Performance analysis of a novel liquid desiccant-vapor compression hybrid air-conditioning system

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ABSTRACT

Liquid desiccant-vapor compression hybrid (LDCH) air-conditioning systems are popular for reducing energy consumption. This work tests a conventional LDCH air-conditioning experimental setup and establishes the corresponding mathematical model to analyze the effect of the concentrated solution branch in the solution-solution heat exchanger (SSHE) on the cooling capacity of the evaporator; the results show that the percentage of cooling capacity loss of the evaporator exceeds 10% with the small concentration difference of 1.5% in the conventional LDCH air-conditioning system. Afterwards, a new LDCH air-conditioning system is proposed by adopting an auxiliary regenerator to cut down the cooling capacity loss of the evaporator, and the analysis results show that there is a big temperature drop of the concentrated solution branch after being pretreated by the auxiliary regenerator; under the condition of concentration difference of 2.65%, the inlet temperature of concentrated solution branch from the regeneration side in the SSHE can decrease over 6 °C; and the extra heat load entering the dehumidification side from the regeneration side obviously decreases. Consequently, the evaporator only needs to spend 1.5% of its cooling capacity on the compensation for the extra heat load.

Keywords: liquid desiccant; auxiliary regenerator; solution branch; concentration difference.

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1. Introduction

Reducing energy consumption of air-conditioning systems is very important for improvement of building energy efficiency. In a conventional air-conditioning system, air is conditioned by passing it over a cooling coil to achieve both cooling and dehumidification; dehumidification is realized by cooling the air to below its dew point. As a result, the dehumidified air is cooler than the required indoor comfort level, which in turn causes reheating and further energy loss. Meanwhile, since chilled water with a relatively lower temperature is needed, the performance of chillers is decreased. Furthermore, health problems can occur because the condensed water makes the coil surface a breeding ground for bacteria. A Liquid desiccant air-conditioning system has been gaining attentions as an alternative to the conventional dehumidification system. The liquid desiccant system can dehumidify the humid air by the direct contact between the humid air and concentrated solution. The diluted solution after the dehumidification has to be concentrated to use it again. Therefore, the system mainly consists of a dehumidifier and a regenerator. Moreover, the liquid desiccant air-conditioning system can be driven by low-temperature heat source. From this feature, the efficiency of the system is expected to be enhanced by use of the waste heat from a compression heat pump for the regeneration of the diluted solution. In this study, we focus on the liquid desiccant-vapor compression hybrid (LDCH) air-conditioning system consisting of a liquid desiccant system and a vapor compression heat pump [1-3].

Some previous studies on the LDCH air-conditioning system are described as follows. 1) Prior works on developing new systems of LDCH air-conditioning units: Yadav (1995) investigated the effect of operating parameters on the performance of the hybrid solar air conditioning system. The system consists of liquid desiccant cycles and conventional vapor compression. Lithium bromide is used as solution in the liquid desiccant cycle and R-11 is used as refrigerant in the vapor compression system. The results show that the system is more promising under high latent heat load [4]. Ani et al. (2005) established a hybrid

system consisting of vapor compression system, liquid desiccant system and solar collector, and investigated the optimization of the absorber packing height by experiments [5]. Gasparella et al. (2005) simulated a hybrid system of ground-source heat pump and liquid desiccant. This novel system can save around 30% of the primary energy annually compared with the conventional systems [6]. Zhang et al. (2010 and 2012) proposed a new frost-free hybrid air-conditioning system, which integrated with liquid desiccant dehumidification units. In the system, a heat pump is used for the solution regeneration and pre-cooling in summer. In winter, frosting can be avoided by dehumidifying the air before it enters into the evaporator of the heat pump. The results show that the coefficient of performance (COP) is 30–40% higher than that of the heat pump integrated with an electric heater [7-8]. Mohan et al. (2015) investigated a new liquid desiccant-vapor compression hybrid air conditioner. In this hybrid system, the desiccant dehumidifier is placed after the evaporator, and the regenerator is after the condenser of the AC unit [9].

2) Prior works on seeking methods to improve the designs of configuration: Wu et al. (2006) investigated an internally cooled liquid desiccant dehumidifier. The evaporator of the vapor compression system is set in the dehumidifier to remove the condensing heats of vapor and cool process air, and the condenser equipped in the regenerator is used to heat the regenerating solution [10]. Mohan et al. (2008) investigated the characteristics of absorber and regenerator columns for hybrid liquid desiccant air-conditioning system. The simulation model of countercurrent absorber and regenerator is developed on the basis of heat and mass balance and heat and mass analogy [11].

