using the enthalpy change before and after passing

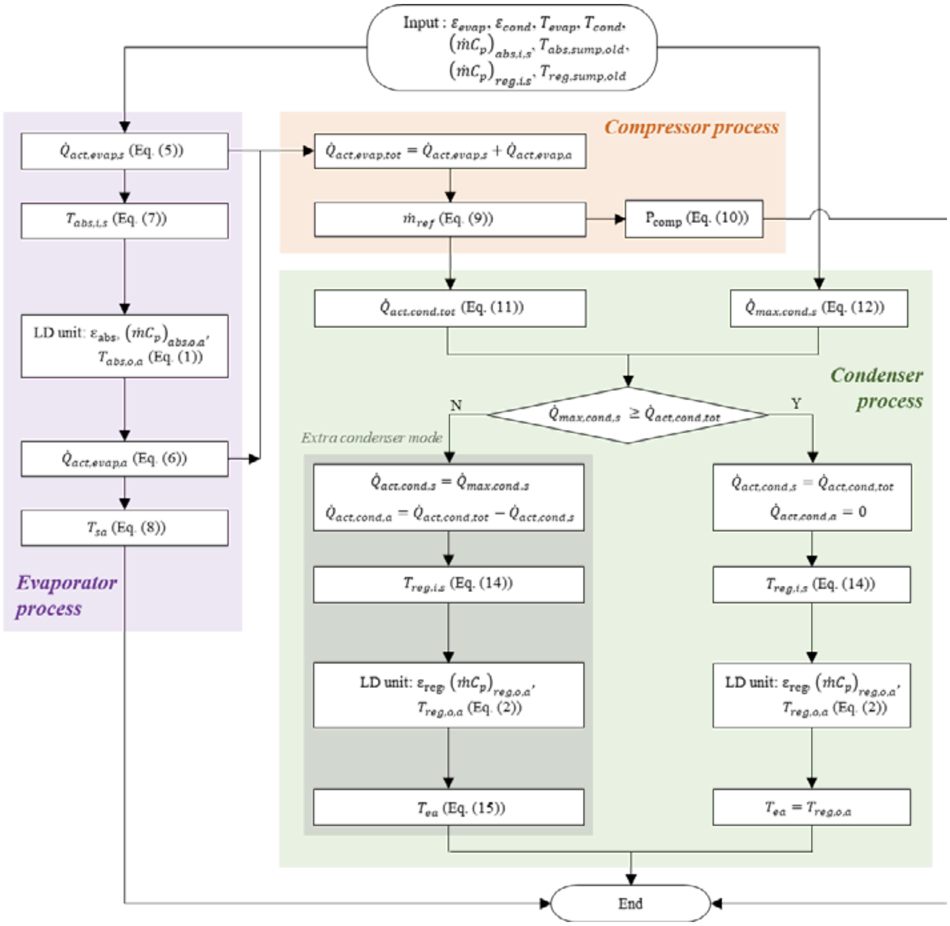


Fig. 3. Flow chart of the calculation process of the heat pump unit

through the evaporator. Then, the required input power (P_{comp}) to the compressor can be assessed by Eq. (10) with the calculated refrigerant rate.

$$\dot{Q}_{act,evap,tot} = \dot{m}_{ref}(h_{evap,o} - h_{evap,i}) \tag{9}$$

$$P_{comp} = \dot{m}_{ref}(h_{act,comp,o} - h_{evap,o}) \tag{10}$$

The total heat that should be completely released to the heat sink ($\dot{Q}_{act,cond,tot}$) can be estimated by Eq. (11) using the calculated refrigerant rate and the enthalpy change before and after passing through the condenser. The maximum heat transfer rate of the solution-side condenser ($\dot{Q}_{max,cond,s}$) and air-side condenser ($\dot{Q}_{max,cond,a}$) can be estimated by Eq. (12) and (13) based on the condensing temperature (T_{cond}) and heat exchange effectiveness (ε_{cond}). When the total heat to be released ($\dot{Q}_{act,cond,tot}$) is higher than the maximum heat transfer rate of the solution-side condenser ($\dot{Q}_{max,cond,s}$), extra condensing load occurs, which should be released to the air-side condenser. Otherwise, the solution-side condenser releases all the heat to the solution. According to the decision of the actual heating capacity of each condenser ($\dot{Q}_{act,cond,s}$,$\dot{Q}_{act,cond,a}$), the heated regenerator inlet solution ($T_{reg,i,s}$) and exhaust air (T_{ea}) can be obtained by Eq. (14) and (15).

$$\dot{Q}_{act,cond,tot} = \dot{m}_{ref}(h_{act,comp,o} - h_{cond,o}) \tag{11}$$

$$\dot{Q}_{max,cond,s} = \varepsilon_{cond}\left(\dot{m}C_p\right)_{reg,i,s}\left(T_{cond} - T_{reg,sump,old}\right) \tag{12}$$

$$\dot{Q}_{max,cond,a} = \varepsilon_{cond}\left(\dot{m}C_p\right)_{reg,o,a}\left(T_{cond} - T_{reg,o,a}\right) \tag{13}$$

$$\dot{Q}_{act,cond,s} = \left(\dot{m}C_p\right)_{reg,i,s}\left(T_{reg,i,s} - T_{reg,sump,old}\right) \tag{14}$$

$$\dot{Q}_{act,cond,a} = \left(\dot{m}C_p\right)_{reg,o,a}\left(T_{ea} - T_{reg,o,a}\right) \tag{15}$$

