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Energy benefits of ventilation, dehumidification, and water heating system based on liquid desiccant cycle

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Abstract

This study aims to propose a liquid desiccant ventilation system and investigate the energy benefits of this system in comparison with a conventional variable air volume system. The proposed system is operated to achieve ventilation, dehumidification, and water heating. An energy simulation is performed via a series using TRNSYS 18 and the Engineering Equation Solver program. The simulation results demonstrate that the proposed system can save 27% of the primary energy in comparison with the reference system. The use of tap water for solution cooling instead of a cooling device is the main reason behind the significant energy savings. The heat exchanger used to provide hot water for the building with the aid of the scavenger air also reduces the overall operating energy of the proposed system.

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Keywords: Ventilation, dehumidification, evaporative cooling, water heating, liquid desiccant ;

1. Introduction

Recently, an alternative energy saving air-conditioning method has been introduced, which involves a liquid desiccant system. In a liquid desiccant system, a solution absorbs water vapor from the air in the absorber, thereby dehumidifying the air and reducing the concentration of the solution. The solution is then re-concentrated in a regenerator using solar, waste, or district heat to desorb the water vapor to the air. Previous studies have reported upon the operation of liquid desiccant systems to achieve dehumidification using variable heat sources [1-5]. Such liquid desiccant systems can significantly reduce primary energy consumption because the liquid desiccant system is a decoupled system, which can be summarized as a decoupling of the sensible and latent load functions. Dong et al. [3] suggested a desiccant-enhanced evaporative (DEVap) cooling system, and the system involving a district heat source consumed 46% less primary energy compared with the system using a conventional gas boiler. Kim et al. [6] investigated the annual operating energy performance of a desiccant- and evaporative cooling-assisted 100% outdoor air system. The proposed system showed an 82% energy saving potential over the conventional variable air volume (VAV) system during the summer. Kim et al. [7] suggested the integration of a liquid desiccant system into an evaporative cooling-assisted 100% outdoor air system. The solar water heating system used to regenerate the desiccant solution contributed to the reduction of the required operation energy. Ham et al. [8] developed a liquid desiccant and dew point evaporative cooling-assisted 100% outdoor air system (LDEOAS) and evaluated its energy saving potential. Simulation results showed that LDEOAS could save 12% of the primary energy consumption compared with a conventional VAV system.

A liquid desiccant system requires a heat source for cooling the absorber inlet solution, as well as for providing heat for regeneration. Although a cooling tower (instead of a chiller) is used to provide energy savings associated with the cooling process, chilled water is not stably produced from the cooling tower in hot and humid climates.

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In this study, tap water is used for solution cooling as the water recovered the heat from the solution. After undergoing auxiliary heating to achieve a target temperature, the water is supplied into the building as hot water. Consequently, a liquid desiccant ventilation system, which can provide both ventilation and water heating, is proposed. This system consists of several components based on the liquid desiccant system and it is operated to achieve ventilation, dehumidification, and water heating. The energy saving potential of the proposed system over the conventional VAV system is estimated by conducting an energy simulation using a dynamic simulation program (i.e., TRNSYS 18 and the Engineering Equation Solver (EES)).

2. System Overview

2.1. Liquid desiccant ventilation system

The liquid desiccant ventilation system was proposed to accommodate the ventilation, dehumidification, and water heating processes that are essential for buildings in hot and humid climates. Fig. 1 illustrates the proposed system, which consists of an absorber, a regenerator, three heat exchangers, a heating coil, and an indirect evaporative cooler.

