Energy Performance Comparison Between a Heat Pump-driven Liquid Desiccant System and Conventional Liquid Desiccant System

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ABSTRACT

Heat pump-driven liquid desiccant (HPLD) systems, which take advantage of the cooling and heating capacities from the evaporator and the condenser, respectively, have been developed rapidly in recent years. The purpose of this paper is to evaluate the energy saving potential of an HPLD system compared with a conventional liquid desiccant (LD) system. R134a was used for the refrigerant in the heat pump and an aqueous solution of LiCl was used in the LD system. A simulation based on an empirical model of the absorber was conducted, and total energy consumption in the proposed HPLD system was analyzed and compared with the conventional LD system. The results showed the proposed system had better performance in the aspect of energy savings than the conventional LD system. Moreover, HPLD is expected to have a great advantage in terms of compactness from the replacement of the cooling and heating devices in the conventional LD system with a heat pump.

KEYWORDS

Heat pump-driven liquid desiccant system, Liquid desiccant system, Heat pump, Energy performance

INTRODUCTION

According to the 2014 report by the Department of Energy (DOE), a liquid desiccant (LD) air conditioning system is one of the promising air-conditioning technologies for the next generation (Goetzler et al. 2014). The LD system can be operated with a low temperature heat source in regenerating the desiccant material. Regarding this feature, the LD system has a high potential for reducing energy consumption once combined with other low grade heat supply sources. Vibrant researches on hybrid liquid desiccant systems that use diverse types of heat supply systems to treat the loads in the dehumidifying and regenerating processes have been conducted. Among the many proposed systems, an LD system combined with a vapor compression heat pump system is currently being discussed and studied.

Previous studies on heat pump-driven liquid desiccant (HPLD) systems have focused mainly on system performance and capacity matching between required solution

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cooling and heating loads and heat pump-generated load. Zhang et al. (2012) determined that HPLD systems with additional condensers showed better performance than a basic HPLD system. Nie et al. (2012) concluded that solution flow rate and the air flow rate in the condenser, and the revolution of the compressor should be simultaneously controlled for capacity matching in the HPLD system.

In the present study, a quantitative analysis comparison on the energy consumption between a conventional LD system and the proposed HPLD system was conducted. The suggested HPLD system in this paper was comprised of a packed-bed type LD system and a vapor compression heat pump. A LiCl solution was adopted as the working solution in the LD system and R134a was chosen to be the refrigerant of the vapor compression heat pump. The simulation was conducted with the commercial engineering equation solver (EES) and the properties of air and LiCl solution embedded in the EES program were utilized for the simulation.

PROPOSED SYSTEM OVERVIEW

Liquid desiccant (LD) system

The two most important components of the LD system are an absorber and a regenerator. In the absorber, a concentrated solution (or strong solution) dehumidifies the process air. The diluted solution (or weak solution) leaving the absorber enters the regenerator to be regenerated to a strong solution by humidifying the regeneration air. These two processes of dehumidification and regeneration are repeated in the system, and the driving force of moisture transfer is the vapor pressure difference between the desiccant solution and the air passing through solution. Also, before entering the components, the solution is thermally treated in the heating or cooling coil for stable system performance.

