TRNSYS Modelling of a Hybrid Membrane Liquid Desiccant Air Conditioning System

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Abstract

The attention to liquid desiccant air conditioning (LDAC) systems is increasing worldwide. This is mainly due to the ability of these systems to efficiently maintain indoor air humidity within acceptable ranges in energy efficient way compared to conventional air conditioning systems. LDAC systems still under R&D in order to reduce/eliminate common drawbacks, such as the carryover of desiccant droplets in air streams, and to develop novel configurations which aim to achieve higher energy efficiency. To address these, two improvements are implemented on a LDAC system in this study. First, liquid-to-air membrane energy exchangers (LAMEEs) are used as the dehumidifier and regenerator, in order to avoid the carryover of desiccant droplets in air streams. Second, the desiccant solution heating and cooling energy requirements are simultaneously provided using a hybrid liquid desiccant heat pump (LDHP). The proposed hybrid membrane LDAC (H-M-LDAC) system is annually simulated using TRNSYS for a small office building located in Miami, FL. Results show that the COPs of the return air heat pump (RAHP), LDHP and H-M-LDAC system vary under different operating conditions with an approximate average annual values of 4.5, 7.8 and 1, respectively.

1 Introduction

The last few decades have witnessed the development of several novel air conditioning technologies. With the increasing concerns regarding humidity control, liquid desiccant air conditioning (LDAC) technologies have become promising competitors with the widely used conventional air conditioning systems. The main components of a LDAC system are dehumidifier, regenerator, cooling coil and heating coil. Although liquid desiccant systems may still include one heat pump or more, in order to provide the heating and cooling requirements of the desiccant solution streams, the operating temperatures of desiccant solution in LDAC systems enable the used heat pumps to be operated in energy efficient manner. For instance, the humid air is dehumidified in a conventional air conditioning system by overcooling the humid air stream below its dew point temperature in order to condense the moisture required to be removed, which requires low evaporating temperatures. While, in LDAC systems, the air dehumidification is achieved using a concentrated and cool liquid desiccant solution stream, which has a lower surface vapor pressure than the humid air stream required to be dehumidified.

In LDAC systems, the desiccant solution should be regenerated, after it absorbs the moisture from humid air. Liquid desiccant regeneration is achieved by heating the desiccant solution up to 45^{o}C-65^{o}C in order to increase its surface vapor pressure, and then transferring the

moisture from the desiccant solution to the regeneration air, which could be either exhaust air from the building or ambient air. Since considerable amount of heating is consumed in the regeneration of dilute desiccant solution, many studies have investigated the potential of integrating LDAC system with solar thermal system (Abdel-Salam et al. 2014; Crofoot and Harrison 2012; Li et al. 2010; Gommed and Grossman 2007; Davies 2005). The results of solar LDAC systems were promising from energy consumption point of view. However, the achieved energy savings using solar LDAC systems are accompanied with higher initial costs (Abdel-Salam et al. 2014). Although they seemed to be promising several years ago, solar LDAC systems do not seem to be the most energy efficient configuration any more, especially considering the fact that a heat pump may be still needed in a solar LDAC system in order to cool the desiccant solution before it removes the moisture from the humid air. Some researchers have investigated the potentiality of using the heat rejected from the solution cooling heat pump in order to cover the heating requirements of dilute desiccant solution, and this was found to improve the coefficient of performance (COP) of the system. Thus, a hybrid liquid desiccant heat pump (LDHP) will be used in the current study in order to simultaneously provide the solution heating and cooling requirements.

Another issue with majority of LDAC systems is that direct contact conditioners are used as dehumidifier and regenerator. This may cause the carryover of desiccant droplets in supply and exhaust air streams, which might lead to several undesirable consequences such as the degradation of indoor air quality (IAQ) and corrosion of downstream ducting and equipment. Previous studies have showed that there are two ways to eliminate the desiccant droplets carryover problem. Internally cooled/heated low solution flow rate conditioner is one promising design (Lowenstein 1994; Lowenstein et al. 2006), where it is characterized by its low solution flow rate which may be up to 20 times lower than conventional conditioners. This leads to a rapid change in its temperature, and thus this design should be internally cooled/heated, and cannot be operated in adiabatic mode. This design was investigated in several previous studies by the Solar Calorimetry Laboratory at Queen's University (Mesquita 2007; Jones 2008; Crofoot 2012). Another design, which is able to eliminate the desiccant droplets carryover problem, is the liquid-to-air membrane energy exchanger (LAMEE). In a LAMEE, the air and solution streams are separated using semi-permeable membranes, which allow the transfer of water vapor but do not permit any liquid droplets to be transferred. Several experimental and theoretical studies have been performed on the LAMEE by the Thermal Laboratory at University of Saskatchewan (Abdel-Salam et al. 2013a; Ge et al. 2013a). The authors have performed several studies on different configurations of membrane LDAC (M-LDAC) systems, which use LAMEEs as conditioners (Abdel-Salam and Simonson 2014; Abdel-Salam et al. 2014; Abdel-Salam et al. 2013b-2013d). The performances of the studied M-LDAC systems were found to be promising, and thus LAMEEs are used in the current study. A schematic diagram which shows the specifications of the used LAMEEs is presented in Fig.1. More information about the properties of the semi-permeable membranes, which are used in the LAMEEs, is presented in a previous study by the authors (Abdel-Salam and Simonson 2014).