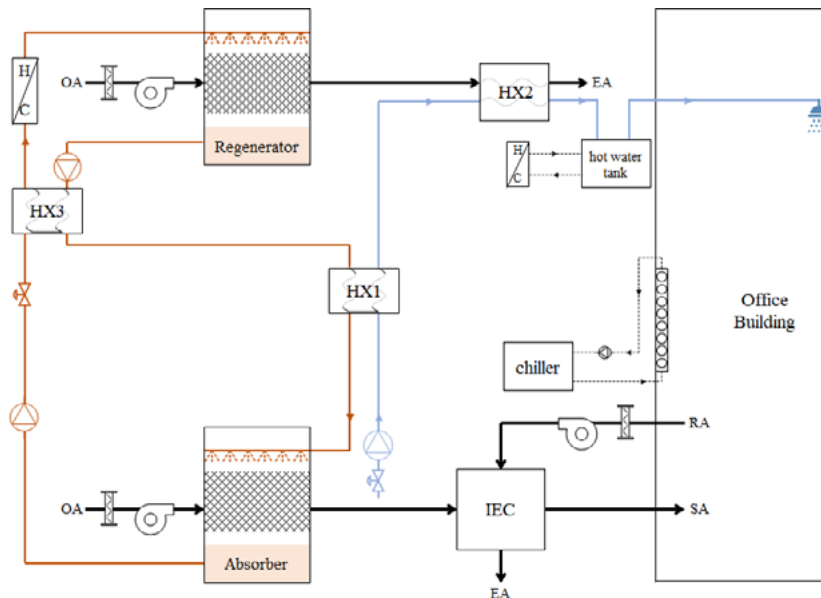


Fig. 1. Schematic of the liquid desiccant ventilation system

The process air is dehumidified by an absorber in a hot and humid climate. The air at the outlet of the absorber is dry and hot, and it must therefore be cooled to maintain the indoor air temperature. An indirect evaporative cooler is used to cool the process air. Room air is used as secondary source of air in the indirect evaporative cooler. The process air is cooled in the indirect evaporative cooler to a temperature below the average room temperature.

The process water flows into a liquid-to-liquid heat exchanger from tap water. The tap water recovers the sensible heat of the solution while the solution is cooled to be available for absorber. Next, the process water passes into an air-to-liquid heat exchanger, wherein the sensible heat of the air leaving the regenerator is recovered. If the process water does not reach a set temperature, it is supplied to the building after undergoing the auxiliary heating process.

The proposed liquid desiccant system was operated for the dehumidification of the process air. A strong solution absorbs the water vapor from the process air in the absorber. Following this, a weak solution desorbs the water vapor to the scavenger air in the regenerator. These processes occur simultaneously, and the liquid desiccant system can be used continuously.

2.2. Variable air volume system

In a hot and humid climate, the supply air volume flow rate of a VAV system is determined based on the sensible and latent loads of the building. As shown in Fig. 2, outdoor air is mixed with the return air. The mixed air is cooled by a cooling coil. The cooling coil is activated to achieve dehumidification as well as cooling. The process air is then reheated after encountering the cooling coil to achieve the desired supply air conditions (i.e., 15°C, 80%).

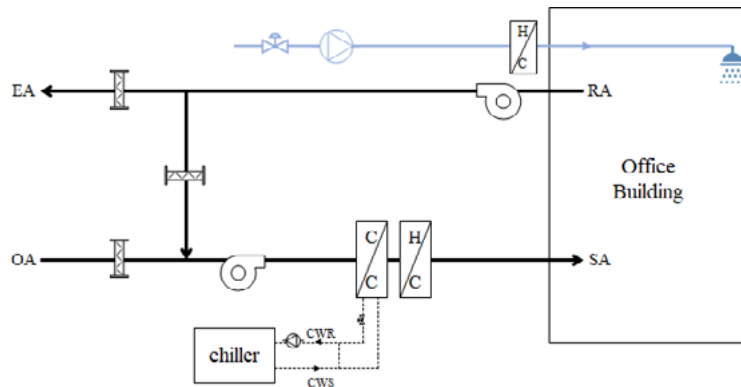


Fig. 2. Schematic of a VAV system

3. Simulation Overview

In this study, the hourly load profile of the model building was obtained by using energy simulation software (i.e., TRNSYS 18). The operating energy consumptions of both the systems required to meet the thermal load of the building were estimated using a commercial equation solver program (i.e., EES) by integrating both the system component models.