3) Prior works about examination, test and simulation of system performance: Bergero and Chiari (2010 and 2011) examined the performance of an air dehumidification system integrated with a vapor compression inverse cycle. Air is dehumidified in an air-solution membrane contractor and cooled by a vapor compression cycle; lithium chloride solution is cooled by a vapor compression unit and regenerated in another membrane. The results show 60% energy savings at high latent heat load compared with a

traditional direct expansion air conditioning system [12-13]. Yamaguchi et al. (2011) conducted the performance evaluation test of a hybrid liquid desiccant air-conditioning system consisting of a conventional liquid desiccant system and a vapor compression heat pump. The main feature of this system is that the absorber and regenerator are integrated with the evaporator and condenser, respectively. The results showed that the COP of the system was 2.71 [14]. Zhao et al. (2011) investigated the real operating performance of the temperature and humidity independent control air-conditioning system. The system is composed of a two stage total heat recovery device and a two-stage air handling device coupled with refrigeration cycles. The total heat recovery device is used to recover the energy from return air to decrease the energy consumption in the fresh air handling process. In the heat pump driven air handling device, the diluted solution from the dehumidification module is heated by the exhaust heat from the condenser, and the hot concentrated solution from the generation module is cooled by the evaporator. [15]. Lazzarin and Castellotti (2007) conducted energy savings of a novel desiccant air-handling unit with that of a traditional mechanical dehumidifier. In the air-handling unit, the diluted desiccant solution is regenerated using the thermal energy supply from a primary heat pump condenser, while air regeneration is used a secondary heat pump condenser [16]. Li et al. (2005) carried out numerical simulation to predict the performance a new type of fresh air processor system in different meteorological conditions. The system is combined of liquid desiccant total heat recovery device and a refrigeration cycle, and divided into two parts, the first part is appeared in the middle solution for the purpose of total heat recovery and the second part appeared in the left and right side where liquid desiccant exchanges sensible heat with the refrigeration cycle [17].

4) Prior works on removing the superfluous heat capacity of the condenser: the amount of heat from the condenser in a LDCH air-conditioning system is often more than the heat needed in desiccant regeneration process. To remove the superfluous condensing heat, Zhang et al. (2012) used two different methods for removing the extra heat. In the first method, they used a water-cooled assistant condenser; in

the second method, they added an air-cooled assistant condenser. The coefficient performance of the system with the water-cooled condenser was approximately 35% higher than the basic system [18]. Niu et al. (2012) studied the capacity matching of a new hybrid liquid desiccant air conditioning system, which consisted of a liquid desiccant dehumidification and a heat pump system with double condensers. The results show that the system with a double-condenser is a feasible configuration for achieving capacity matching [19]. Chen et al. (2014) proposed a novel temperature and humidity independent control air-conditioning system based on liquid desiccant. In the system, an air-cooled condenser works in series with a solution-cooled condenser, and the refrigerant from compressor rejects its condensation heat firstly to the solution-cooled condenser and then to the air-cooled condenser. This makes capacity matching between the dehumidification and regeneration processes more efficient [20].

However, these reviewed studies above had paid few attentions to the heat exchanging process between the concentrated solution from regeneration side and the diluted solution from the dehumidification side. In a LDCH air-conditioning system, the solution-solution heat exchanger (SSHE) lies between the concentrated solution branch from regeneration side and the diluted solution branch from dehumidification side, and the concentrated solution branch from the regeneration side is pre-cooled in the SSHE before entering the dehumidification side, but the temperature of concentrated solution branch leaving from the SSHE is still higher than the temperature of solution in dehumidification side. Therefore, the concentrated solution branch delivered from the SSHE will increase the additional consumption of cooling capacity of the evaporator, and this process becomes energy inefficient. Consequently, the effect of concentrated solution branch on the heating exchanging process of the SSHE deserves a further investigation.

Obviously, the temperature and the flow rate of concentrated solution branch are the key factors which heavily influence the additional consumption of cooling capacity of the evaporator. One of the novelties of this work is that the effect of concentrated solution branch fluid on the heat exchanging process of the

SSHE is analyzed based on a conventional LDCH air-conditioning experiment setup. A further contribution is that a new LDCH air-conditioning system is proposed to reduce the additional consumption of evaporating load of heat pump. The results can help to design energy-efficient LDCH air-conditioning systems.