Heat pump driven liquid desiccant (HPLD) system

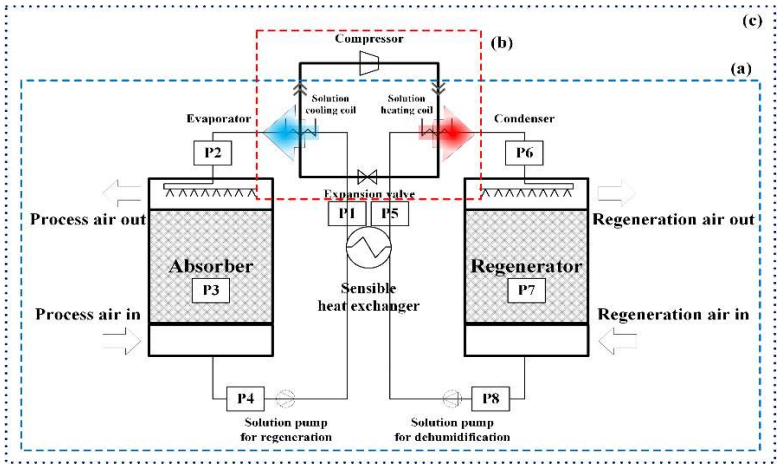

Figure 1. *Schematic of the heat pump-driven liquid desiccant (HPLD) system;(a)-LD, (b)-HP, (c)-HPLD*

The schematic of the suggested HPLD system is shown in figure 1-(c). The proposed HPLD system consisted of a conventional LD system (figure1-(a)) and a vapor

compression heat pump (figure1-(b)). An evaporator and a condenser in the heat pump were combined with the cooling coil and the heating coil, respectively. In other words, the cooling effect from heat absorption in the evaporator and heat release in the condenser were used to treat the solution cooling load in the absorber and the solution heating load in the regenerator, respectively. From the energy consumption point of view, the power input in a compressor of a heat pump in the HPLD system replaced the energy required in an air-cooled chiller and a boiler in a conventional LD system.

ENERGY SIMULATION OVERVIEW

In the energy simulation, energy consumption needed for the required load in the cooling coil and the heating coil to meet the set inlet temperature in absorber and regenerator was compared between the conventional system and the HPLD system. In the conventional system, an air-cooled chiller and gas boiler were used to treat the loads, whereas the evaporator and condenser in a heat pump treated the loads in HPLD system. The basic assumptions for the simulation were as follows. The air flow rate in the absorber and regenerator were set at 2000CMH and 4000CMH, respectively, to be constant flow. Additionally, the liquid-to-gas ratio was set at 1, meaning the solution flow rate was designed to be 0.67 kg/s. Furthermore, the solution inlet temperatures in the absorber and regenerator were maintained at 25 °C and 60 °C, respectively, and the concentration of the solution entering the absorber was set at 38%. Along with the above conditions, the TMY2 weather data of Seoul, South Korea for 744 h from August 1st to August 31st was applied to predict the performance of the LD system and evaluate the operating energy comparison between the conventional and suggested systems.

Liquid desiccant (LD) system

To compute the required cooling and heating loads, the temperature of the solutions leaving the absorber and regenerator must be determined. The calculation process required three solution conditions: solution mass flow rate, solution concentration and solution temperature, for every point from 1 to 8 as marked in the figure 1. The calculation also required three air conditions: air flow rate, air temperature, and air humidity ratio, before entering and after leaving the absorber and regenerator. The whole process of the condition change in the solution and the air were based on the mass balance equation and energy balance equation. To interpret the effectiveness of the absorber, the model created by Park et. Al. (2005) was adopted. The effectiveness of the regenerator was defined with the humidity ratio as well as the temperature of the regeneration air and the temperature of the solution in the regenerator.

(1) Absorber

The dehumidification process occurs as the process air passes through the strong solution of 38% at 25 °C inside the absorber. The effectiveness of the absorber was determined using a linear regression model created by Park et al. (2005). The model can be expressed with the following six operation parameters: air mass flow rate, solution mass flow rate, ambient air temperature, ambient air humidity ratio, solution inlet temperature, and solution concentration. Additionally, the effectiveness of the

absorber can be expressed with the temperature or the humidity ratio of the process air entering and leaving the absorber (Eq.(1) and Eq.(2)), through which air conditions leaving the absorber can be determined with Park et al.'s model. The equilibrium humidity ratio ($w_{abs,e}$) of the solution in the absorber can be determined using Eq.(3). A second order polynomial model suggested by Fumo and Goswami (2002) was used to obtain the vapor pressure at saturation condition of the desiccant solution in the absorber (Eq.(4)). The equilibrium temperature of the solution in the absorber was assumed the same as the temperature of the solution entering the absorber (25 °C).