3.1. Model space

The model space was a 1,000 m² office space with a 3 m ceiling height. Two 30 m² windows were located on the south and north exterior walls, and the window-to-wall ratio was 0.16. The heat generation from the occupants and light were 115 W/person and 6 W/m², respectively, and the occupancy and system schedules recommended by ASHRAE Standard 90.1 [9] for an office building were used to obtain the thermal loads by using TRNSYS 18. The room set conditions were a 26°C dry-bulb temperature and 60% relative humidity for cooling. The physical conditions of the model space are summarized in Table 1.

Table 1. Physical parameters of the model space

Location	Seoul, South Korea	
Weather data	TMY2	
Building type	Open plan office	
Geometry	31.62 m (W) × 31.62 m (L) × 3 m (H)	
U-Values	Ceiling	0.297 W/m ² K
	Wall	0.252 W/m ² K
	Windows	1.4 W/m ² K
Internal heat gain	Occupants	115 W/person [10]
	Light	6 W/m ²
Room set points	Temperature	26°C
	Relative humidity	60%

3.2. Liquid desiccant unit

An empirical model for predicting the effectiveness of an absorber to absorb moisture was found in literature [11]. The effectiveness values of the absorber to affect the moisture and temperature of the input air indicate the degree of heat and mass transfer. The temperature differences between the air and desiccant solution allows for the exchange of thermal energy due to heat and mass transfer. The difference in the moisture contents between the air and desiccant solution permits mass transfer between the air and solution. In addition, the mass flow rates of the fluids and the system design parameters are also important factors in determining the effectiveness of the absorber. The initial mass flow rate and concentration of the desiccant solution are 0.5 kg/s and 42%, respectively. The absorber inlet temperature of the solution is 20-25°C. The process air that flows into the absorber is outdoor air. The equilibrium humidity ratio and enthalpy of the desiccant solution were calculated using previous studies [12,13].

The regenerator model was suggested by Martin and Goswami [13]. The equation for this model is a function of several variables. These variables are combined to form non-dimensional parameters, and this model can predict the effectiveness of the regenerator using these non-dimensional parameters. The liquid-to-gas ratio is set at 4.0, and the temperature effectiveness of the regenerator is same as the mass effectiveness [7]. The equilibrium humidity ratio of the regenerator is calculated using the same study that the absorber was based upon [12]. The inlet temperature of the solution is set at 60°C. The effectiveness of the dehumidification and regenerator were determined using Eq. 1 and 2. The absorber and regenerator outlet conditions of the process air were determined using Eq. 3.

$$\epsilon_{abs} = \frac{\left(1 - \frac{0.024 \left(\frac{\dot{m}_a}{\dot{m}_s}\right)^{0.6} \exp\left(1.507 \frac{T_{a,in}}{T_{s,in}}\right)}{(aH)^{-0.185} \pi^{0.638}}\right)}{\left(1 - \frac{0.192 \exp\left(0.615 \frac{T_{a,in}}{T_{s,in}}\right)}{\pi^{-21.498}}\right)} \tag{1}$$

$$\epsilon_{reg} = 1 - 48.3 \left(\frac{\dot{m}_s}{\dot{m}_a}\right)^{0.396} \left(\frac{Y_L}{Y_C}\right)^{-1.57} \left(\frac{h_{a,in}}{h_{s,in}}\right)^{-0.751} (aH)^{0.0331} \left(\frac{Y_L}{Y_C}\right)^{-0.906} \tag{2}$$

$$\epsilon_{abs,reg} = \frac{w_{a,in} - w_{a,out}}{w_{a,in} - w_{eq}} = \frac{T_{a,in} - T_{a,out}}{T_{a,in} - T_{s,in}} \tag{3}$$

