Energy Simulation of a Heat Pump-driven Liquid Desiccant System Using Dynamic Analysis

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Abstract

A heat pump driven liquid desiccant (HPLD) system, a highly energy efficient system in terms of waste heat recovery utilizing the cooling capacity of an evaporator and the heating capacity of a condenser. In this study, we assess and compare the energy saving potential of an HPLD system with a conventional liquid desiccant (LD) system. A packed-bed tower type LD system with an aqueous solution of lithium chloride (LiCl) and a water-to-water heat pump with a refrigerant of R134a were used in this study. The performance of the LD system was interpreted in an engineering equation solver (EES) and the heat pump was simulated through TRNSYS 17, a dynamic analysis program. When the total energy consumption of the proposed HPLD system and the conventional LD system are analyzed and compared, the HPLD system showed better performance than the conventional LD system in both aspects of COP and energy saving.

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Keywords: Underground spaces; liquid desiccant and evaporative cooling-assisted 100% outdoor air system (LD-IDECOAS); ground source heat pump (GSHP)

1. Nomenclature

T	Temperature [°C]
w	Humidity ratio [kg/kg]
Q	Thermal load [W]
$\dot{m}$	Mass flow rate [kg/s]
P	Vapor pressure [kPa]
C	Concentration [-]
h	Enthalpy [kJ/kg]
h_{fg}	Heat of vaporization of water [=2257 kJ/kg]
a0 – a2, b0 – b2, c0 – c2	Coefficients of vapor pressure equation

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A1 – A5, B1 – B5	Coefficients of water-to-water heat pump model for cooling mode

Greek symbols

ε	Effectiveness [-]

Subscripts

p1 - p8	Designated points in the liquid desiccant system
e	Equilibrium
moi	Moisture
abs	Absorber
reg	Regenerator
sol	Desiccant solution
in	Inlet
out	Outlet
load	Load side
source	Source side
c	Cooling mode

Abbreviations

COP	coefficient of performance
LD	liquid desiccant
LiCl	lithium chloride
HP	Heat pump
HPLD	Heat-pump-driven liquid desiccant
SHE	sensible heat exchanger

2. Introduction

According to the 2014 report by the Department of Energy (DOE), liquid desiccant (LD) air conditioning system applied HVAC technologies are ranked in high stage for highly promising air-conditioning options for the next generation in terms of their energy saving potential [1]. The liquid desiccant system, which removes the moisture content in the outdoor air (i.e. process air) has its own unique feature of requiring heating and cooling sources at the same time for its effective operation and stable system performance. Regarding this feature, researchers have conducted vibrant studies to determine adequate heat supply sources for the increase in the system's performance. Among various proposed systems, the integration of a vapor compression heat pump system with an LD system is being actively discussed and investigated lately.

Previous studies of heat-pump-driven liquid desiccant (HPLD) systems have focused primarily on energy consumption, overall system performance, and capacity matching between the required solution cooling load, heating load, and heat-pump-generated load. Yamaguchi et al. [2] conducted a performance evaluation and discussed methods for improving system efficiency via mathematical calculations. They concluded that the coefficient of performance (COP) could be increased by enhancing the isentropic efficiency of the compressor and solution heat exchanger. Bergero and Chiari [3] evaluated the system performance of HPLD through simulation. In their study, they suggest that the HPLD system is driven with a hygroscopic solution and a hydrophobic membrane. The simulation results indicate that the proposed hybrid system can lead to energy savings. Zhang et al. [4] focused on methods for effectively removing the heat left over after regenerating solution. Two different methods for exhausting the extra heat are suggested which are utilizing either an air-cooled assistant condenser or a water-cooled assistant condenser, and the results are compared from the perspective of COP. They found out that systems with air-cooled condenser and water-cooled condenser exhibit better performance compared to a basic HPLD system with no assistant condenser with COP values approximately 18% and 35% higher, respectively. Niu et al. [5] suggested methods for matching the capacity of four major heat and mass transfer components in the HPLD, which are absorber, regenerator, evaporator, and condenser. The results indicate that the solution flow rate in the condenser, revolution of the compressor, and air flow rate in the air-cooled condenser should be simultaneously controlled for capacity matching in the HPLD system.

