Impact of Multi-Stage Liquid Desiccant Dehumidification in a Desiccant and Evaporative Cooling-Assisted Air Conditioning System

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SUMMARY

The purpose of this study is to investigate the energy conservation of multi-stage liquid desiccant system operation in a liquid desiccant and evaporative cooling-assisted 100% outdoor air system (LD-IDECOAS). To enhance the dehumidification performance, a cascade liquid desiccant approach is applied in the conventional LD-IDECOAS (CLD-IDECOAS). The CLD-IDECOAS consists of a cascade liquid desiccant (CLD) unit along with direct and indirect evaporative coolers. The process air is dehumidified twice with the CLD to improve the cooling capacity of the evaporative coolers. Both conventional and proposed systems were integrated with water-side free cooling, by using the cooling tower to cool the desiccant solution. To evaluate the energy performance, both systems were simulated to serve an office space, and the results of the simulation were analyzed. The program TRNSYS 17 was used to estimate the thermal load of the model office space and cooling tower performances, whereas the energy consumption of both proposed systems were predicted using a commercial equation solver software, EES. The results showed that the CLD-IDECOAS consumed similar primary energy, but had lower thermal COP (TCOP) than the base case, due to more energy being used in the pumps and cooling tower.

INTRODUCTION

An indirect and direct evaporative cooling-assisted 100% outdoor air system (IDECOAS) has been suggested as an alternative solution to conventional vapor-compression (CFC, HCFC) heating, ventilating, and air conditioning (HVAC) systems (Kim et al. 2012, Kim and Jeong 2013). However, during the hot and humid season, the operating energy consumption of the IDECOAS significantly increased due to the low performance of the evaporative cooling system (Kim et al. 2012).

To reduce the operating energy consumption of the IDECOAS during the hot and humid season, a liquid desiccant system is applied to pre-dehumidify the induced outdoor air (OA) at the upstream of the IDECOAS (Henning 2007, La et al. 2010, Kim et al 2014). Based on the previous study by Kim et al. (2014 and 2013), a liquid desiccant and indirect/direct evaporative cooling-assisted 100% outdoor air system (LD-IDECOAS) has been proposed for use as an HVAC system in building spaces. The LD-IDECOAS is a non-vapor compression system, a type of LD-based air-conditioning system used to adjust the conditions (i.e., 15°C) of the supply air (SA) using 100% OA without mixing exhaust air (EA). This dual-function system can supply 100% OA to process SA. The sufficient ventilation of OA also improves indoor air quality. While processing the OA, the LD-IDECOAS operates like a decoupled system, which means that the latent heat load of the OA is removed by the LD unit. The sensible cooling load of the OA is then adjusted by using the indirect evaporative cooler (IEC) and direct evaporative cooler (DEC).

Kozubal et al. (2012) suggested a desiccant-enhanced evaporative air conditioner (DEVAP), which is similar to the LD-IDECOAS. The DEVAP system operates with an LD unit and high-performance IEC unit, to meet the target SA condition (i.e., 15°C) using mixed air (OA + room air [RA]). These systems use an LD unit to remove moisture from OA and an evaporative cooler to treat the sensible cooling load of the process air. Due to these characteristics, the LD unit and evaporative cooler should be operated as major components in both systems.

During the hot and humid season, the OA is dehumidified using the LD unit. Subsequently, the IEC and DEC adjust the target SA conditions. The LD unit is used to enhance the cooling effect of the evaporative cooling equipment, although in a previous research on the LD-IDECOAS, the pilot LD-IDECOAS demonstrated the limitations of the LD unit (Kim and Jeong 2014). When water-side free cooling was employed to cool the desiccant solution, the dehumidification performance of the LD unit was not sufficient to remove the latent cooling load of the OA. Due to this low performance of the LD unit, the evaporative cooler showed limited ability to meet the SA conditions. According to the results of the retrofit study (Kim et al. 2016), when water-side free cooling is used as the cooling source for a desiccant solution, the issue of low dehumidification should be solved to reach the target SA conditions.