2. Description of a conventional LDCH air-conditioning system

2.1 Working principle

The schematic of a conventional LDCH air-conditioning system is shown in Fig.1, which consists of regeneration module, dehumidification module, SSHE, and a vapor compression heat pump. The desiccant solution is concentrated in the regenerator after being heated by the condenser of heat pump; the desiccant solution imbibes water from the processed air in the dehumidifier after being cooled in the evaporator. In other words, the condenser of heat pump is used to supply heat load for inspissating desiccant solution, and the evaporator serves to provide cooling capacity for dehumidifying process.

It should be noticed that the liquid desiccant circulates in the LDCH system. As shown in Fig.1, in the dehumidification side, the desiccant solution (point 4) from the diluted solution pump flows successively through the evaporator (point 8), the rotameter, dehumidifier, diluted solution tank and the rotameter (point 2), afterwards mixes with the concentrated solution branch (point 7) which comes from the SSHE, finally backs to the diluted solution pump (point 4); the solution branch (point 2) from the diluted solution tank passes through the SSHE (point 6). Similarly in the regeneration side, the desiccant solution (point 3) from the concentrated solution pump flows ordinally through the condenser (point 9), the rotameter, regenerator, concentrated solution tank (point 1) and the rotameter, afterwards combines with the diluted solution branch (point 6) from the SSHE, finally returns to the concentrated solution pump (point 3); the solution branch (point 5) from the concentrated solution tank passes through the SSHE (point 2).

Fig.2 presents the T - x diagram of desiccant solution, the process 4-8-2 stands for the solution process of

being cooled in the evaporator and diluted in the dehumidifier, the process 3-9-1 is the solution process of being heated in the condenser and dehydrated in the regenerator; and the process 2-6 and process 5-7 indicate the heat transfer between diluted solution and concentrated solution in the SSHE. This work defines the temperature difference between point 7 and point 2 as temperature difference of Δt ; as shown in Fig.2, we can see that the temperature of solution point 7 is higher than those of point 2, so the enthalpy of solution point 7 is commonly larger than that of solution point 2; that is to say, accompanied by the concentrated solution branch (point 7), some additional heat load is carried into the dehumidification side from regeneration side. Since the temperature of spraying solution above the dehumidifier needs to remain invariable and maintain at a lower level, so the evaporator has to spend a considerable share of its cooling capacity on countervailing the extra heat load, so the cooling capacity of evaporator using for dehumidification process decays and gets an unsatisfactory loss.

2.2 Mathematical model

The dehydration rate from liquid desiccant to outdoor air in the regenerator is commonly more than the moisture absorption rate from the processed air to liquid desiccant in the dehumidifier, to prevent the balance disorder of solution concentration in the LDCH air-conditioning system, a certain amount of water will be supplemented into regeneration side. By solving the mass and energy balances of different components simultaneously, key equations of the LDCH air-conditioning system are presented in detail in Appendix A.

It should also be noted that the concentrations in the concentrated solution tank and diluted solution tank are defined as x_1 and x_2 respectively; the average value of x_1 and x_2 is a fixed value of 26%, and the solution temperature of diluted solution tank is assumed to remain 25°C.

2.3 Experimental setup

The experimental facility is shown in Fig.3, the rated input power of the scroll compressor is 8.0 kW,

the refrigerant of the vapor compression system is R407C and the liquid desiccant is aqueous solution of lithium chloride. The main body of the dehumidifier and regenerator is the structure of a packed-bed tower. The packed-bed consists of specially impregnated and corrugated cellulose paper sheets with 30° flute angles. The packed-bed tower is cubical with 90 cm in width, 78 cm in height and 30 cm in thickness; and its specific surface area is $450 \text{ m}^2/\text{m}^3$. There are two solution storage tanks and two pumps in this system. One tank is used for storing diluted liquid desiccant solution while the other is for the concentrated solution. The concentration of liquid desiccant is regulated by adding water or lithium chloride particles to the solution. During the experiment process, the core devices of the LDCH air-conditioning system, such as the dehumidifier, the regenerator and solution tanks, are thermally insulated.

The temperatures of the air and the desiccant solution are measured by K-type thermocouples; also the humidity and temperature sensors are used to measure the parameters of the inlet and outlet air. The density of desiccant solution is measured by the specific gravity hydrometer and the air flow rate by the spraying nozzle. The mass concentration of the aqueous lithium chloride is indirectly tested by measuring the density and temperature of the solution, and the properties of the aqueous lithium chloride can be found in Ref. [21]. According to the density and temperature of the desiccant solution, its weight concentration can be calculated. All the experiments are finished in the environmental laboratory, and the main measurement instruments are shown in Table 1.