In this study, the energy consumption between conventional liquid desiccant system and proposed HPLD system have been analyzed and compared quantitatively. The suggested HPLD system in this paper is comprised of a counter-flow packed-bed type liquid desiccant system and a vapor compression heat pump. Lithium chloride (LiCl) solution is adopted as the working solution in the liquid desiccant system and R-134a is chosen as the refrigerant of the vapor compression heat pump. Also the evaporator and the condenser in the heat pump is connected with cooling coil and heating coil in the liquid desiccant system respectively right after sensible heat exchanger to treat the required solution cooling and heating load. The simulation for performance and required load of the liquid desiccant system is interpreted using the commercial engineering equation solver (EES). Based on these results, a water-to-water heat pump model in the TRNSYS17 is used to simulate the proposed system. The thermal properties of the air and LiCl solution embedded in the EES program are utilized for the simulation.

3. Proposed system overview

3.1. Liquid Desiccant (LD) system

A liquid desiccant (LD) system is used to treat the latent load of the outdoor air before it is transferred indoors. A typical LD system is composed of an absorber, a regenerator, a sensible heat exchanger, a heating coil, and a cooling coil, as shown in Figure 1. In the absorber, a concentrated solution (i.e., strong solution) dehumidifies the process air that passes through the absorber. The diluted solution (i.e., weak solution) leaving the absorber enters the regenerator to be regenerated to strong solution by humidifying the regeneration air that passes through the regenerator. The two processes of dehumidification and regeneration are repeated in the LD system, and the driving force of moisture transfer in the system is the difference in vapor pressure between the desiccant solution and the air passing through the solution. Namely, the direction of vapor pressure difference is the determinant for which process between dehumidification and regeneration will take place [6]. Solution temperature is one of the main key factors s for effective operation of the liquid desiccant system (or stable system performance), making thermal treatment of the solution indispensable. Generally, desiccant solution should be heated to 45 – 80 °C for the regeneration process, whereas it should be cooled to 15 – 30 °C for the dehumidification process [7-8]. To meet the required solution temperature, the solutions leaving the absorber and regenerator are firstly met in the sensible heat exchanger in terms of recovering the heat. Solutions leaving the sensible heat exchanger are heated and cooled in the cooling and the heating coils, in which a gas boiler and chiller are generally used to treat the required loads in the coils for a conventional liquid desiccant system. Dehumidification and regeneration units can be sorted into two types based on the heat extraction process: adiabatic and internally cooled type. In this study, a packed-bed tower type is used; which is the most commercialized among adiabatic types [9].

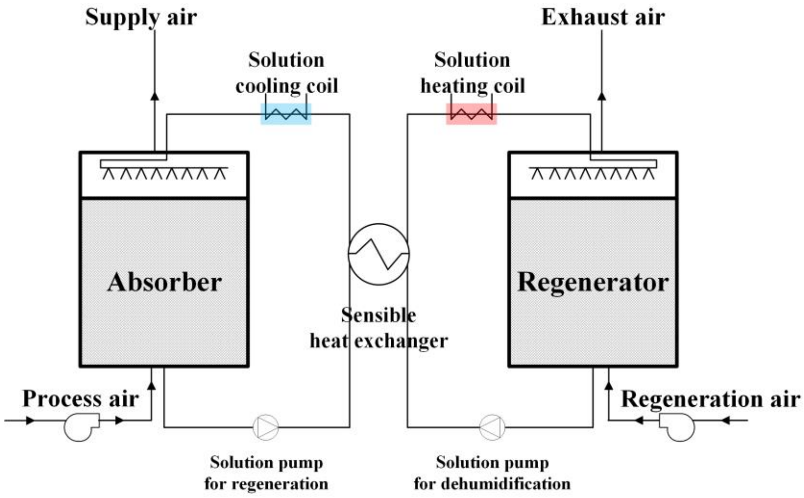


Fig. 1. Schematic diagram of the liquid desiccant (LD) system

3.2. Heat pump (HP) system

A heat pump, a vapor compression refrigeration system, utilizes the reverse flow of the basic existing heat transfer law (i.e., the second law of thermodynamics) which is that heat cannot be transferred from a cooler to a hotter body spontaneously [10]. In other words, the heat pump transports the heat from a lower temperature heat source to a higher temperature heat sink. The system is comprised of an evaporator, a compressor, a condenser, and an expansion valve. In the evaporator, the refrigerant flowing in the heat pump absorbs the heat as it changes its phase from liquid to gas (i.e., evaporation), whereas in the compressor the reverse phenomenon (i.e., condensation) takes place. Additionally, the refrigerant is pressurized in a compressor to a higher temperature and high pressure state in order to increase the energy level to a higher one. To drive this thermodynamic refrigeration cycle in a heat pump, power input in a compressor from an external energy source is required [11].