In this study, a multi-stage LD unit is utilized in the LD-IDECOAS, to improve the dehumidification performance while using the water-side free cooling for desiccant solution cooling. This study proposes a new type of LD-IDECOAS with the enhanced dehumidification performance of a liquid desiccant unit (CLD-IDECOAS). The CLD-IDECOAS is coupled with a cascade liquid desiccant (CLD) unit with an indirect and direct evaporative cooling system. The cooling tower was simulated using the TRNSYS 17 program, to analyze the desiccant solution cooling of the LD-IDECOAS and CLD-IDECOAS. The energy performance of each system configuration during the cooling season was also estimated.

SYSTEM OVERVIEW

The LD-IDECOAS uses 100% OA as the SA for the conditioned space. The system is divided into two main parts, as shown in Figure 1. An LD unit is installed upstream of the process air for dehumidification of the OA, whereas an IEC and a DEC are installed for sensible cooling of the process air to meet the SA conditions (i.e., 15°C). Induced OA is dehumidified through the liquid desiccant system, and then sensible cooling is carried out by the continuous passage of air through the IEC and DEC. When the wet-bulb temperature (WBT) of the exhaust air (EA) is lower than that of the OA, EA should be supplied into the secondary channel of the IEC to improve the sensible cooling effect of the IEC. The sensible heat exchanger (SHE) and heating coil (HC) are located in the EA duct to maintain the EA condition for using the secondary air of the IEC. The SA flow rate is adjusted based on the required cooling load of the room as in a conventional variable air volume (VAV) system.

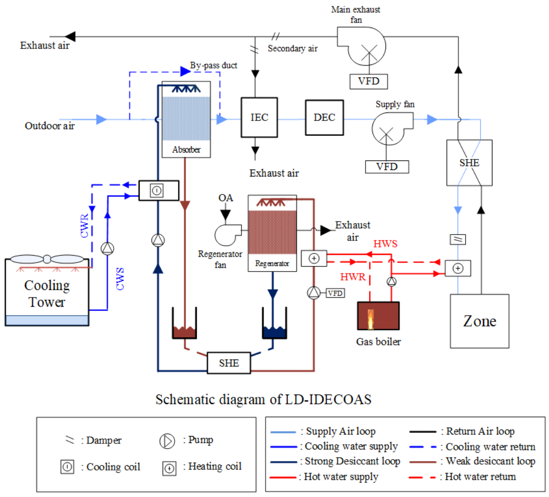


Figure 1. Schematic of the LD-IDECOAS

The cascade LD unit is applied in the base case of the LD-IDECOAS configuration and the CLD-IDECOAS focuses on the retrofitted process of the LD-IDECOAS. The cascade LD unit can achieve deep dehumidification to a specified level of induced OA humidity to enhance the sensible cooling effect of the IEC. Following this dehumidification process, the IEC and DEC operate to meet the target SA conditions. Under hot and humid OA conditions, the LD-IDECOAS cannot dehumidify the OA to meet the target SA conditions. This is because the water-side free cooling source is not sufficient to cool the desiccant solution for it to produce a suitable dehumidification effect in the single absorber tower (Kim and Jeong 2014). In this case, the cascade absorber tower can remove sufficient moisture from the OA when the free-cooling system is used to cool the desiccant solution.

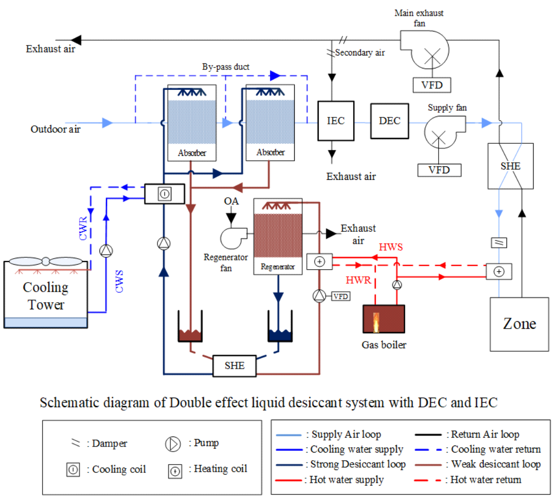


Figure 2. Schematic of the CLD-IDECOAS

SIMULATION OVERVIEW

The peak and hourly thermal loads for the model space were estimated using TRNSYS 17 (TRANSYS 17. 2009) software with IWEC summer weather data in Seoul, Korea (ASHRAE 2011, 2013) as the input. The model space was two open plan office spaces, each assumed to have a floor area of 42 m^2 with five occupants. The space conditions were maintained at the dry-bulb temperature (DBT) of 24°C with 60% relative humidity (11.24 g/kg absolute humidity ratio, and 18.6°C wet-bulb temperature) during the cold season. The internal heat gain from the occupants was taken to be 75 W of sensible and latent heat from each person, respectively. Electronic equipment are considered to produce sensible heat of 15 W/m^2 from lighting and 70 W/m^2 from personal computer.