3. Experimental results and analysis

Fig.4 shows the variation of the branch fluid percentage of liquid desiccant with the solution concentration difference. It can be claimed that the branch fluid percentage declines sharply with the accretion of solution concentration difference; also the trend of experiment results is in good agreement with that of the model.

Fig.5 illustrates the variation of cooling capacity loss of the evaporator with the concentration

difference of desiccant solution. It can be observed that with the decrease of concentration difference, the cooling capacity loss of the evaporator increases quickly. Particularly, as depicted in Fig. 6, when the concentration difference of desiccant solution drops to 0.5%, the percentage of cooling capacity loss reaches unbelievably as high as 37%. The dramatic rise of cooling capacity loss is due to the increment of the solution branch flow rate while decreasing the concentration difference; and the higher the branch flow rate of concentrated solution, the more the extra heat load being carrying to dehumidification side. Also, from the Fig.5 and Fig.6, we can conclude that a bigger concentration difference of solution should be recommended in the conventional LDCH air-conditioning system.

As shown in Fig.5 and Fig.6, it also can be pointed out that the main trend of experiment results is consistent with the modeling value, the deviation value between experiment value and modeling value is due to the real effect of the heat insulation in the experimental setup, and the energy leakage is unavoidable during the experimental tests. Compared with the smaller concentration difference, the influence of energy leakage on the bigger concentration difference is more obvious. This can be explained that the bigger concentration difference of desiccant solution, the lower the absolute value of cooling capacity loss; but the absolute value of energy leakage of cooling capacity hardly fluctuates at different concentration differences. Consequently, energy leakage shows a heavier influence on the cooling capacity loss at a bigger concentration difference of desiccant solution, and the deviation value between experiment value and modeling value increases with the accretion of concentration difference.

As presented in Fig.7, we can conclude that percentage of cooling capacity loss with bigger temperature difference Δt is significantly higher than that with smaller temperature difference. Even the temperature difference is as low as 8 °C, the percentage of cooling capacity loss still stands up to 10% with concentration difference of 1.5%, and this significantly restricts the improvement of system performance of conventional LDCH air-conditioning system. Detail experimental results are provided in Table.2, the

average of cooling capacity loss percentage of the evaporator exceeds 10%. Measures should be taken to change this unfavorable situation, and this work proposes a new LDCH air-conditioning system by adopting an auxiliary regenerator in the following section.

4. Description of a new LDCH air-conditioning system

A new conceiving LDCH air-conditioning system is shown in Fig.8; an auxiliary regenerator in the regeneration side is introduced to reduce the cooling capacity loss of the evaporator. The desiccant solution point 3 leaving from the concentrated solution pump is separated into the condenser and the auxiliary regenerator; the solution branch (point 3) sent into the auxiliary regenerator is dehydrated and cooled down to state point 5, then flows through the regulative valve, and is delivered into the SSHE to be further cooled down to a lower temperature (point 7). Compared with a conventional LDCH air-conditioning system, this method can reduce the temperature difference between the solution point 7 from the regeneration side and the solution (point 2) from the dehumidification side; and the lower temperature difference can abate the cooling capacity loss of the evaporator.

Fig.9 presents the T - x diagram of desiccant solution in this new LDCH air-conditioning system. The process 4-8-2 stands for the solution process of being cooled in the evaporator and diluted in the dehumidifier, and the process 3-9-1 is the solution process of being heated in the condenser and inspissated in the regenerator. The process 2-6 indicates that the diluted solution is pre-heated in the SSHE, the process 3-5 means that the concentrated solution branch is pre-cooled and dehydrated in the auxiliary regenerator, and process 5-7 indicates that the concentrated solution branch is further cooled in the SSHE. Comparing Fig.2 and Fig.9, we can find out that the temperature difference between solution point 2 and point 7 declines obviously, this is own to the temperature difference of $\Delta t_{3,5}$ caused by the auxiliary regenerator.

Also, by solving the mass and energy balances of different components simultaneously, key equations of the new LDCH air-conditioning system are presented in detail in Appendix B, and the modeling of new

LDCH air-conditioning system is based on the assumption that the operation condition and parameters of regeneration side, such as the outdoor air flow rate, the circulating flow rate of solution pump and the temperature of solution tank, are exactly the same with that for the conventional LDCH air-conditioning system.