3.3. Heat pump driven liquid desiccant (HPLD) system

Figure 2 shows a schematic of the proposed HPLD system, which is comprised of a conventional liquid desiccant system and a vapor compression heat pump. The condenser and evaporator of the heat pump are directly integrated with the heating and cooling coils, respectively, to utilize the heating and cooling load generated in the heat pump. Namely, the heat release in the condenser and the cooling effect from the heat absorption in the evaporator are used simultaneously to treat the solution heating load in the regenerator and solution cooling load in the absorber, respectively. In terms of energy consumption, the energy requirement in an air-cooled chiller and a boiler in a conventional liquid desiccant system can be replaced with the power input in the compressor of a heat pump in an HPLD system. Furthermore, the feature of using waste heat in the compression heat pump can enhance the efficiency of the HPLD system.

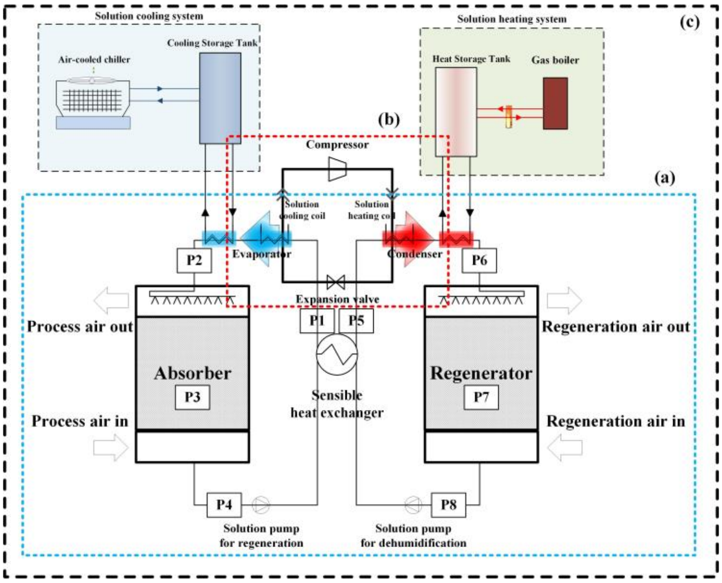


Fig. 2. Schematic diagram of the heat pump driven liquid desiccant (HPLD) system;
(a) – LD, (b) – HP, (c) – HPLD with auxiliary devices

4. Energy Simulation Overview

In the energy simulation, the energy consumption needed to treat the required load in the cooling and heating coils to match the set temperature of the solution entering the absorber (25 °C) and the regenerator (60 °C) is compared between the conventional system and the proposed HPLD system. The basic assumptions for the simulation are as follows: the air flow rate in the absorber and the regenerator are set at a constant 2000 and 4000 cubic meters per hour, respectively. Additionally, liquid to gas ratio is set at 1 in the absorber, which means that the solution flow rate is designed to be 0.67 kg/s from the constant air flow rate. Moreover, the solution inlet temperature in the absorber and the regenerator are maintained at 25 °C and 60 °C, respectively, and the concentration of the solution entering the absorber is set at 38%. Along with the above conditions, the international weather for energy calculations (IWEC) weather data of Seoul, South Korea, for 744 hours from August 1st to August 31st from ASHRAE [12], is applied to predict the performance of the liquid desiccant system and evaluate the operating energy comparison between the conventional and suggested systems.

4.1. Liquid desiccant (LD) system

Table 1. Coefficients of vapor pressure equation

Coefficients of vapor pressure equation			
Dehumidification process	a_0	a_1	a_2
	4.58208	-0.159174	0.0072594
	b_0	b_1	b_2
	-18.3816	0.5661	-0.019314
	c_0	c_1	c_2
	21.312	-0.666	0.01335
Regeneration process	a_0	a_1	a_2
	16.294	-0.8893	0.01927
	b_0	b_1	b_2
	74.3	-1.8035	-0.01872
	c_0	c_1	c_2
	-226.4	7.49	-0.039

The temperature of the solutions leaving the absorber and the regenerator must be found in order to compute the required cooling and heating loads [13]. In the calculation process, three solution conditions: the solution mass flow rate, solution concentration, and solution temperature for every point from 1 to 8 as marked in the Figure 2 and three air conditions: the air flow rate, air temperature, and air humidity ratio before entering and after leaving the absorber and regenerator are needed. The entire process of the condition changes in the solution and the air are based on the mass balance and energy balance equations. To interpret the effectiveness of the absorber, the model created by Park [14] is adopted among the several existing models. The effectiveness of the regenerator is defined with the humidity ratio and the temperature of the regeneration air, and the temperature of the solution in the regenerator. Furthermore, the efficiency of the sensible heat exchanger in this simulation is assumed to be 80 %.