$$\varepsilon_{abs} = \frac{w_{air,abs,in} - w_{air,abs,out}}{w_{air,abs,in} - w_{abs,e}} \quad (1)$$

$$\varepsilon_{abs} = \frac{T_{air,abs,in} - T_{air,abs,out}}{T_{air,abs,in} - T_{abs,e}} \quad (2)$$

$$w_{eq} = 0.622 \frac{p_s}{101.325 - p_s} \quad (3)$$

$$p_s = (a_0 + a_1 \cdot T_L + a_2 \cdot T_L^2) + (b_0 + b_1 \cdot T_L + b_2 \cdot T_L^2) \cdot C_L^2 + (c_0 + c_1 \cdot T_L + c_2 \cdot T_L^2) \cdot C_L^2 \quad (4)$$

The dehumidification rate ($\dot{m}_{p3,moi}$) in the absorber was obtained using the moisture mass balance equation (Eq.(5)). The solution mass flow rate and the concentration before entering the absorber were the same in p1, p2 and p8 (Eq.(6) and Eq.(7)), and those of the solution leaving the absorber were expressed in Eq.(8) and Eq.(9).
The temperature of the solution leaving the absorber was acquired from using the solution concentration from Eq.(9) and the solution enthalpy determined from the energy balance equation (Eq.(10)).

$$\dot{m}_{p3,moi} = \dot{m}_{air,abs,in} \cdot (w_{air,abs,in} - w_{air,abs,out}) \quad (5)$$
$$\dot{m}_{p1,sol} = \dot{m}_{p2,sol} = \dot{m}_{p8,sol} \quad (6)$$
$$C_{p1,sol} = C_{p2,sol} = C_{p8,sol} \quad (7)$$
$$\dot{m}_{p4,sol} = \dot{m}_{p1,sol} + \dot{m}_{p3,\mathrm{moi}} \quad (8)$$
$$\dot{m}_{p1,sol} \cdot C_{p1,sol} = \dot{m}_{p4,sol} \cdot C_{p4,sol} \quad (9)$$
$$\dot{m}_{p4,sol} \cdot h_{p4,sol} = \dot{m}_{p2,sol} \cdot h_{p2,sol} + \dot{m}_{p3,moi} \cdot h_{fg_{p3},moi} \quad (10)$$

(2) Regenerator
The solution concentration leaving the absorber becomes lower compared to that in the entering state, which has to be regenerated in the regenerator for the performance of the LD system. For balance in overall system, the regeneration rate in the regenerator was assumed to be same as the dehumidification rate in the absorber, through which the humidity ratio of the air leaving the regenerator was obtained using Eq.(11) and Eq.(12).

$$\dot{m}_{p3,moi} = \dot{m}_{p7,moi} \quad (11)$$
$$\dot{m}_{p7,moi} = \dot{m}_{air,reg,in} \cdot (w_{air,reg,out} - w_{air,reg,in}) \quad (12)$$

The regeneration effectiveness (ε_{reg}) was defined with the humidity ratio change or temperature change in regeneration air (Eq.(13) and Eq.(14)). The regeneration rate

from Eq.(12) was used in Eq.(13) to calculate the regeneration effectiveness (ε_{reg}) in Eq.(14) to obtain the temperature of the regeneration air leaving the regenerator ($T_{SA,state2}$). The equilibrium ratio in the regenerator ($w_{reg,eq}$) was determined using Eq.(3) and Eq.(4). The equilibrium temperature of the solution in the regenerator was assumed the same as the temperature of the solution entering the regenerator (60 °C).

$$\varepsilon_{reg} = \frac{w_{air,reg,out} - w_{air,reg,in}}{w_{reg,e} - w_{air,reg,in}} \quad (13)$$

$$\varepsilon_{reg} = \frac{T_{air,reg,out} - T_{air,reg,in}}{T_{reg,e} - T_{air,reg,in}} \quad (14)$$