It may also be noted that, while m_{e1} is equal to zero in the equation (15), the new LDCH air-conditioning system can be simplified, and the solution concentration of point 5 equals to that of point 3. But for the conventional LDCH air-conditioning system, the liquid desiccant concentration of point 5 is equal to that of point 1(Fig.1), this simplified LDCH system is not the same with the conventional LDCH system.

5. Discussions

5.1 Variation of temperature difference

Comparison of the experiment value of conventional LDCH air-conditional system and modeling value of the new LDCH air-conditioning system is presented in Table.3. As shown in Table.3, there is a big temperature drop of concentrated solution branch after being pretreated by the auxiliary regenerator. Under the condition of concentration difference of 2.65%, the temperature drop $\Delta t_{3,5}$ is about 7.0 °C while both β_e and β_d are 1.0, and $\Delta t_{3,5}$ is still as high as 6.3 °C even with a deduction of viewing both β_e and β_d as 0.9; in other words, the inlet temperature of concentrated solution branch from the regeneration side in the SSHE can decrease over 6 °C. Expected in an ideal circumstance, the temperature difference between solution point 7 and point 2 will be less than 1.8 °C. Compared Table.2 and Table.3, under the condition of concentration difference of 2.65%, the temperature difference between solution point 7 and point 2 declines from 7.8 °C to 1.5 °C; from Fig.7, we can conclude that the loss percentage θ_c of cooling capacity will decrease from 8.2% to about 1.5%.

From Table.3, we can see that with the decrease of the concentration difference Δx , the temperature

difference of $\Delta t_{3,5}$ slightly reduces; even the solution concentration difference of Δx falls to 1.26%, the temperature difference of $\Delta t_{3,5}$ can still be as high as 6.0 °C with a fixed value of 0.9 to both β_e and β_d . Consequently, there is a significantly decline in temperature difference between point 7 and point 2, and the extra heat load entering the dehumidification side from the regeneration side obviously decreases, and the cooling capacity of evaporator gets a less loss to compensate the extra heat load.

Fig.10 depicts the variation of solution temperature difference with solution concentration difference. From fig.10, we can see that the solution temperature difference of the new LDCH system significantly decreases to about 2 °C compared with the conventional LDCH air-conditioning system; and that the solution temperature difference of the new LDCH system slightly reduces with the increase of the solution concentration difference, this indicates that a bigger concentration difference of liquid desiccant is more beneficial to lighten the cooling capacity loss of the evaporator.

5.2 Variation of solution concentrations

Figs. (11-13) depict the variation of solution concentration. From Fig.10 and Fig.11, It can be seen that, comparing with the conventional LDCH air-conditioning system, the concentrations of both solution point 1 and point 5 obviously decline in the new LDCH air-conditioning system. On the contrary, As shown in Fig.13, we can find out that solution concentrations of both point 1 and point 5 in the new LDCH air-conditioning system are slightly higher than those in the conventional LDCH air-conditioning system. Figs. (11-13) indicate that the concentrated solution branch (point 5) can obtain higher solution concentration under the condition of bigger solution concentration difference, and this can benefit the dehumidifying process for the humid air, because a higher solution concentration has a lower surface water vapor pressure, and the lower vapor pressure promotes the mass transfer process and gives better dehumidification effect. All this suggests that the liquid desiccant with bigger concentration difference of 2.65% of is more suitable for the new LDCH air-conditioning system.

6. Conclusions

The present work tests a conventional LDCH air-conditioning system and establishes the corresponding mathematical model to analyze the effect of the concentrated solution branch in the SSHE on the cooling capacity of the evaporator; also a new LDCH air-conditioning system is proposed by adopting an auxiliary regenerator to cut down the cooling capacity loss of the evaporator. The conclusions are as follows:

(1) As far as the conventional LDCH air-conditioning system is concerned, even the temperature difference between the outlet of concentrated solution branch and the inlet of diluted solution branch in the SSHE is as low as 8 °C, the percentage of cooling capacity loss of the evaporator still exceeds 10% with the concentration difference of 1.5%; and this significantly restricts the enhancement of system performance of the conventional LDCH air-conditioning system.

(2) For the new LDCH air-conditioning system, there is a big drop of temperature of concentrated solution branch after being pretreated by the auxiliary regenerator. Under the condition of concentration difference of 2.65%, the inlet temperature of concentrated solution branch from the regeneration side in the SSHE can decrease over 6 °C, the extra heat load entering the dehumidification side from the regeneration side obviously decreases, and the loss percentage θ_c of cooling capacity lowers from 8.2% to about 1.5%.