The OA peak cooling conditions in Seoul were 34.8°C with 50% relative humidity (17.6 g/kg absolute humidity ratio and 25.9°C wet-bulb temperature). In addition, the peak thermal load was calculated as 3.29 kW of sensible load and 0.75 kW of latent load. The hourly cooling load profile of the model space was obtained using a dynamic building energy simulation software during a period of the cooling season in Seoul (e.g., July–August). For simplicity, infiltration and leakage of the model space were not considered in the thermal load calculation. Table 1 summarizes the physical information of the model space used in the thermal load estimation.

Table 1. Thermal load parameters

Number of spaces	2	
Floor area	84 m^2 (42 m^2 each room)	
Occupants	10 persons (5 persons each room)	
Schedule	ASHRAE Standard 90.1	
Room conditions	24°C, 60%	
Design SA temperature	15 C	
Window to wall ratio	18%	
U-value	Exterior wall	0.52 W/m^{2}K
	Ceiling And Floor	0.84 W/m^{2}K
	Window	5.68 W/m^{2}K
Internal heat gain	Sensible	75 W/person
	Latent	75 W/person
	lighting	15 W/m^2
	PC	70 W/m^2

To estimate the energy consumption of each system case during the cooling season, detailed energy simulations were conducted by integrating established models for each system component. The DBT of the process air leaving the LD absorber (T_{LD}) was determined using Equation (1) when the OA temperature (T_{oa}), solution inlet temperature (T_{si}), and temperature efficiency ($\varepsilon_{LD,T}$) were known. In addition, the IEC leaving process air temperature (T_{IEC}) was calculated using Equation (2), based on the efficiency of the IEC (ε_{IEC}). In this study, it was assumed that the IEC efficiency was 70% and the exhaust air from the conditioned spaces was used as the scavenged air of the IEC. The DEC leaving air temperature (T_{DEC}) was estimated using Equation (3), when the efficiency of the DEC (ε_{DEC}) was given. Based on information obtained from literature, the DEC efficiency was set at 95%.

The humidity ratio of the process air at the LD absorber outlet (ω_{LD}) was determined using Equation (4), when the dehumidification efficiency of the LD (ε_{deh}) absorber was known. According to the existing literature (Katejanekarn et al. 2009, Katejanekarn and Kumar 2008), the temperature efficiency of an LD absorber ($\varepsilon_{LD.T}$) is very close to its dehumidification efficiency (ε_{deh}).

$$\varepsilon_{LD.T} = \frac{(T_{oa} - T_{LD})}{(T_{oa} - T_{si})} \quad (1)$$

$$\varepsilon_{IEC} = \frac{(T_{LD} - T_{IEC})}{(T_{LD} - WBT_{sec,IEC})} \quad (2)$$

$$\varepsilon_{DEC} = \frac{(T_{IEC} - T_{DEC})}{(T_{IEC} - WBT_{IEC})} \quad (3)$$

$$\varepsilon_{deh} = \frac{(\omega_{oa} - \omega_{LD})}{(\omega_{oa} - \omega_{e})} \quad (4)$$

The dehumidification effectiveness of the LD unit was determined using established models suggested in literature. Chung and Luo (1999) suggested a dehumidification effectiveness model for an LD unit with LiCl solution as expressed by Equations (5) and (6). The regeneration effectiveness was estimated by Equation (7) and was obtained to be in the range of 15–30%. Thermodynamic properties of the LiCl solution were also acquired from existing literature (Conde 2004, Klein 2004).