4.1.1 Absorber

The dehumidification process occurs as the process air is brought into contact with the strong solution of 38 % and 25 °C inside the packed-bed type absorber. Several models have been established in current literature to interpret the performance of the absorber and predict the dehumidification effectiveness ($\varepsilon_{\mathrm{abs}}$). In this research, a parameter-estimation-based linear regression model developed by Park is selected. The empirical model is a function of six operating parameters: air mass flow rate ($\dot{m}_{OA,state1}$), ambient air temperature ($\mathrm{T}_{OA,state1}$), ambient air humidity ratio ($\mathrm{w}_{OA,state1}$), solution mass flow rate ($\dot{m}_{p2,sol}$), solution inlet temperature ($\mathrm{T}_{p2,sol}$), and solution concentration ($\mathrm{C}_{p2,sol}$). Additionally, the effectiveness of the absorber can be defined with the temperature change or humidity ratio change of the process air entering and leaving the absorber (Eq. (1), Eq. (2)). Once the empirical model is combined with the definitions, one can estimate the air conditions leaving the absorber. The equilibrium humidity ratio of the solution entering the absorber ($\mathrm{w}_{\mathrm{abs,e}}$) can be obtained with the function of the vapor pressure (p_s) at the saturation condition of the desiccant solution in the absorber (Eq. (3)), in which a second-order polynomial model suggested by Fumo and Goswami [15] was used (Eq. (4)). The constants needed in the model are presented in table 1. The equilibrium temperature of the solution in the regenerator is assumed to be the same as the temperature of the solution entering the regenerator, which is 25 °C.

$$\varepsilon_{abs} = \frac{w_{air,abs,in} - w_{air,abs,out}}{w_{air,abs,in} - w_{abs,eq}} \tag{1}$$

$$\varepsilon_{abs} = \frac{T_{air,abs,in} - T_{air,abs,out}}{T_{air,abs,in} - T_{abs,eq}} \tag{2}$$

$$w_{eq} = 0.622\,\frac{p_s}{101.325 - p_s} \tag{3}$$

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

The mass and energy balance around the absorber can be expressed via Eq. (5) to Eq. (10), through which solution conditions leaving the absorber can be obtained. The dehumidification rate ($\dot{m}_{p3,sol}$) in the absorber can be acquired using the moisture bass balance equation (Eq. (5)). The solution conditions (mass flow rate and concentration) before entering the absorber are the same in p1, p2, and p8 (Eq. (6), Eq. (7)), regardless of the presence of a sensible heat exchanger or a cooling coil. Also, those of the solution leaving the absorber can be obtained using Eq. (8) and Eq. (9).

$$\dot{m}_{p3,moi} = \dot{m}_{air,abs,in}(w_{air,abs,in} - w_{air,abs,out}) \tag{5}$$

$$\dot{m}_{p1,sol} = \dot{m}_{p2,sol} = \dot{m}_{p8,sol} \tag{6}$$

$$C_{p1,sol} = C_{p2,sol} = C_{p8,sol} \tag{7}$$

$$\dot{m}_{p4,sol} = \dot{m}_{p1,sol} + \dot{m}_{p3,moi} \tag{8}$$

$$\dot{m}_{p1,sol} \cdot C_{p1,sol} = \dot{m}_{p4,sol} \cdot C_{p4,sol} \tag{9}$$

The solution temperature leaving the absorber can be acquired through the LiCl properties embedded in the EES with the solution concentration from Eq. (9), and the solution enthalpy, which can be determined from the energy balance equation (Eq. (10)).

$$\dot{m}_{p4,sol} \cdot h_{p4,sol} = \dot{m}_{p2,sol} \cdot h_{p2,sol} + \dot{m}_{p3,moi} \cdot h_{fg_{p3},moi} \tag{10}$$