The solution mass flow rate and the concentration before entering the regenerator were the same in p4, p5 and p6 (Eq.(15) and Eq.(16)), and those of the solution leaving the regenerator expressed in Eq.(17) and Eq.(18).
The temperature of the solution leaving the regenerator was acquired using the solution concentration from Eq.(18) and the solution enthalpy from Eq.(19).

$$\dot{m}_{p4,sol} = \dot{m}_{p5,sol} = \dot{m}_{p6,sol} \quad (15)$$
$$C_{p4,sol} = C_{p5,sol} = C_{p6,sol} \quad (16)$$
$$\dot{m}_{p8,sol} = \dot{m}_{p6,sol} - \dot{m}_{p7,\mathrm{moi}} \quad (17)$$
$$\dot{m}_{p6,sol} \cdot C_{p6,sol} = \dot{m}_{p8,sol} \cdot C_{p8,sol} \quad (18)$$
$$\dot{m}_{p6,sol} \cdot h_{p6,sol} = \dot{m}_{p8,sol} \cdot h_{p8,sol} + \dot{m}_{p7,moi} \cdot h_{fg_{p7},moi} \quad (19)$$

(3) Solution cooling and heating loads
The temperature of the solutions leaving the absorber and regenerator ($T_{p4,sol}$, $T_{p8,sol}$) from Eq.(1) through Eq.(19) were used to estimate the cooling and heating loads in the cooling and heating coils. The required loads indicated the power needed to meet the solution inlet set-point temperature in the absorber and the regenerator ($T_{p2,sol} = 25$ °C, $T_{p6,sol} = 60$ °C). The solutions leaving the absorber and the regenerator first exchange the sensible heat in terms of heat recovery before entering the coils. In this simulation, the efficiency of the sensible heat exchanger (ε_{SHE}) was 80%. The temperature of the solutions leaving the heat exchanger ($T_{p1,sol}$, $T_{p5,sol}$) was determined from Eq.(20) and Eq.(21). The required loads in the cooling coil and the heating coil were estimated using Eq.(22) and Eq.(23), respectively.

$$\varepsilon_{SHE} = \frac{T_{p8,sol} - T_{p1,sol}}{T_{p8,sol} - T_{p4,sol}} = 0.8 \quad (20)$$

$$\dot{m}_{p8,sol} \cdot C_{p_{p8},sol} \cdot (T_{p8,sol} - \mathrm{T_{p1,sol}}) = \dot{m}_{p4,sol} \cdot C_{p_{p4},sol} \cdot (T_{p5,sol} - \mathrm{T_{p4,sol}}) \quad (21)$$
$$Q_{cooling} = \dot{m}_{p1,sol} \cdot C_{p_{p1},sol} \cdot (T_{p1,sol} - \mathrm{T_{p2,sol}}) \quad (22)$$
$$Q_{heating} = \dot{m}_{p5,sol} \cdot C_{p_{p5},sol} \cdot (T_{p6,sol} - \mathrm{T_{p5,sol}}) \quad (23)$$

Conventional liquid desiccant system with air-cooled chiller and gas boiler
In the conventional LD system, the air-cooled chiller and boiler are generally used to treat the required cooling and heating loads. In this study, the required loads were

simulated using the TRANE chiller CGAM35 model and gas boiler model in EnergyPlus (US DOE 2010).

Heat pump driven liquid desiccant system

In the HPLD system, the cooling capacity from the evaporator and the heating capacity from the condenser were used to manage the required solution cooling and heating loads. The heat pump model used in the simulation was a parameter estimation-based water-to-water curve-fit model developed by Jin (2002) and implemented in EnergyPlus. The model is provided in two different modes: cooling mode and heating mode. In each mode, the capacity in the source side and load side and the power input in the compressor can be estimated from the following four variables: source side volumetric flow rate, source side inlet temperature, load side volumetric flow rate and load side inlet temperature. In this paper, the heat pump was simulated in the cooling mode due to its focus on the required solution cooling load for meeting the solution inlet temperature in the absorber where the purpose of the LD system occurs.