(3) A bigger concentration difference of liquid desiccant can be more beneficial to lighten the cooling capacity loss of the evaporator; also can help the concentrated solution branch in the SSHE obtain higher solution concentration, this can gives better dehumidification effect for the humid air. The liquid desiccant with bigger concentration difference of 2.65% is optimal for the new LDCH air-conditioning system.

In a word, adding an auxiliary regenerator is a valid method to cut down the pernicious influence of the concentrated solution branch on the cooling capacity of the evaporator. The results and conclusions can be useful to design an energy-efficient LDCH air-conditioning system.

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Appendix A. Details of key equations of heat and mass transfer

To analyze the effect of temperature difference of the SSHE on the cooling capacity of evaporator, the mass and energy balances of hybrid system should be established. The main equations for heat and mass transfer process are listed as follows:

1) Mass balance of dehumidification side

During the steady operation of the conventional LDCH air-conditioning system, by assuming that the circulating solution flow rate of the diluted solution equals to that of the concentrated solution pump, the concentration of solution in dehumidification side remains constant, therefore,

$$(m_c \alpha - m_d) x_7 = m_c \alpha x_2 \quad (1)$$

$$(m_c \alpha - m_d) x_7 + (m_c - m_c \alpha + m_d) x_2 = m_c x_4 \quad (2)$$

Where m_c stands for the circulating solution flow rate of the diluted and concentrated solution pump, and α means the percentage of solution branch fluid in the dehumidification side to the circulating solution flow rate m_c ; x_2 , x_4 and x_7 refer to the solution concentrations of point 2, 4 and 7, respectively. And m_d is the water absorption rate of the solution from the processed air in the dehumidifier.

2) Mass balance of regeneration side

$$m_c \alpha x_6 + m_c (1 - \alpha) x_1 = m_c x_3 \quad (3)$$

$$x_1 = x_5 = x_7 \quad (4)$$

$$x_2 = x_6 \quad (5)$$

Where m_{e1} means the released moisture rate from the liquid desiccant to outdoor air in the auxiliary regenerator, and m_e represents the total released moisture rate from the liquid desiccant to outdoor air in the regeneration side. x_1 , x_3 , x_5 and x_6 are the different solution of point 1, 3, 5 and 6, respectively.

Combining equations (1) and (4), we have

$$x_1 = \frac{m_c \alpha}{(m_c \alpha - m_d)} x_2 \quad (6)$$

Substituting equation (6) in equation (3) yields

$$x_3 = \frac{(m_c - m_d) \alpha}{m_c \alpha - m_d} x_2 \quad (7)$$

Using equations (2), (4) and (6), we can obtain

$$x_4 = (1 + \frac{m_d}{m_c}) x_2 \quad (8)$$

Defining the concentration difference of solution Δx as

$$\Delta x = x_1 - x_2 \quad (9)$$

While Δx and x_2 as known parameters, according to the above equations (8-12), all the concentrations of other liquid desiccant points can be calculated.

3) Supplemental filling water rate in regeneration side

The supplemental filling water can be defined as,

$$m_s = m_e - m_d \quad (10)$$

Where m_e represents the total dehydration rate from the desiccant solution to the outdoor air in the regeneration side, and m_s is the supplemented filling water in the regeneration side.

4) Cooling capacity loss of the LDCH air-conditioning system

The loss of cooling capacity can be calculated as follows:

$$q_{co} = (m_c \alpha - m_d) h_7 - m_c \alpha h_2 \quad (11)$$

$$q_{eva} = m_c (h_4 - h_8) \quad (12)$$

$$\theta_c = q_{co} / q_{eva} \quad (13)$$

Where q_{co} means the loss of cooling capacity of the evaporator and q_{eva} denotes the cooling capacity of the evaporator. θ_c is the loss percentage of cooling capacity of the evaporator.

Appendix B. Key equations of new LDCH air-conditioning system

1) Mass and energy transfer of the auxiliary regenerator

$$q_{e1} = \beta_e (\alpha q_e) \quad (14)$$

$$m_{e1} = \beta_d (\alpha m_e) \quad (15)$$

Where q_{e1} stands for the removal heat load from liquid desiccant to outdoor air in the auxiliary regenerator, m_{e1} represents the dehydration rate from the desiccant solution to the outdoor air in the auxiliary regeneration, and q_e stands for the total removal heat load in the regeneration side, m_e represents the total dehydration rate in the regeneration side; β_e and β_d are the effective coefficients of heat transfer and mass transfer, respectively; in an ideal heat and mass transfer process, both β_e and β_d equal to 1.0, in a real heat and mass transfer process, this work defines both β_e and β_d as 0.9.