$$\varepsilon_{Deh.\omega} = \frac{\left[1-(0.024(\dot{m}_{ai,in}/\dot{m}_{so,in})^{0.6}\exp(1.057\,(T_{ai,in}/T_{so,in})))/((aZ)^{-0.185}\pi^{0.638})\right]}{\left[1-(0.192\exp(0.615\,(T_{ai,in}/T_{so,in})))/(\pi^{-21.498})\right]} \quad (5)$$

where

$$\pi = (P_{wa(T_{so,in})} - P_{so(T_{so,in},X_{so,in})})/(P_{wa(T_{so,in})}) \quad (6)$$

$$\varepsilon_{Reg.\omega} = \frac{(\omega_{ai,in} - \omega_{ai,out})}{(\omega_{ai,in} - \omega_{e,so,in})} \quad (7)$$

The LD absorber operated with a 40% LiCl solution at with a solution inlet temperature of 30°C. The diluted desiccant solution was heated to 60°C before it entered the regenerator. It was assumed that the regeneration rate in the regenerator was identical to from the moisture removal rate in the absorber, expressed by Equation (8), when the liquid–-to-gas ratio was maintained at 1.0. Consequently, the amount of heat required for desiccant solution regeneration (Q_{Reg}) through the heating coil can be calculated using Equation (9).

$$\delta_{Deh} = \dot{m}_{SA}\ (\omega_{ai,in} - \omega_{ai,out}) \quad (8)$$

$$Q_{Reg} = \dot{m}_{so,in}\, C_{p,so,in}\, (T_{so,in} - T_{so,SHE,out}) \quad (9)$$

Conversely, in CLD operation, the operating conditions of the primary absorber tower were the same as those of the single absorber tower case. However, the process air conditions entering the secondary absorber tower were dependent on the primary absorber tower operation. The regeneration energy consumption in the CLD was the total heating energy used in the regeneration of the diluted solution from the primary and secondary absorber towers.

The OA introduced in CLD operation should be dehumidified to a suitable humidity ratio while passing through the CLD unit. The suitable ratio is defined as the attained humidity ratio of the process air to the target humidity ratio shown in Figure 3. Moreover, the target humidity ratio of the process air is the humidity ratio of the typical SA (i.e., 13°C DBT and 80% relative humidity).

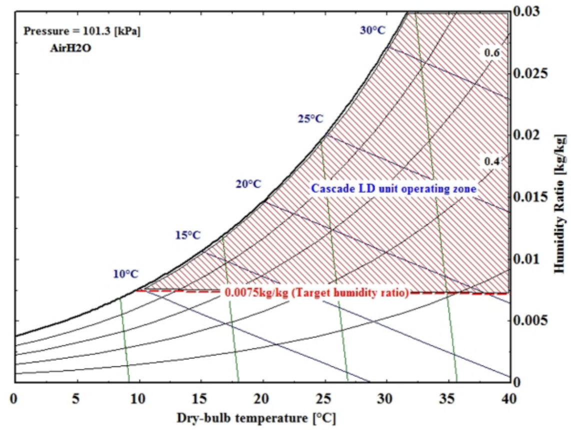


Figure 3. The target humidity ratio of CLD operation

The suggested CLD operating strategy used to achieve the target humidity ratio of the process air is as follows. If the humidity ratio of the process air entering the CLD is higher than the target humidity ratio, the process air should be initially dehumidified in the primary absorber tower. Afterwards, if the humidity ratio remains higher than the target humidity ratio, the secondary absorber tower should further dehumidify the process air. Otherwise, the process air will bypass the secondary tower. Similarly, when the humidity ratio of the process air entering the CLD is lower than target humidity ratio, the process air will bypass the CLD unit.

A cooling tower was used to cool the desiccant solution before the solution enters the absorber tower. As illustrated in Figure 4, the performance of the cooling tower was simulated by TRNSYS 17. The cooling tower module (i.e., type 51a) was selected to estimate the cooling water temperature variation, depending on the required cooling capacity of the desiccant solution. The required cooling tower capacity can be estimated using Equation (10). The cooling water mass flow rate was maintained at 0.43 kg/s, which is the design mass flow rate of the desiccant solution when the liquid–gas ratio was maintained at 1.0. Furthermore, the tower fan was considered to be at a default value from the type 51a module in TRNSYS 17. Assuming 60% heat exchanger effectiveness, Equation (11) is a simple heat exchanger equation used to estimate the cooling coil outlet solution temperature.