3.3. Indirect evaporative cooler

The dry-bulb temperature at the outlet of the indirect evaporative cooler (IEC) can be estimated using Eq. 4 when the IEC effectiveness is provided. In the proposed system, the humidity ratio of the process air that leaves the IEC is identical to that of the exiting liquid desiccant because condensation and dehumidification does not occur in the IEC. The IEC effectiveness was assumed to be 0.7.

$$\epsilon_{IEC} = \frac{T_{pri,in} - T_{pri,out}}{T_{pri,in} - WB T_{sec,in}} \tag{4}$$

3.4. Heat exchanger

Two different types of heat exchangers were used in this study for the heat recovery processes. One is a liquid-to-liquid type and the other is liquid-to-air type. The liquid-to-liquid type heat exchanger was used to transfer the heat from the solution leaving the regenerator to the solution leaving the absorber. The other liquid-to-liquid type heat exchanger was used to transfer the heat from the solution leaving heat exchanger 3 to the tap water. The liquid-to-air type heat exchanger was used to transfer the heat from the scavenger air to the process water. The outlet conditions of each exchanger were estimated using Eqs. 5 to 7 and the efficiency of each heat exchanger was assumed to be 0.8, in accordance with literature [5].

$$C^* = \text{MIN}(\dot{m}_{\text{hot}}C_{p,\text{hot}}, \dot{m}_{\text{cold}}C_{p,\text{cold}}) \quad (5)$$

$$\dot{Q}_{\text{ideal}} = C^*(T_{\text{hot},\text{in}} - T_{\text{cold},\text{in}}) \quad (6)$$

$$\dot{Q}_{\text{real}} = \varepsilon\dot{Q}_{\text{ideal}} = \dot{m}_{\text{hot}}C_{p,\text{hot}}(T_{\text{hot},\text{in}} - T_{\text{hot},\text{out}}) = \dot{m}_{\text{cold}}C_{p,\text{cold}}(T_{\text{cold},\text{out}} - T_{\text{cold},\text{in}}) \quad (7)$$

3.5. Supply air flow rate

In the VAV system, the supply air flow rate was determined based on the sensible and latent loads using Eq. 8. The return air flow rate was calculated using the supply air flow rate and the minimum required outdoor air ventilation rate, as recommended by the ASHRAE Standard 62.1. On the other hand, in the liquid desiccant ventilation system, the supply air flow rate was same as the minimum required outdoor air ventilation rate. The required outdoor air flow rates per person and per unit area are 2.5 L/(s person), and 0.3 L/(s m²), respectively, based on the ASHRAE guidelines for an office space. Thus, the minimum required outdoor air ventilation rate for the both systems was 0.51 kg/s, and the maximum supply air flow rate for the VAV system was determined to be 1.265 kg/s.

$$\dot{m}_{\text{sa}} = \frac{\dot{Q}_{\text{sen}} + \dot{Q}_{\text{lat}}}{h_{\text{ra}} - h_{\text{sa}}} \quad (8)$$

3.6. Cooling and heating coil

A cooling coil is activated to achieve dehumidification and cooling of the process air in the VAV system. This coil performs dehumidification via condensation, as well as cooling of the process air, to satisfy the target humidity ratio of the process air. The cooling load of the coil was calculated using the difference between the inlet and outlet enthalpies. The inlet condition of the cooling coil is same the mixed air condition and the outlet condition is set to be the dew point.

A heating coil heated the solution in the regenerator of the liquid desiccant ventilation system. In the VAV system, the process air is reheated by a heating coil. The outlet temperature of the cooling coil is low such that reheating is necessary to satisfy the set point conditions. Additionally, a heating coil is used for water heating in both systems.