2) Mass balance of dehumidification side

While remaining the solution concentration of dehumidification side invariable, the key equations are the same as those of conventional LDCH hybrid system.

3) Mass balance of regeneration side

$$(m_c \alpha - m_d) x_5 = (m_c \alpha - m_d + m_{e1}) x_3 \quad (16)$$

$$x_5 = x_7 \quad (17)$$

Combining equations (1) and (17), we have

$$x_5 = \frac{m_c \alpha}{(m_c \alpha - m_d)} x_2 \quad (18)$$

Substituting equation (18) in equation (16) gives

$$x_3 = \frac{m_c \alpha}{m_c \alpha - m_d + m_{e1}} x_2 \quad (19)$$

Using equations (2) and (18), we can obtain

$$x_4 = \left(1 + \frac{m_d}{m_c}\right) x_2 \quad (20)$$

Inserting the equations (5) and (19) into equation (3) yields

$$x_1 = \frac{\alpha}{1 - \alpha} \left[\frac{m_c}{m_c \alpha - m_d + m_{e1}} - 1 \right] x_2 \quad (21)$$

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Fig.1. Schematic of a conventional LDHP air-conditioning system

Fig.2. T-x diagram of solution in the conventional LDCH air-conditioning system

Fig.3. Photo of the LDCH air-conditioning experiment setup

Fig.4. Branch fluid percentage of solution with solution concentration difference

Fig.5. Cooling load loss with concentration difference of solution

Fig.6. Loss percentage of cooling load with solution concentration difference

Fig.7. Loss percentage of cooling load of the evaporator with Δx and Δt

Fig.8. Schematic diagram of a new LDCH air-conditioning system

Fig.9. T - x diagram of solution in the new LDCH air-conditioning system

Fig.10. Variation of the solution temperature difference with concentration difference

Fig.11. Comparison of solution concentration ($\Delta x=1.26\%$)

Fig.12. Comparison of solution concentration ($\Delta x=1.67\%$)

Fig.13. Comparison of solution concentration ($\Delta x=2.65\%$)

Table 1

Specification of the different measuring devices

Parameters	Devices	Accuracy	Operational range
Air dry-bulb temperature	K-type thermocouple	$\pm 0.1\text{ }^{\circ}\text{C}$	0-100 $^{\circ}\text{C}$
Humidity and temperature sensor	Rotronic HF534	$\pm 0.8\%$ RH/ $\pm 0.1\text{ }^{\circ}\text{C}$	0-99%/ -40-85 $^{\circ}\text{C}$
Air flow rate	Spray nozzle	$\pm 3\%$	500-4500 m^3/h
Desiccant solution temperature	K-type thermocouple	$\pm 0.1\text{ }^{\circ}\text{C}$	0-100 $^{\circ}\text{C}$
Desiccant solution flow rate	ABGLW-25	$\pm 1\%$ R	0.5-10 m^3/h
Desiccant solution density	Specific gravity hydrometer	$\pm 1\text{ kg/m}^3$	1100-1900 kg/m^3
Input power of the compressor	Digital power meter (WT230)	$\pm (0.1\% \text{ R} + 0.1\% \text{ FS})$	15-600 V/ 0.5-50 A