$$Q_{CT} = \dot{m}_{cw}\, C_{p,wa}\, (T_{CWR} - T_{CWS}) \quad (10)$$

$$\varepsilon_{SHE.sc} = \frac{\dot{m}_s\, C_{p,so}\, (T_{so,in} - T_{so,out})}{\min(\dot{m}_s\, C_{p,so}, \dot{m}_{cw}\, C_{p,wa})(T_{so,in} - T_{CWS})} \quad (11)$$

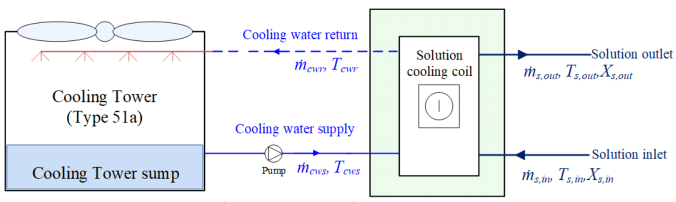


Figure 4 Schematic of the cooling tower

To evaluate the water-side free cooling performance of the cooling tower, the coefficient of performance of free cooling (COP_{sc}) was calculated by Equation (13); that is, a ratio of solution cooling load, given by Equation (12), to the operating energy consumption of the cooling tower.

$$Q_{sc} = \dot{m}_s\, C_{p,so}\, (T_{so,in} - T_{so,out}) \qquad (12)$$

$$COP_{sc} = \frac{Q_{sc}}{E_{CT}} = \frac{\dot{m}_s\, C_{p,so}\, (T_{so,in} - T_{so,out})}{E_{CT}} \qquad (13)$$

To evaluate system performance, the cooling capacity and thermal COP (TCOP) were used to estimate the LD-IDECOAS and CLD-IDECOAS. The performance indicators of both systems were evaluated using Equations (14) and (15). Based on the required heating demand for desiccant solutions, as given by Equation (9), the TCOP can be calculated using Equation (15). Additionally, the overall system energy consumption of the LD-IDECOAS and CLD-IDECOAS are converted into primary energy consumption. The primary energy conversion factors assumed were 2.75 for electricity and 1.1 for fossil fuel.

$$Q_{cooling} = \dot{m}_{SA}\, (H_{oa} - H_{SA}) \qquad (14)$$

$$TCOP = \frac{Q_{cooling}}{Q_{Reg}} = \frac{\dot{m}_{SA}\, (H_{oa} - H_{SA})}{\dot{m}_{so,in}\, C_{p,so,in}\, (T_{so,in} - T_{so,SHE,out})} \qquad (15)$$

SIMULATION RESULTS

During the simulation, peak system load occurred under the highest OA enthalpy condition, which was at 34.8° C with 50% relative humidity. Figure 5 shows the psychrometric behavior of the LD-IDECOAS under the peak load condition. It can be seen that initially, the LD unit was dehumidified by the OA. The dehumidified process air entered the IEC for sensible cooling, and then underwent adiabatic cooling in the DEC. Although it was at the peak OA condition, the base case could not meet the target SA temperature (i.e., lower than 15°C) because of inadequate dehumidification in the LD unit.

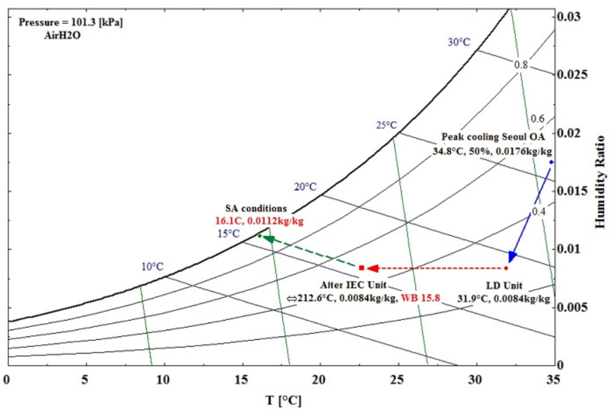


Figure 5 Psychrometric chart for the LD-IDECOAS

Figure 6 shows the psychrometric behavior of the CLD-IDECOAS under the peak load condition. In this case, the entering OA was first dehumidified in the primary absorber, and thereafter, the secondary absorber provided additional dehumidification so that the process air can attain the target humidity.