3.7. Parallel unit

In the proposed system, the supply air did not remove the sensible load of the conditioned zone, whereas the supply air of the reference system sufficiently removed the sensible load. Therefore, a fan coil unit was used as a parallel system in the proposed system, in which the remaining sensible load of the conditioned zone was removed. The load of the parallel unit was determined by Eq. 9.

$$\dot{Q}_{\text{parallel}} = \dot{Q}_{\text{sen}} - \dot{m}_{\text{sa}}(h_{\text{ra}} - h_{\text{sa}}) \quad (9)$$

3.8. Chiller and boiler

Energy derived from a chiller was consumed in both systems. In the proposed system, it is used as a parallel unit that is located in the building. In the reference system, it is used as a cooling coil. The air-cooled chiller model provided by EnergyPlus [14] was used for estimating the chiller energy consumption.

A conventional gas boiler was used for the reheating coil of the VAV system, heating coil of the liquid desiccant ventilation system, and water heater for both systems. The boiler was operated to achieve the heating loads required by the reheating coil, heating coil, and water heaters. The boiler was modeled using a function of boiler efficiency and the efficiency performance curve. The efficiency curve is a cubic efficiency curve of the part load ratio that was provided by EnergyPlus [14]. The part load ratio is the ratio of the current heating load to the boiler capacity. The theoretical boiler efficiency was set at 82% in this study [3].

3.9. Fan and pump

The design of the fan powers in both systems were calculated using a fan efficiency of 0.5 and Eq. 10. The maximum air flow rate was used as the volume flow rate of this design. The simulation of the proposed system was constructed using three fans: variable volume supply, return, and constant volume liquid desiccant (LD). The reference system was designed with variable volume supply and return fans. The fan power at the variable volume flow rate was estimated using Eq. 11 based on the generic fan power curve.

$$P_{fan,design} = \frac{\dot{V}_{design} \times \Delta P}{\eta} \tag{10}$$

$$P_{fan} = (0.0013 + 0.0447PLR + 0.9506PLR^2 - 0.0998PLR^3) \times P_{fan,design} \tag{11}$$

The pump powers for both systems were calculated using Eq. 12 with a pump efficiency of 0.6. In the proposed system, the constant speed pump supplies the liquid desiccant circulation, hot water, and the chilled water for the cooling coil and fan coil unit (FCU). In the reference system, the pump supplies the chilled water for the cooling coil and the hot water.

$$P_{pump} = \frac{\dot{m} \times H \times g}{\eta} \tag{12}$$

4. Simulation Results

4.1. Thermal behavior of the air

The conditions of the process air over the course of a day are depicted in Fig. 3. The hot and humid outdoor air was dehumidified to the required humidity ratio in the liquid desiccant unit and cooled to below the set temperature of the room in the liquid desiccant unit and indirect evaporative cooler. The temperature gradient of the process air was greatest in the indirect evaporative cooler. Finally, the supply air possessed approximately a 0.007 kg/kg humidity ratio and 22°C dry-bulb temperature.

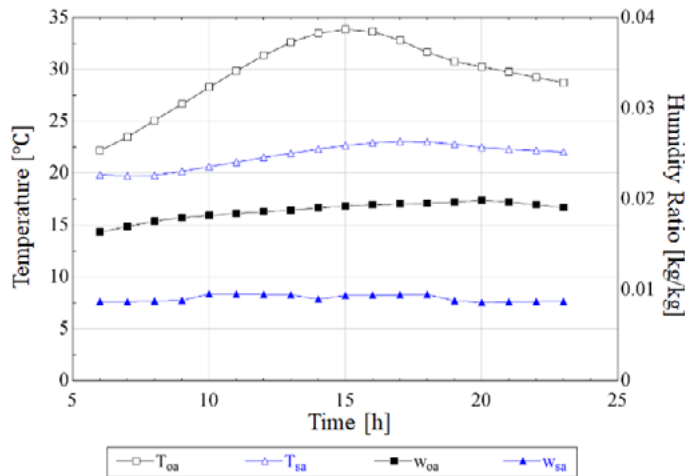


Fig. 3. Temperature and humidity ratio profile of the process air

4.2. Primary energy consumption

To compare the overall energy performances of the proposed and reference systems, the operating energy consumption was converted into the primary energy consumption for each heat source.