4.1.2 Regenerator

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

$$\dot{m}_{p3,moi} = \dot{m}_{p7,moi} \tag{11}$$

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

The regeneration effectiveness (ε_{reg}) can be defined with the change in humidity ratio or the change in temperature of the regeneration air (Eq. (13), Eq. (14)). The regeneration rate from Eq. (12) can be used in Eq. (13) to calculate the regeneration effectiveness (ε_{reg}), which can be used in Eq. (14) to obtain the temperature of the regeneration air leaving the regenerator ($T_{SA,state2}$). The equilibrium ratio in the regenerator ($T_{OA,state1}$) can be determined using Eq. (3) and Eq. (4). The equilibrium temperature of the solution in the regenerator is assumed to be the same as the temperature of the solution entering the regenerator, which is 60 °C.

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

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

The solution mass flow rate and the solution concentration before entering the regenerator are the same in p4, p5, and p6 (Eq. (15) and Eq. (16)), and those of the solution leaving the regenerator can be expressed as Eq. (17) and Eq. (18).

$$\dot{m}_{p4,sol} = \dot{m}_{p5,sol} = \dot{m}_{p6,sol} \tag{15}$$

$$C_{p4,sol} = C_{p5,sol} = C_{p6,sol} \tag{16}$$

$$\dot{m}_{p8,sol} = \dot{m}_{p6,sol} - \dot{m}_{p7,moi} \tag{17}$$

$$\dot{m}_{p6,sol} \cdot C_{p6,sol} = \dot{m}_{p8,sol} \cdot C_{p8,sol} \tag{18}$$

The outlet solution temperature in the regenerator can be estimated with the solution concentration from Eq. (18) and the solution enthalpy, which can be determined from the energy balance equation (Eq. (19)).

$$\dot{m}_{p6,sol} \cdot h_{p6,sol} = \dot{m}_{p8,sol} \cdot h_{p8,sol} + \dot{m}_{p7,moi} \cdot h_{fg_{p7},moi} \tag{19}$$

4.1.3 Solution cooling and heating load

The solution cooling and heating load in the cooling and heating coil can be estimated with the temperature of the solutions leaving the sensible heat exchanger. The temperature of the solutions leaving the absorber and the regenerator ($T_{p4,sol}$, $T_{p8,sol}$) from Eq. (1) to Eq. (19) can be used to estimate the cooling and heating loads in the cooling and heating coils. The required loads indicate 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 cooling and heating coils. In this simulation, the efficiency of the sensible heat exchanger (ε_{SHE}) is 80 %. The temperature of the solutions leaving the heat exchanger ($T_{p4,sol}$, $T_{p8,sol}$) can be determined from Eq. (20) and Eq. (21). The required load in the cooling and heating coils can be estimated using Eq. (22) and Eq. (23).

$$\varepsilon_{reg} = \frac{T_{air,reg,out} - T_{air,reg,in}}{T_{reg,e} - T_{air,reg,in}} \tag{20}$$

$$\dot{m}_{p8,sol} \cdot C_{p_{p8},sol} \cdot (T_{p8,sol} - T_{p1,sol}) = \dot{m}_{p4,sol} \cdot C_{p_{p84},sol} \cdot (T_{p5,sol} - T_{p4,sol}) \tag{21}$$

$$Q_{cooling} = \dot{m}_{p1,sol} \cdot C_{p_{p1},sol} \cdot (T_{p1,sol} - T_{p2,sol}) \tag{22}$$

$$Q_{heating} = \dot{m}_{p5,sol} \cdot C_{p_{p5},sol} \cdot (T_{p6,sol} - T_{p5,sol}) \tag{23}$$

4.2. Conventional liquid desiccant system with air-cooled chiller and gas boiler

In the conventional liquid desiccant system, among various types of heating and cooling sources, air cooled chillers and boilers are commonly used to treat the required cooling and heating loads. In this paper, a TRANE chiller CGAM35 model and gas boiler model furnished in EnergyPlus [16] were adopted to compute the energy consumptions to treat the required loads.