The process air that left the CLD was cooled by the IEC and DEC. Consequently, it can be seen that the SA temperature was near the target temperature (i.e., 15°C), which was not achieved in the base case.

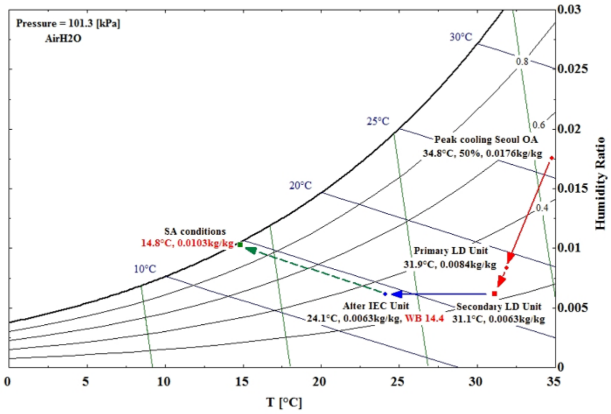


Figure 6 Psychrometric chart for the CLD-IDECOAS

The required design cooling load of desiccant solution in the LD-IDECOAS and CLD-IDECOAS were shown to be 8.8 kW and 9.8 kW, respectively. In the simulation, the cooling tower was adjusted to meet the required solution cooling capacity. Accordingly, based on the solution cooling load, the COP of the desiccant solution cooling (i.e., COPsc) was evaluated through the estimated energy consumption of the cooling tower, using the TRNSYS 17 software.

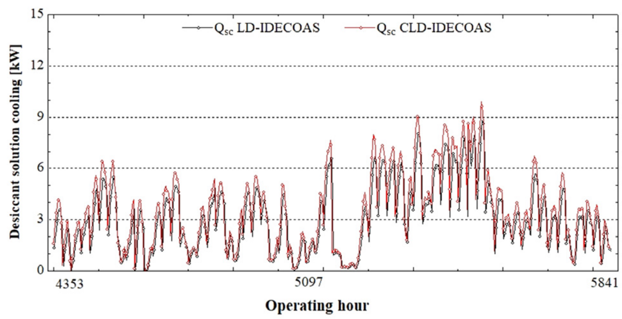


Figure 7 Solution cooling load

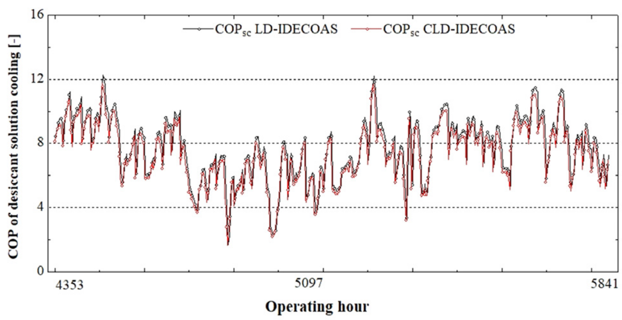


Figure 8 COP of the desiccant solution cooling

Figure 7 shows the variation of desiccant solution cooling load during the operation of each system. The cooling tower was operated to satisfy the required solution cooling capacity. Figure 8 shows the COP of the cooling tower during the desiccant solution cooling (COPsc) operation. When the tower was cooling the solution, the average COPsc was 7.7 in the LD-IDEOCAS, and 7.4 in the CLD-IDECOAS. It can be observed that the COPsc of the CLD-IDECOAS is lower than that of the LD-IDECOAS, because it consumed more energy in the cooling tower operation than LD-IDECOAS.

Through energy simulation, the operating energy consumption of each system case was estimated for the cooling periods of July and August. Figure 9 shows the hourly variation of the SA temperature delivered to the conditioned zone by each system. The figure indicates that the SA temperatures of the LD-IDECOAS and CLD-IDECOAS were maintained near the target temperature, although the SA temperature of the LD-IDECOAS deviated more from the target temperature compared to the CLD-IDECOAS variation.

