COMPARISON OF DEHUMIDIFICATION PERFORMANCE OF A SINGLE AND TWO STAGE DEHUMIDIFIERS IN A LIQUID DESICCANT-ASSISTED DEDICATED OUTDOOR AIR SYSTEM

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SUMMARY

The aim of this study is to investigate the dehumidification performance and energy conservation of a two-stage liquid desiccant dehumidifier and a single-stage dehumidifier. To enhance the dehumidification performance, a cascade liquid desiccant (CLD) system was applied in an outdoor air unit to adjust the target humidity condition of the induced outdoor air. The CLD consists of a two-stage liquid desiccant dehumidifier unit to carry out deep dehumidification of process air. Process air is dehumidified twice by the CLD system to improve the dehumidification performance of process air compared to a single dehumidifier. The single and cascade liquid desiccant systems were integrated with water-side free cooling using cooling towers for desiccant solution cooling. The energy consumption performance of both systems was evaluated through simulation when operating in an office space as a dedicated outdoor air system (DOAS). The thermal load of a model office space was estimated using TRNSYS 17, and the cooling coil load of DOAS was predicted using a commercial equation solver program. The results show that compared to a single dehumidifier system, CLD requires a lower cooling coil load.

Keywords: Liquid desiccant system, Cascade liquid desiccant, Dedicated outdoor air system

1 INTRODUCTION

Liquid desiccant (LD)-based air conditioning systems have been proposed as alternatives to conventional vapor compression (i.e., CFC, HCFC) heating, ventilating, and air conditioning (HVAC) systems. In addition, a dedicated outdoor air system (DOAS) has been proposed to enhance energy conservation and indoor air quality. DOAS is a decoupled system, which supplies 100% outdoor air (OA) with minimum ventilation air for sensible and latent cooling applications.

Recently, LD integrated DOAS has been studied to enhance the dehumidification performance and energy conservation of DOAS. An LD system has an advantage over conventional dehumidification applications in terms of the dehumidification of OA using low-grade heating source. Ge et al. (2011) proposed LD-assisted DOAS. In addition, Ge et al. (2011) also investigated an LD-assisted DOAS simulator built on a TRNSYS platform. They evaluated the dehumidification performance of the LD system at varying system parameters. Xiao et al. (2011) carried out a simulation study of LD-assisted DOAS at different operating conditions.

In this study, a single-stage LD-assisted DOAS (case 1: LD-DOAS) and a multi-stage LD-assisted DOAS (case 2: CLD-DOAS) are proposed based on previous studies. This study compared the dehumidification performance of case 1 and case 2 for varying humidity ratio at the outlet of the LD unit in each system case. The supply air (SA) condition of DOAS was estimated according to its design procedure. The energy consumption and indoor thermal comfort of LD-assisted DOAS were evaluated using a model-based control strategy. To evaluate the system energy conservation, the remained sensible cooling load of OA was calculated using the enthalpy difference between the inlet and the outlet of the cooling coil. Results from this study show that LD-assisted DOAS can improve the system energy performance.

2 SYSTEM OVERVIEW

2.1 Case 1 (LD-DOAS)

LD-DOAS is divided into two main parts. As shown in Figure 1, an LD unit is installed upstream of the process air for dehumidification of OA, whereas a cooling coil is installed for remained sensible cooling of ventilation air to meet design conditions (i.e., 15 °C, 6.26 g/kg). Induced ventilation air is dehumidified by the LD system, and then sensible cooling is implemented through the cooling coil. The sensible heat exchanger (SHE) is located in the SA duct to maintain SA conditions for using waste heat reclaimed from room air. The SA flow rate is maintained according to the required minimum ventilation air flow recommended in the ASHRAE standard 62.1 (ASHRAE 2013).