4.3. Heat pump driven liquid desiccant (HPLD) system

In the HPLD system, both the cooling capacity from the evaporator and the heating capacity from the condenser in a heat pump are used, as opposed to using separate devices for treating the required loads in the conventional LD system. The heat pump performance was simulated with a parameter estimation based water-to-water curve-fit model implanted in TRNSYS 17, a transient analysis program. In the simulation, type927, a single stage water-to-water heat pump model was used. The model treats the main liquid stream line (the solution that flows through the evaporator in this stimulation) by rejecting energy to (cooling mode) or absorbing energy from (heating mode) a second liquid stream (the solution that flows through the condenser). It is operated in an on/off control with temperature level changes much like an actual heat pump does; once the user defined control signal input is given to be ON in either cooling or heating mode, the system is operated at its capacity level until there's a change in the signal value. The model can be used in two different modes: cooling and heating mode. In each mode, the capacity in the source and load sides and the power input in the compressor can be computed with the following four input parameters: source side flow rate, source side inlet temperature, load side flow rate, and load side inlet temperature. In this research, the heat pump was sized for a rated cooling capacity of 16 kW and was operated in the cooling mode to focus on the demanded solution cooling load to meet the temperature of the solution entering the absorber, where the purpose of the LD system occurs. Therefore, in this simulation, the load side and source side capacities are the load generated in the evaporator and condenser, respectively. Also, the estimated power input indicates the energy needed to operate the heat pump system.

5. Energy Simulation Results

Air and solution condition 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 (i.e., exothermic reaction) which are influenced by the properties of the air and the solution. In this simulation, due to the relatively lower temperature of the solution entering the absorber (25 °C) compared to the temperature of the process air entering the absorber, the solution temperature leaving the absorber rises from the heat exchange with the process air and the absorption of the generated heat in the absorber. Also, the solution flow rate, which is maintained at a constant 0.67 kg/s when entering the absorber, increased by the amount of the dehumidification rate due to the moisture absorption from the process air.

5.1. Energy consumption in the conventional liquid desiccant system

Figure 3 shows the required cooling and heating load and the energy consumption of the air-cooled chiller and gas boiler in the conventional LD system in accordance with the required solution cooling and heating load. One can see that the heating load is generally higher than the cooling load.

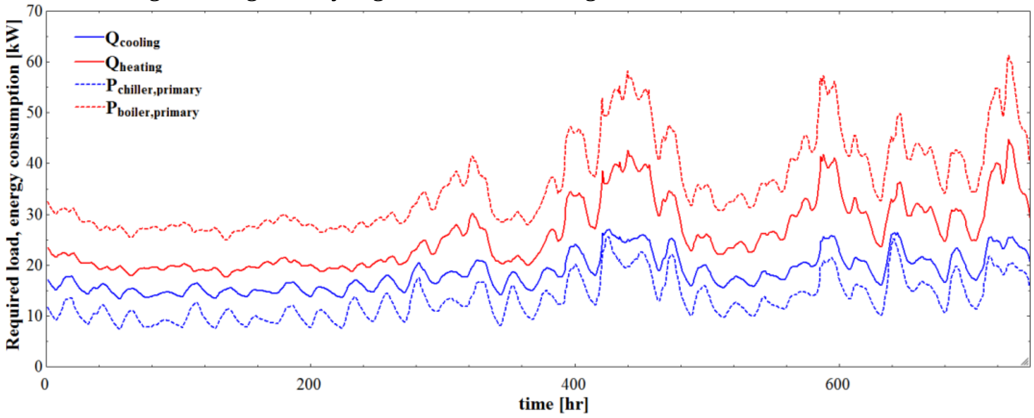


Fig. 3. Required load and energy consumption in the conventional LD system

5.2. Required heating and cooling load for LD operation and heat pump generated capacity

Figure 4 shows the required cooling and heating load and heat pump generated load. One can see that there's quite a difference between the demanded load and the generated load which was caused due to the on/off control in the heat pump. In the proposed system, a control strategy that treats most of the required load with a primary system (i.e., the heat pump) with full speed control and the rest of the load with supplementary systems (i.e., auxiliary devices) was used: the heat pump was operated with an on/off control at full speed and the insufficient cooling and heating load were treated with an auxiliary air-cooled chiller and an auxiliary gas boiler, respectively.