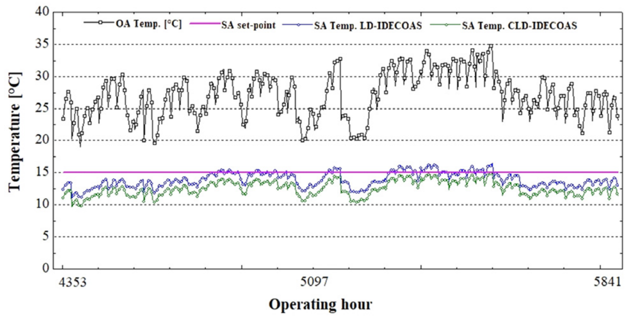


Figure 9 SA temperature variation

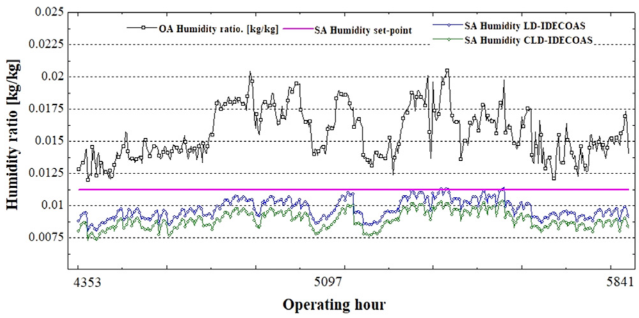
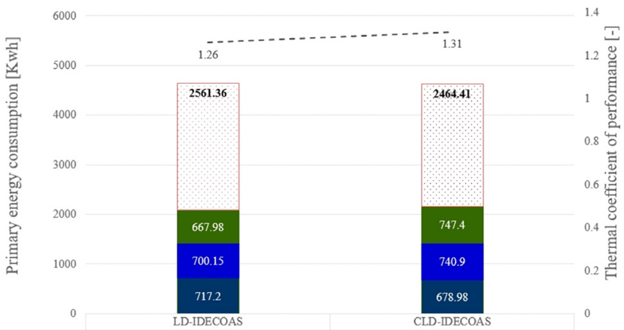


Figure 10 SA humidity ratio variation

Figure 10 compares the SA humidity ratio variations in each system case, and shows that LD-IDECOAS and CLD-IDECOAS remained at the humidity levels lower than the target SA humidity ratio during the operation period.



	LD-IDECOAS	CLD-IDECOAS
Heating energy	2561.36	2464.41
Cooling tower energy	667.98	747.4
Pump energy	700.15	740.9
Fan energy	717.2	678.98
TCOP	1.26	1.31

Figure 11 Energy consumptions and TCOP

Figure 11 presents the primary energy consumption of LD-IDECOAS and CLD-IDECOAS. It is evident that during the cooling period, both systems consumed similar primary energy.

Moreover, Figure 11 also shows that the heating energy consumption for desiccant solution regeneration of the CLD-IDECOAS was lower than that of the LD-IDECOAS. This was because the lower SA temperature acquired in CLD-IDECOAS decreased the SA flow rate to the conditioned space, and subsequently reduced the mass flow rate of the desiccant solution, which should be regenerated during CLD operation.

The thermal coefficients of each system are also shown in Figure 11. In view of the lower heating energy demand for desiccant solution regeneration, the TCOP of the CLD-IDECOAS was higher than LD-IDEOCAS.

CONCLUSIONS

In this study, system performance of the LD-IDECOAS and CLD-IDECOAS were evaluated through energy simulation. Under the peak load condition during the cooling season, the CLD-IDECOAS satisfied the SA temperature and humidity target conditions, but it consumed a slightly higher primary energy than LD-IDECOAS. However, LD-IDECOAS could not meet the target SA conditions under the peak load condition due to inadequate dehumidification in the LD unit.

Based on the results obtained from the energy simulation, both LD-IDECOAS and CLD-IDECOAS successfully conditioned the OA to satisfy the target SA conditions. In terms of energy consumption, the primary energy consumption for CLD-IDECOAS was a little lower than the LD-IDECOAS. In the TCOP evaluation, CLD-IDECOAS exhibited higher values than LD-IDECOAS.

Consequently, during the hot and humid season, the CLD unit can be applied to generate deep dehumidification of induced OA, in contrast to a single LD unit that only uses free-cooling to cool the desiccant solution. Moreover, from the results of this study, it can be seen that CLD-IDECOAS successfully conditioned the OA to the target SA condition while consuming similar primary energy to LD-IDECOAS.

ACKNOWLEDGEMENT

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