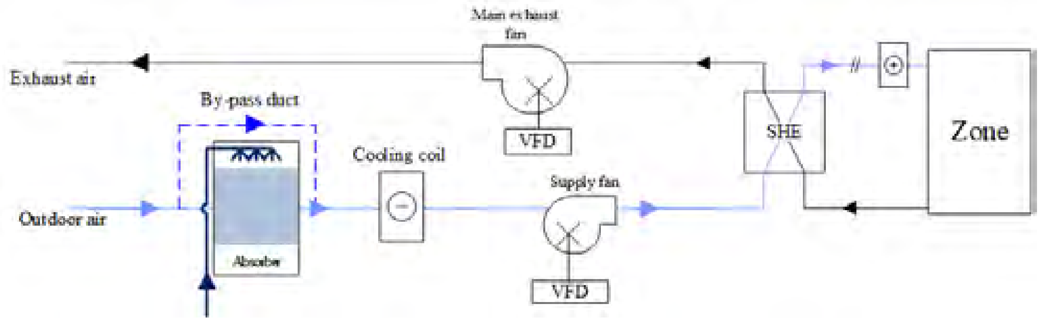

Figure 1. Schematic diagram of the LD-DOAS

2.2 Case 2 (CLD-DOAS)

A cascade LD (CLD) unit is applied in case 1 of LD-DOAS configuration; CLD-DOAS focuses on the retrofitting process of LD-DOAS. The cascade LD unit can achieve deep dehumidification to a specified level of induced ventilation air humidity. Following this dehumidification process, the cooling coil operates to meet the design conditions. Under hot and humid OA conditions, LD-DOAS cannot dehumidify OA to meet the target design conditions. This is because the single absorber unit did not generate suitable dehumidification effect. In this case, the cascade absorber tower can remove sufficient moisture from ventilation air.

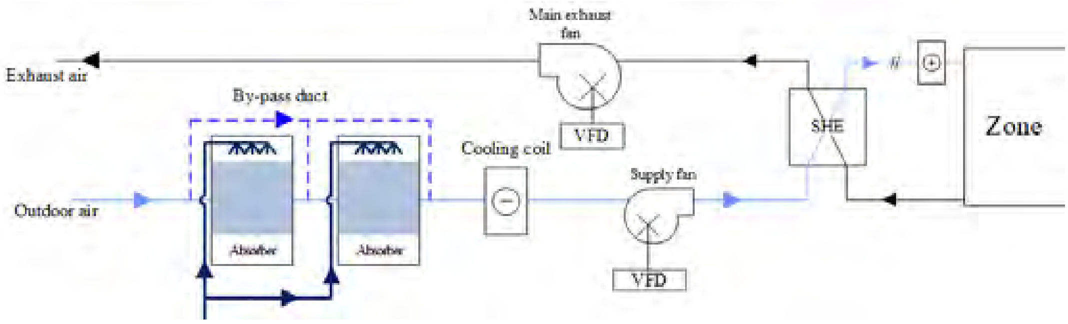

Figure 2. Schematic diagram of the CLD-DOAS

In CLD operation, the operating conditions of the primary absorber tower are the same as those of the single absorber tower in case 1. However, the conditions of the process air entering the secondary absorber tower depends on the primary absorber tower operation. The regeneration energy consumption in CLD is the total heating energy used for 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. A suitable ratio is defined as the attained humidity ratio of process air to the target humidity ratio of 3. Moreover, the target humidity ratio of process air is the humidity ratio of the required design SA condition of DOAS (i.e., 6.26 g/kg).

The proposed CLD operating strategy used to achieve the target humidity ratio of process air is as follows. If the humidity ratio of process air entering CLD is higher than the target humidity ratio, the process air should be initially dehumidified in the primary absorber tower. Then, 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 process air entering CLD is lower than the target humidity ratio, the process air will bypass the CLD unit.

In addition, for CLD operation of the case 2 system, an absorber tower was designed by using a suitable system sizing method. The CLD absorber tower packing for each stage is assumed to be half that of the single absorber of LD-DOAS (Table 1). The liquid-to-gas ratio of each CLD unit was maintained at 0.75 in the absorber tower because the supply of the desiccant solution was split in the single desiccant solution line (Figure 3). To estimate the dehumidification performance, the Chung and Luo model can be used for both the primary and secondary absorbers. The outlet desiccant solution conditions from the CLD unit was mixed at the single desiccant solution return line.