ENERGY SIMULATION RESULTS

Air and solution conditions leaving the absorber are determined by the heat and mass transfer between the air and the solution entering the absorber and the heat generated during the dehumidification process, which are influenced by the properties of the air and the solution. In this simulation, since the temperature of the solution entering the absorber (25 °C) was relatively lower than the temperature of the process air entering the absorber, the outlet temperature of the solution tended to increase from the heat exchange with the process air and the absorption of the generated heat in the absorber. Additionally, the constant solution flow rate entering the absorber, increased by the amount of dehumidification rate after the dehumidification process was completed.

Energy consumption comparison

(1) The conventional LD system with an air-cooled chiller and gas boiler

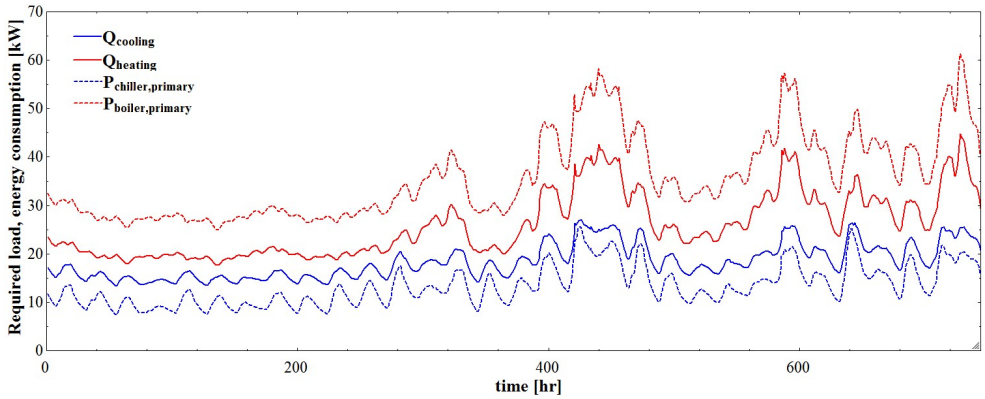

Figure 2. *Required solution cooling and heating load, and energy consumption in air cooled chiller and gas boiler*

Figure 2 shows the energy consumption of the air-cooled chiller and gas boiler in the conventional LD system according to the required solution cooling and heating loads. As shown in figure 2, the heating load was generally higher than the cooling load.

(2) Heat pump driven liquid desiccant system

Figure 3 shows the energy consumption of the heat pump and the auxiliary boiler in the HPLD system according to the required solution cooling and heating loads. In the simulation, the heat pump was driven in the cooling mode considering the part load ratio with the focus on matching the generated cooling capacity in the evaporator with the required cooling load. However, the generated heating capacity ($Q_{source,c,PLR}$) according to the cooling capacity generated in the part load ratio operation and the compressor energy could not meet the required heating load ($Q_{heating}$). Accordingly, an additional heating source to provide the insufficient heating load was needed in which an auxiliary gas boiler was used in this simulation. Generally, a gas boiler is not appropriate to treat such a fluctuating load as shown in figure 3. However, in this simulation, a gas boiler was used for auxiliary heating in the HPLD system to meet the same condition for using a heating source as the conventional LD system.