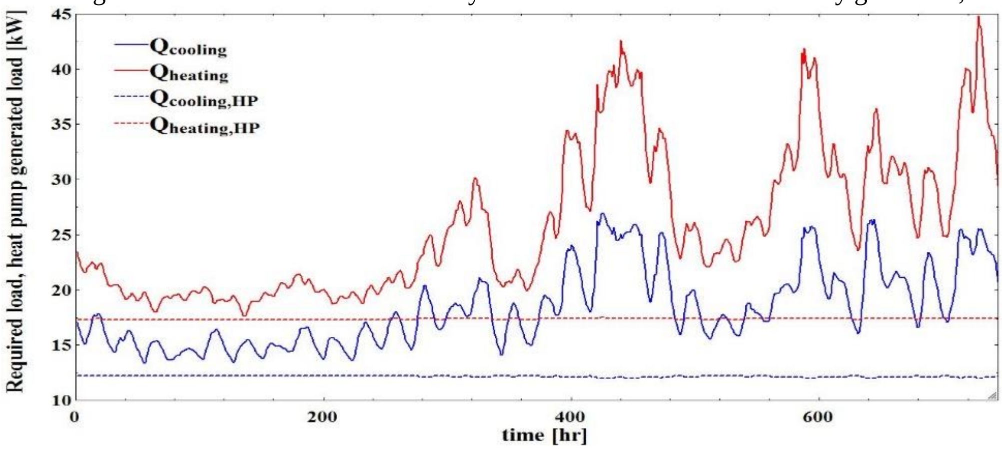
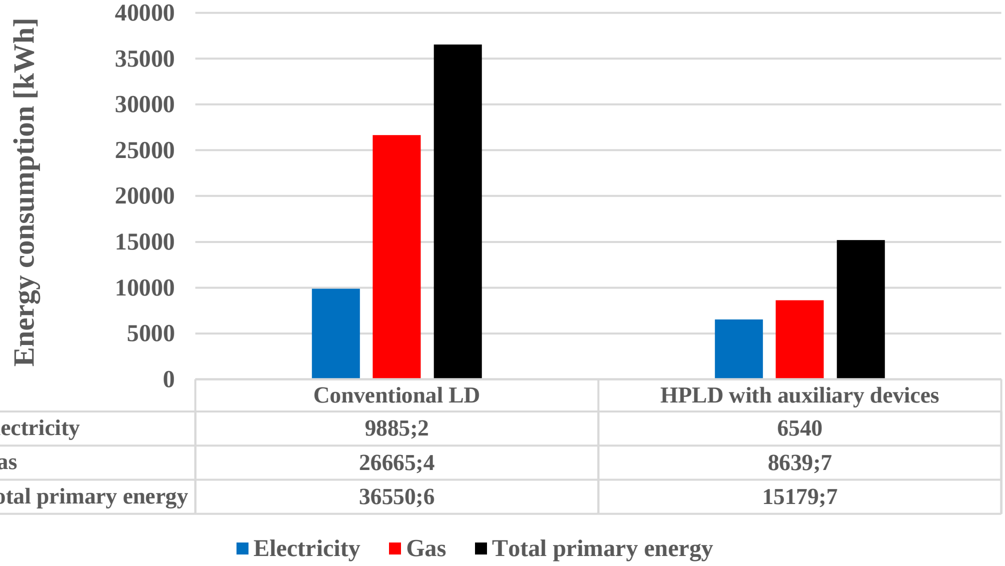


Fig. 4. Required load and heat pump generated load in the HPLD system

5.3. Total energy consumption comparison between conventional LD system and HPLD system

Figure 5 shows a comparison of the energy consumption between the conventional LD system and the proposed HPLD system when treating the required solution cooling and heating loads. When comparing the total energy consumption, the primary energy conversion factors, for which 2.75 for electricity and 1.1 for gas were used [17]. In the conventional system, electricity and gas were used to operate the air-cooled chiller and the gas boiler, respectively. Whereas in the HPLD system, electricity and gas were used in the heat pump in the auxiliary chiller and auxiliary gas boiler, respectively. The HPLD system showed 34 % less electricity consumption than the conventional LD system, of which 68 % was used for the heat pump and 32 % was used for the auxiliary air-cooled chiller. Furthermore, the HPLD system showed 68 % reduction in gas energy consumption when compared to the conventional system. Due to the on/off control in the heat pump, capacity matching could not be made between the heat pump generated load and the required load, making the need for auxiliary device operation inevitable. As a result, the HPLD system showed 58 % reduction in the total energy consumption when compared to the conventional LD system. The significant energy saving is due to the reduction in energy from the gas boiler, which takes the majority of the total energy consumption in the liquid desiccant system. In the HPLD system only the insufficient heating load, or that left after the solution is treated with the heating capacity from the condenser in the heat pump, is processed with the auxiliary gas boiler. Whereas in the conventional LD system the total required heating load is treated with the gas boiler