Table 1. Liquid desiccant unit assumption parameters

	Single LD unit	CLD unit
Specific surface area (a) [m^2/m^3]	223	223 (111.5 each)
Liquid-to-gas ratio [-]	1.5	1.5 (0.75 each)
Solution concentration [-]	0.4	0.4

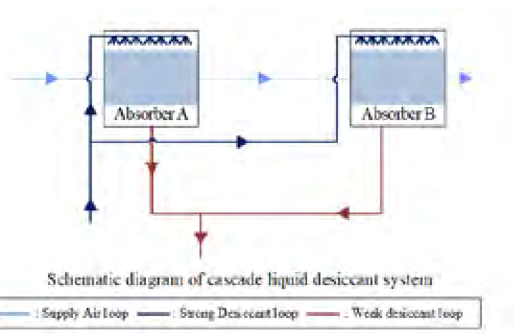

Figure 3. Schematic diagram for CLD unit

3 SIMULATION OVERVIEW

3.1 Thermal load estimation

The thermal loads for the model space were estimated using TRNSYS 17 (TRNSYS 17. 2009) software with IWEC summer weather data in Seoul, Korea (ASHRAE 2013). The model space was two open plan office spaces; each office was assumed to have a floor area of 84 m2 with ten occupants. The space conditions were maintained at dry-bulb temperature (DBT) of 24 °C with relative humidity of 60% (11.24 g/kg humidity ratio) during cold season. The internal heat gain from the occupants was assumed to be 75 W of the sensible and latent heat from each person, respectively. Electronic devices were assumed to produce sensible heat of 15 W/m2 from lighting and 70 W/m2 from a personal computer.

The hourly cooling load profile of the model space was obtained using a dynamic building energy simulation software during a period in the cooling season in Seoul (e.g., July to August). For simplicity, infiltration and leakage of the model space were not considered in the thermal load calculation. Table 2 summarizes the physical parameters of the model space used in the thermal load estimation.

Table 2. Thermal load parameters

Floor area	84 m^2	
Occupants	10 persons	
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^2K
	Ceiling And Floor	0.84 W/m^2K
	Window	5.68 W/m^2K
Internal heat gain	Sensible	75 W/person
	Latent	75 W/person
	lighting	15 W/m^2
	PC	70 W/m^2

3.2 Supply air condition of DOAS

The required minimum ventilation outdoor air flow of DOAS is designed according to ASHRAE Standard 62.1 (ASHRAE 2013). The volume flow of ventilation air was calculated using Equation (1) based on the floor area and number of occupants. In this study, the required minimum ventilation outdoor air was calculated as 50.2 l/s according to the floor area and the number of occupants listed in Table 2 (0.06 kg/s mass flow rate).

$$\mathrm{V}_{bz} = R_p \cdot P_z + R_A \cdot A_z \tag{1}$$

Where, V_{bz} : Breathing zone ventilation rate [l/s]

R_p : Ventilation rate per occupant [l/person]

P_z : Number of occupants [person]

R_A : Ventilation rate per floor area [l/m^2]

A_Z : Floor area [m^2]

The SA temperature of DOAS was set as 15 °C of DBT, which is the cooling season SA temperature condition for a typical DOAS. In addition, the humidity ratio of the conditioned outdoor air supplied by DOAS was calculated using Equation (2). The design SA humidity ratio was calculated as 6.26 g/kg using the calculation result from Equation (2); thus, case 1 and case 2 should be operated to meet the target SA conditions using the LD unit and cooling coil operation.

$$\mathrm{W}_{SA} = W_{room} - \frac{Q_{latent}}{3.0 \cdot V_{bz}} \tag{2}$$

Where, W_{SA} : Humidity ratio of the supply air [g/kg]

W_{room} : Humidity ratio set-point in the room [g/kg]

Q_{latent} : Latent load in the room [W]

V_{bz} : Breathing zone ventilation rate [l/s]

3.3 Liquid desiccant system

The dehumidification effectiveness of the LD unit was determined using established models proposed in the literature. Chung and Luo (1999) proposed a dehumidification effectiveness model for an LD unit with LiCl solution, which was expressed by Equations. The thermodynamic properties of the LiCl solution were also obtained from existing literature (Klein 2004).

$$\varepsilon_{Deh} = \frac{[1-(0.024(\dot{m}_{ven}/\dot{m}_{sol})^{0.6}\exp(1.057\,(T_{OA}/T_{sol})))/((aZ)^{-0.185}\pi^{0.638})]}{[1-(0.192\exp(0.615\,(T_{OA}/T_{sol})))/(\pi^{-21.498})]} \tag{3}$$

The LD absorber operated with an initial condition of the desiccant solution, that is, 40% LiCl solution at a solution inlet temperature of 30 °C. The specific surface area (a) of the absorber tower was assumed to be 223 m2/m3 in case 1 for a single tower and 111.5 m2/m3 in case 2 for each tower. The mass flow of the desiccant solution was supplied at 0.09 kg/s, so that the liquid-to-gas ratio was maintained at 1.5 in the single LD unit of case 1 and 0.75 in the CLD unit of case 2 (Table 1.).