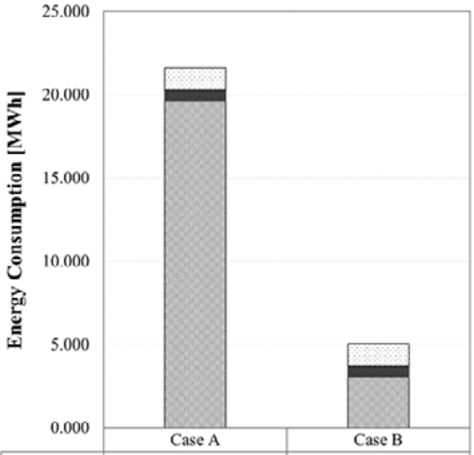
The system COP was defined as the ratio of the heat capacity of the supply air to the total electric energy consumption, as shown in Eq. (16). The total electric energy consumption was derived by the compressor, fans, and pumps. The energy consumption of the fans and pumps was predicted by an empirical model from open literature.

$$COP_{sys} = \frac{\dot{m}_{sa}(h_{ra} - h_{sa})}{P_{comp} + P_{fan} + P_{pump}} \tag{16}$$

4. Simulation Results

4.1. Energy consumption comparison

Figure 4 shows the total energy consumption converted to primary energy for the two HPLD systems using the local conversion factor of 2.75 for electricity. Case A consumed 21.6 MWh of total energy, whereas Case B consumed 5.06 MWh of total energy. The energy consumption of the fan was same for the two systems because the air flow rate of the absorber and regenerator was the same for the two systems. The pump energy consumption was 4 kWh higher in Case B due to the need for solution exchange at 4 L/h. In Case A, the temperature of the solution in the single sump can be constant at 26°C by mixing the cold absorber outlet solution and the hot regenerator outlet solution. Then, the compressor should work in full-load to cool the solution in the single sump of 26°C and heat the solution in the single sump of 26°C. In Case B, the solution in the absorber sump becomes cooled below 26°C (i.e., 15°C) and the solution in the regenerator sump becomes heated above 26°C (i.e., 45°C). Then, the compressor can work in part-load to cool the solution in the absorber sump of 15°C and heat the solution in the regenerator sump of 45°C. The compressor energy consumption saved 16.6 MWh in Case B because the compressor can work in part-load operation mode owing to the decrease in solution cooling and heating loads. Hence, Case B saved 77 % of the total energy consumption than Case A and the key factor of this result is the energy consumption of the compressor.



	Case A	Case B
□ Pumps	1.368	1.372
■ Fans	0.608	0.608
▨ Compressor	19.662	3.083

Fig. 4. Energy consumption comparison

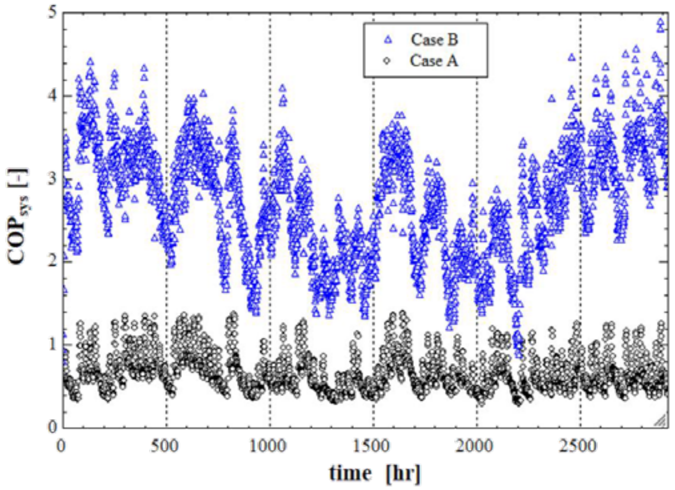


Fig. 5. System COP comparison

4.2. System COP comparison

Figure 5 shows the system COP comparison of the two HPLD systems. In all cases, Case B had a higher system COP than Case A. The mean system COP was 0.7 for Case A and 2.7 for Case B. The reason for this tendency is that both the systems satisfy the target temperature and humidity ratio of the supply air similarly. However, Case B can save operating energy consumption compared to Case A. According to the result, if the heat capacity matching between the HP unit and LD unit is achieved and the solution concentration of each sump is constant, Case B can generate the desired supply air by consuming less operating energy.

5. Conclusion

In this study, two HPLD systems, one using a single sump and another using a dual sump were suggested for an apartment building. The two systems have different mechanisms for pre-conditioning the temperature and concentration of the desiccant solution. The energy consumption and system COP for cooling and dehumidifying were compared between the two systems to analyze the applicability. The results showed that Case B reduced 77 % of the total operating energy consumption. Its system COP was always higher than that of Case A, (mean value of 2.7 against 0.7). The greatest energy-saving component was the compressor that operated in the part-load mode. Therefore, Case B can be an appropriate HPLD system for an apartment building in terms of its productivity and energy efficiency by satisfying the capacity matching and constant solution concentration.

Although, the performance comparison during the summer operation period was conducted in this study, the HPLD system can be reversible and annually operated. Therefore, additional performance comparison during the intermediate and winter season should be conducted and an optimal sequence for annual operation should be considered. In addition, an experimental verification of the simulation results in this study should be conducted in the future.

Acknowledgements

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