	Conventional LD	HPLD with auxiliary devices
Electricity	9885;2	6540
Gas	26665;4	8639;7
Total primary energy	36550;6	15179;7

Fig. 5. Energy consumption comparison between conventional LD and HPLD systems

6. Energy Simulation Results

In this paper, energy consumption to treat the required solution cooling and heating loads in a conventional LD system using a chiller and a gas boiler and in an HPLD system using a heat pump and auxiliary devices were compared. The proposed HPLD system showed significant energy savings over the conventional LD system; with a 34 % reduction in electric energy, a 68 % reduction in gas energy, and a 68 % reduction in the total primary energy. In this simulation, since the heat pump was operated using an on/off control, the heat pump generated capacity were not able to match the required loads causing the needs for the use of auxiliary devices. In terms of using auxiliary devices, further researches on capacity matching between the heat pump generated loads and the required loads would be needed. Once capacity matching is achieved, the proposed HPLD system will be beneficial not only in the aspect of energy saving but also in terms of system compactness.

Acknowledgements

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References

[1] Kishorkumar L Makwana and Ketankumar N Gohel. Review: Potential applications of liquid desiccant technology in dehumidification and cooling process air. Engineering Science and Futurisic Techonology. Vol. 1, no.1, 2015
[2] Seiichi Yamaguchi, Jongsoo Jeong, Kiyoshi Saito, Hikoo Miyauchi, Masatoshi Harada.Hybrid liquid desiccant air-conditioning system: Experiments and simulations Appl. Therm. Eng., vol.31, 2011, p.3741-3747
[3] Stefano Bergero, Anna Chiari. Performance analysis of a liquid desiccant and membrane contactor hybrid air-conditioning system. Energy Buil., vol. 42, no. 11, 2010, p.1976-1986
[4] Tao Zhang, Xiaohua Liu, Yi Jiang. Performance optimization of heat pump driven liquid desiccant dehumidification system. Energy Build., Vol. 52, 2012, p.132-144
[5] Xiaofeng Niu, Fu Xiao, Zhenjun Ma. Investigation on capacity matching in liquid desiccant and heat pump hybrid air-conditioning systems. Int. J. Refrig. Vol. 35, no. 1, 2012, p.160-170
[6] Ahmed A.M., Abdalla K.N., Experimental study of heat and mass transfer of Solar powered liquid desiccant regeneration system. *Annual conference of postgraduate studies and scientific research*, Khartoum: Friendship Hall; 2012, p.299-304
[7] Jaing S., Traipathi S., Das R.S., Experimental performance of a liquid desiccant dehumidification system under tropical climates. Energy Convers. Manag., vol. 52, no. 6, 2011, p.2461-2466
[8] Jaing S., Traipathi S., Das R.S., Experimental performance of a liquid desiccant dehumidification system under tropical climates. Energy Convers. Manag., vol. 52, no. 6, 2011, p.2461-2466
[9] Lowenstein A., Slayzak S., Ryan J., Peasaran A., Advanced commercial liquid desiccant technology development sutdy. Natl. Renew. Energy Lab., bol. NREL/TP-55, 1998
[10] Alasdiar C. M., Paul T., Assessing the performance of a reservoir-based water source heat pump, Department of Mechanical and Aerospace Engineering, 2013
[11] Natural Resource Cananda’s Office of Energy Efficiency, Heating and cooling with a heat pump. 2004
[12] ASHRAE. 2011. International weather for energy calculations, Version 2.0 User’s manual. Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.
[13] Kim M.H., Park J.S., Jeong J.W., Energy saving potential of liquid desiccant in evaporative-cooling-assisted 100% outdoor air system. Energy, vol. 59, 2013, p.726-736
[14] Park J.Y., Yoon D.S., and Jeong J.W. Empirical model for predicting the dehumidification effectiveness of a liquid desiccant system. Energy and Buildings, vol. 126, 2016, p.447-454
[15] Fumo N., Goaswami D.Y. Study of an aqueous lithium chloride desiccant system: Air dehumidification and desiccant regeneration. Sol. Energy, vol. 72, no.4 2002, p.351-361
[16] US Deparment of Energy. EnergyPlus Engineering Reference: The Reference to EnergyPlus Calculation., US Dep. Energy, 2010
[17] Park D.J., Yu K.H., Yoon Y.S., Kim K.H., Kim s.s., Analysis of a building energy efficiency certification system in Korea., Sustainability, vol. 7, no.12, 2015, p.16086-16107