3.4 Performance evaluation

To determine the dehumidification performance of the single absorber tower in case 1 and the cascade absorber tower in case 2, the humidity ratio of process air at the outlet LD unit was compared for the two cases during the cooling season in Seoul (e.g., July to August). Process air condition from the LD unit was evaluated in hourly operating steps based on outdoor air conditions. Furthermore, the variation of the humidity ratio of process air from the LD unit was also evaluated in hourly operating steps based on outdoor air conditions. Both systems should meet the design SA humidity ratio of the DOAS unit, which was set as 6.26 g/kg in the previous section.

In addition, the cooling coil load for case 1 and case 2 was estimated to compare the remained cooling energy to meet SA conditions. The available cooling load on the cooling coil was calculated using the enthalpy difference between the inlet and outlet of the cooling coil. Thus, process air conditions at the outlet of the LD unit and the SA conditions were converted to enthalpy values, respectively.

4 SIMULATION RESULTS

Figure 4 shows a comparison of the humidity ratio variation at the outlet of the LD unit in each system case. The simulation results show that the average humidity ratio of each system case is 7.32 g/kg for case 1 and 4.61 g/kg for case 2. It can be observed that the humidity level of case 2 was maintained at a lower value than the design SA humidity ratio of DOAS during operation. In contrast, the humidity level of case 1 was higher than the design SA humidity ratio of DOAS. This process is inefficient in terms of meeting the target SA humidity ratio.

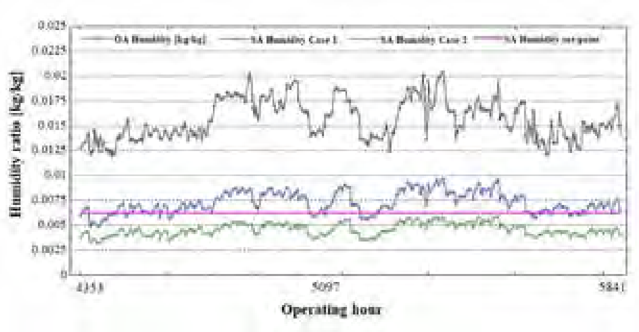

Figure 4. Humidity ratio variatioin at outlet of the LD unit

Figure 5 shows a comparison of the cooling coil load in each system case. It can be observed that case 2 required a lower cooling coil load compared to that of case 1 during operation. Given OA condition, case 1 could not meet the design SA humidity ratio owing to inadequate dehumidification in the LD unit. Therefore, after the dehumidification process, the remained cooling load of outdoor air was higher than that of case 2.

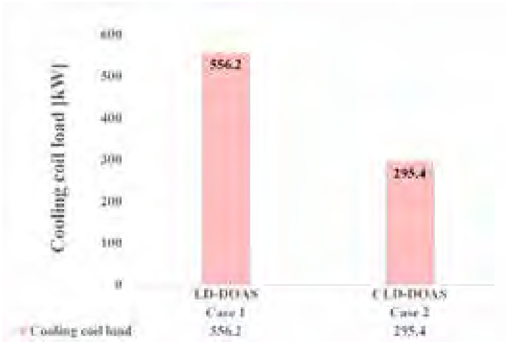

Figure 5 Comparison of the cooling coil load

5 CONCLUSIONS

In this study, the dehumidification performance of LD-DOAS and CLD-DOAS were evaluated through detailed energy simulation. During cooling season, CLD-DOAS satisfied the required humidity level of DOAS, while LD-DOAS could not meet the target humidity level of DOAS owing to inadequate dehumidification in the LD unit. In addition, results obtained from the cooling coil load calculation indicate that CLD-DOAS required a cooling coil load lower than that of LD-DOAS.

Consequently, during hot and humid season, the CLD unit can be applied to generate deep dehumidification of induced OA. Moreover, results of this study indicate that CLD-DOAS successfully conditioned OA to the target SA condition while consuming lower cooling coil energy compared to the energy consumption of LD-DOAS.

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