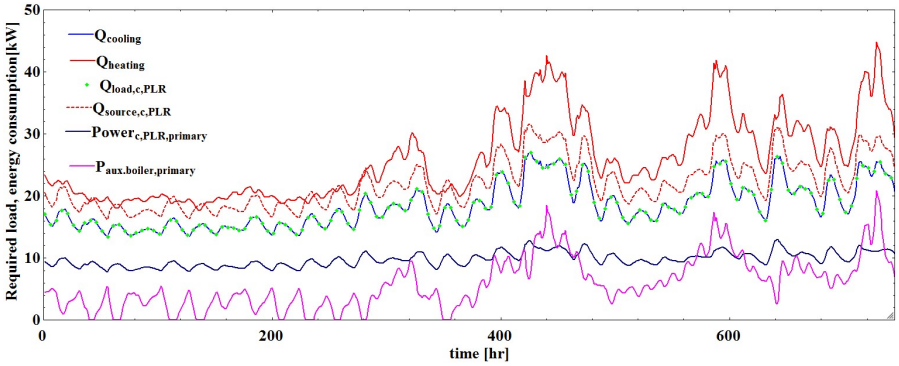

Figure 3. *Required solution cooling and heating load and energy consumption in the heat pump and auxiliary boiler*

(3) Total energy consumption comparison between the conventional LD and HPLD systems

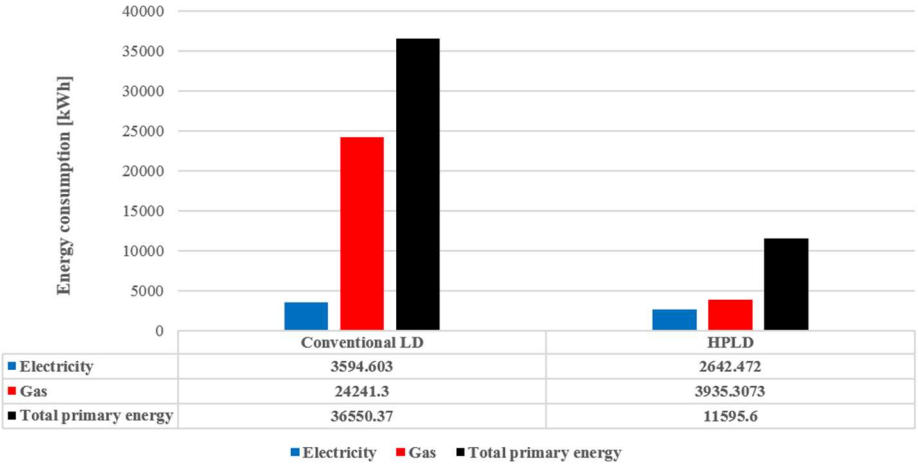

Figure 4. *Total energy consumption comparison between conventional LD and HPLD systems*

Figure 4 shows the energy consumption comparison between the conventional LD system and the proposed HPLD system to treat the required solution cooling and heating loads. When comparing the total energy consumption, the primary energy

conversion factors of 2.75 for electricity and 1.1 for gas were used. Electricity was used to operate the heat pump and the air-cooled chiller in the HPLD system and the conventional LD system, respectively. The HPLD system showed 26% less electricity consumption compared to the conventional LD system. Significant energy consumption difference was observed in the gas energy consumption. In the HPLD system, only the insufficient heating load after treated with the heating capacity from the condenser in the heat pump was processed with the auxiliary gas boiler, whereas in the conventional LD system, the total required heating load was treated with the gas boiler. As a result, the gas energy consumption in the HPLD system showed 84% reduction compared with the conventional LD system. Consequently, the proposed system showed 68% savings in total energy consumption.

CONCLUSION AND DISCUSSION

In this study, energy consumption to treat the required solution cooling and heating load in the conventional LD system, which uses a chiller and a gas boiler, and in the HPLD system, which uses a heat pump, were compared. The proposed HPLD system showed significant energy savings compared with the conventional LD system: 26% reduction in the electric energy, 84% reduction in the gas energy and 68% reduction in the total primary energy. Once the capacity matching between the generated load in the heat pump and the required load for solution cooling and heating is achieved, no further auxiliary boiler usage will be needed. Furthermore, the proposed HPLD system is expected to have an advantage in terms of system compactness.

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