Fig. 4. shows the comparison of the primary energy consumptions in the proposed and reference systems. The water-heating energy is lessened in the proposed system as heat from the two heat exchangers is recovered by the process water, while the reference system consumed this heat. Although the proposed system consumed the heat for the regenerator, the total heating energy of the proposed system is less than that of the reference system because the recovered heat from the solution was used for water heating in the proposed system. The proposed system also demonstrated the requirement of a lower amount of cooling energy for the solution because tap water is used for solution cooling. Additionally, the indirect evaporative cooler was used to supply air cooling. The fans consumed energy in the supply, return, and liquid desiccant cycle processes in the proposed system. The fans in the reference system consumed energy for only the supply and return processes. The proposed system consumed less fan energy than the reference system because the air flow rate was equal to the minimum ventilation rate. In contrast, the reference system air flow rate was much greater than the minimum ventilation rate. Although the proposed system used more pump energy than the reference system because of the liquid desiccant circulation, the pump energy of the proposed system is less than that of the reference system. The chiller pump of the reference system consumed more energy than that required liquid desiccant circulation in the proposed system. Consequently, the proposed system consumed 27.0% less primary energy compared with the reference system, as shown in Fig. 4.

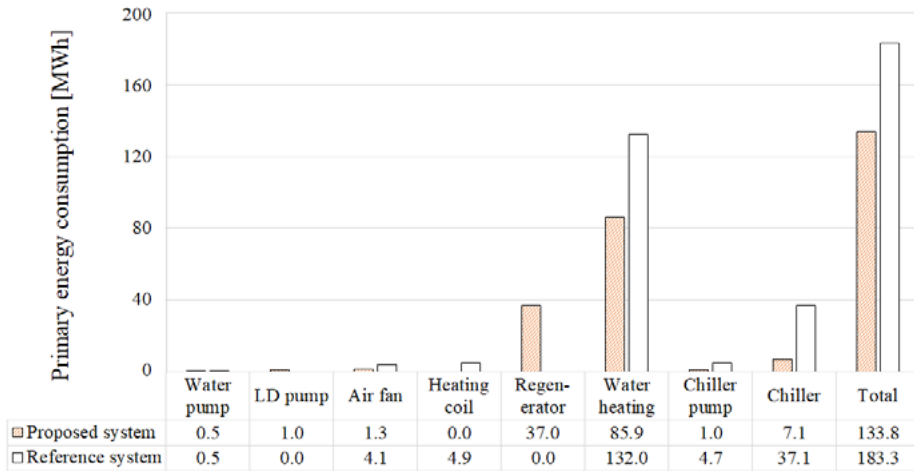


Fig. 4. Comparison of the primary energy consumptions of the two systems

5. Conclusion

This study proposed a liquid desiccant ventilation system that can provide both ventilation and water heating. The energy savings associated with the operation of the proposed system was quantitatively analyzed through comparison with a VAV reference system using a TRNSYS 18 and EES simulation. The integration of the liquid desiccant and water heating cycles was proposed to recover the heat required for water heating and reduce the cooling energy required for dehumidification. The possibility of maintaining the required supply-air conditions using an indirect evaporative cooler was investigated through a detailed energy simulation.

The operating energy consumption of the liquid desiccant ventilation system revealed that the proposed system can provide operating energy savings of 27.0% during the summer in comparison with the reference system. The potential for such significant energy savings is mainly due to the heat transferred from the liquid desiccant to the water heating cycle. This strategy also contributes to reducing the cooling energy required for the solution.

Considering the proposed system for use as a ventilation system, energy benefits were achieved in a hot and humid climate. Heat may be recovered from the solution to the water when the liquid desiccant and water heating cycles are operated simultaneously. This system should next be tested for use in dry and cold climates.

Acknowledgements

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