Table 2

Specification of the experimental results of LDCH system

Parameters	$\Delta x = 2.65\%$	$\Delta x = 1.67\%$	$\Delta x = 1.26\%$
Input power of the compressor, kW	8.47	8.32	8.29
Cooling capacity of the evaporator, q_{eva} , kW	14.81	14.59	14.36
Heating load of the condenser, q_{cond} , kW	22.90	22.73	22.39
Outdoor air flow rate in the regenerator, m^3/h	1701.2	1705.7	1712.5
Inlet dry/wet-bulb temperature of outdoor air in regenerator, $^{\circ}\text{C}$	34.9/ 28.0	35.0/28.0	34.9/28.1
Outlet dry/wet-bulb temperature of outdoor air in regenerator, $^{\circ}\text{C}$	45.5/ 35.2	45.2/35.1	45.2/34.9
Removal moisture by outdoor air in the regeneration, m_e , kg/h	21.43	21.20	20.26
Removal heat load by outdoor air in the regenerator, q_e , kW	21.30	20.96	20.33
Circulating flow rate of solution pump, m_c , kg/h	3506.4	3520.8	3517.2
Solution temperature in the concentrated solution tank, $^{\circ}\text{C}$	50.6	50.9	51.3
Processed air flow rate in the dehumidifier, m^3/h	2852.5	2909.7	2912.5
Inlet dry/wet-bulb temperature of processed air in dehumidifier, $^{\circ}\text{C}$	34.9/ 28.0	35.0/ 28.0	34.9/ 28.0
Outlet dry/wet-bulb temperature of processed air in dehumidifier, $^{\circ}\text{C}$	33.3/ 24.6	33.5/ 24.8	33.6/ 25.1
Solution temperature in the diluted solution tank, $^{\circ}\text{C}$	24.6	24.9	25.3
Temperature difference of point 7 and 2, $\Delta t = t_7 - t_2$, $^{\circ}\text{C}$	7.8	8.1	8.3
Absorbed moisture from processed air in dehumidifier, m_d , kg/h	16.68	16.00	15.10
Removal cooling capacity by processed air in the dehumidifier, kW	13.59	12.97	12.12
Cooling capacity loss, q_{co} , kW	1.22	1.62	2.24
Cooling capacity loss percentage, θ_c	8.2%	11.1%	15.6%
Supplemented filling water in the regeneration side, m_s , kg/h	4.76	5.20	5.16
Percentage of branch flow to circulating flow rate, α	0.047	0.089	0.112
Concentration of Concentrated solution tank, x_1 , %	27.36	26.85	26.63
Concentration of diluted solution tank, x_2 , %	24.73	25.18	25.37
Concentration of solution point 3, x_3 , %	27.38	26.41	26.27
Concentration of solution point 4, x_4 , %	24.85	25.29	25.48

Table 3

Comparison of the experiment and modelling

Parameters	Experiment of conventional LDHP system			Modeling of the new LDHP system					
	$\Delta x = 1.26\%$	$\Delta x = 1.67\%$	$\Delta x = 2.65\%$	$\Delta x = 1.26\%$		$\Delta x = 1.67\%$		$\Delta x = 2.65\%$	
				$\beta_d = \beta_e = 1.0$	$\beta_d = \beta_e = 0.9$	$\beta_d = \beta_e = 1.0$	$\beta_d = \beta_e = 0.9$	$\beta_d = \beta_e = 1.0$	$\beta_d = \beta_e = 0.9$
Air flow rate in the regenerator, m ³ /h	1712.5	1705.7	1701.2	1712.5	1712.5	1705.7	1705.7	1701.2	1701.2
Inlet dry/wet-bulb temperature of air in regenerator, °C	34.9/ 28.1	35.1/ 27.8	34.9/ 28.0	34.9/ 28.1	34.9/ 28.1	35.0/ 28.0	35.0/ 28.0	34.9/ 28.0	34.9/ 28.0
Outlet dry/wet-bulb temperature in regeneration side, °C	45.2/ 34.9	45.2/ 35.1	45.5/ 35.2	45.2/ 34.9	45.2/ 34.9	45.2/ 35.1	45.2/ 35.1	45.5/ 35.2	45.5/ 35.2
Circulating flow rate of solution pump, m _c , kg/h	3517.2	3520.8	3506.4	3517.2	3517.2	3520.8	3520.8	3506.4	3506.4
Percentage of branch flow to circulating flow rate, α	0.112	0.089	0.047	0.112	0.112	0.089	0.089	0.047	0.047
Temperature of concentrated solution tank, °C	51.3	50.9	50.6	51.3	51.3	50.9	50.9	50.6	50.6
Outlet dry/wet-bulb temperature in auxiliary regenerator, °C	/	/	/	36.1/ 28.9	35.9/ 28.8	35.9/ 28.7	35.8/ 28.6	35.4/ 28.4	35.4/ 28.3
Removal moisture by air in the auxiliary regeneration, kg/h	/	/	/	2.27	2.04	1.89	1.70	1.01	0.91
Removal heat load by air in the auxiliary regenerator, kW	/	/	/	2.28	2.05	1.87	1.68	1.00	0.90
Temperature drop, $\Delta t_{3,5} = t_3 - t_5$, °C	/	/	/	6.7	6.0	6.9	6.2	7.0	6.3
Temperature difference of point 7 and 2, $\Delta t = t_7 - t_2$, °C	8.3	8.1	7.8	1.6	2.3	1.2	1.9	0.8	1.5