In the current study, a hybrid M-LDAC (H-M-LDAC) system will be annually simulated, for the first time in the scientific literature, using the TRNSYS building energy simulation software. The proposed H-M-LDAC system is characterized by two advanced modifications, compared to basic LDAC systems as follows. (1) LAMEEs are used in the H-M-LDAC systems as dehumidifier and regenerator, which eliminate the desiccant droplets carry over problem. (2) Only one liquid desiccant heat pump (LDHP) is used in the H-M-LDAC system to

simultaneously cover the heating and cooling requirements of the dilute and concentrated solution streams, respectively. The annual performance of the H-M-LDAC system is investigated in the current study for a small office building located in Miami, FL.

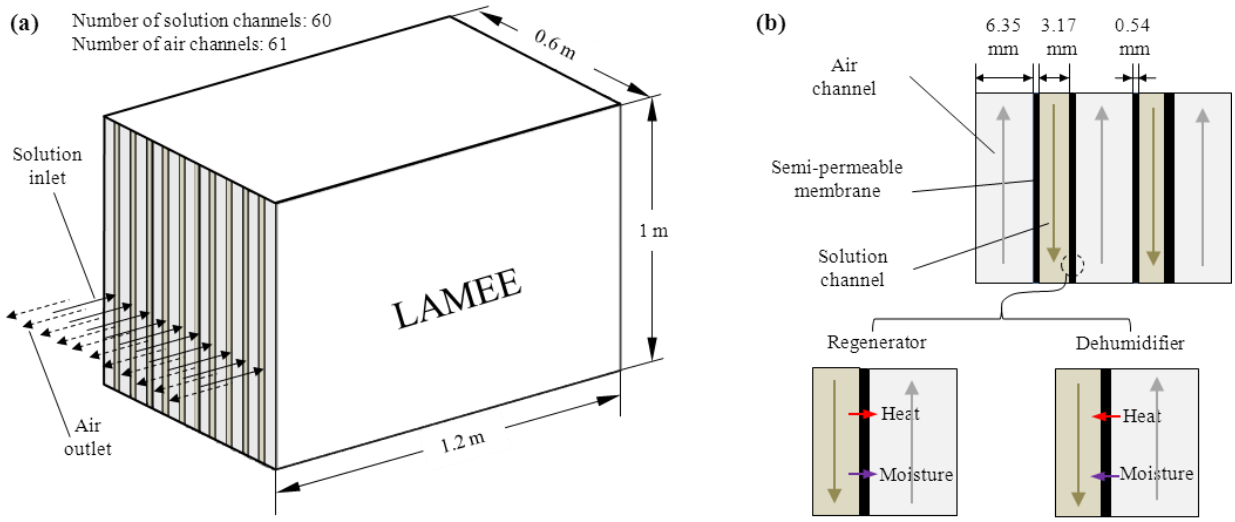

Fig. 1. A schematic diagram of the (a) LAMEE, and (b) heat and mass transfer directions in the LAMEE when it operates as a dehumidifier and regenerator. (Abdel-Salam et al. 2014)

2 System Description

A schematic diagram for the proposed H-M-LDAC system is presented in Fig. 2. The hot-humid outdoor air passes through the dehumidifier, where the entire latent load and a portion of the sensible load are covered. The return air is sensibly cooled as it passes through the evaporator of the return air heat pump (RAHP), in order to cover any additional sensible load not covered by the dehumidifier. The dehumidified fresh air is mixed with the sensibly cooled return air, and then the mix is supplied to the conditioned space.

As the hot and humid fresh air passes through the dehumidifier, it transfers heat and moisture to the desiccant solution stream, and thus the solution leaves the dehumidifier warm and dilute. The desiccant solution should be regenerated before being reused in the dehumidifier, in order to increase its ability to efficiently dehumidify the humid fresh air. Thus, it has to be heated to a high temperature (i.e. 50^{o}C in the current study), in order to increase its surface vapor pressure to a level which leads to the transfer of moisture from it to the regeneration air stream (i.e. ambient air in this study). The warm and dilute solution leaving the dehumidifier is preheated in a solution-to-solution sensible heat exchanger; then, it is heated up to 50^{o}C as it passes through the condenser of the liquid desiccant heat pump (LDHP). As the hot and dilute desiccant solution passes through the regenerator, it transfers heat and moisture to the regeneration air, and consequently it leaves the regenerator at higher concentration. The hot and concentrated desiccant solution has to be cooled before entering the dehumidifier, in order to decrease its surface vapor pressure to a level which makes it capable of absorbing moisture from the humid fresh air. Thus, it is precooled in the solution-to-solution sensible heat exchanger, and then it is cooled down to 20^{o}C in the evaporator of the LDHP. The cool and concentrated desiccant solution enters the dehumidifier in order to dehumidify and precool the hot and humid fresh air. It is worth mentioning that the desiccant solution used in the current study is Lithium Chloride (LiCl).

As can be seen from Fig. 2, the LDHP can be operated in two modes, as it is operated so that either the condenser typically matches with the solution heating requirements (*Case A*), or the evaporator load typically matches with the solution cooling requirements (*Case B*). In *Case A*, the evaporator load is higher than the solution cooling requirements, and thus an auxiliary evaporator is used to cover the evaporator load. In *Case B*, the condenser load is higher than the solution heating requirements, and thus an auxiliary condenser is used to cover the condenser load. More details about each operating condition are presented in Section 3.3.

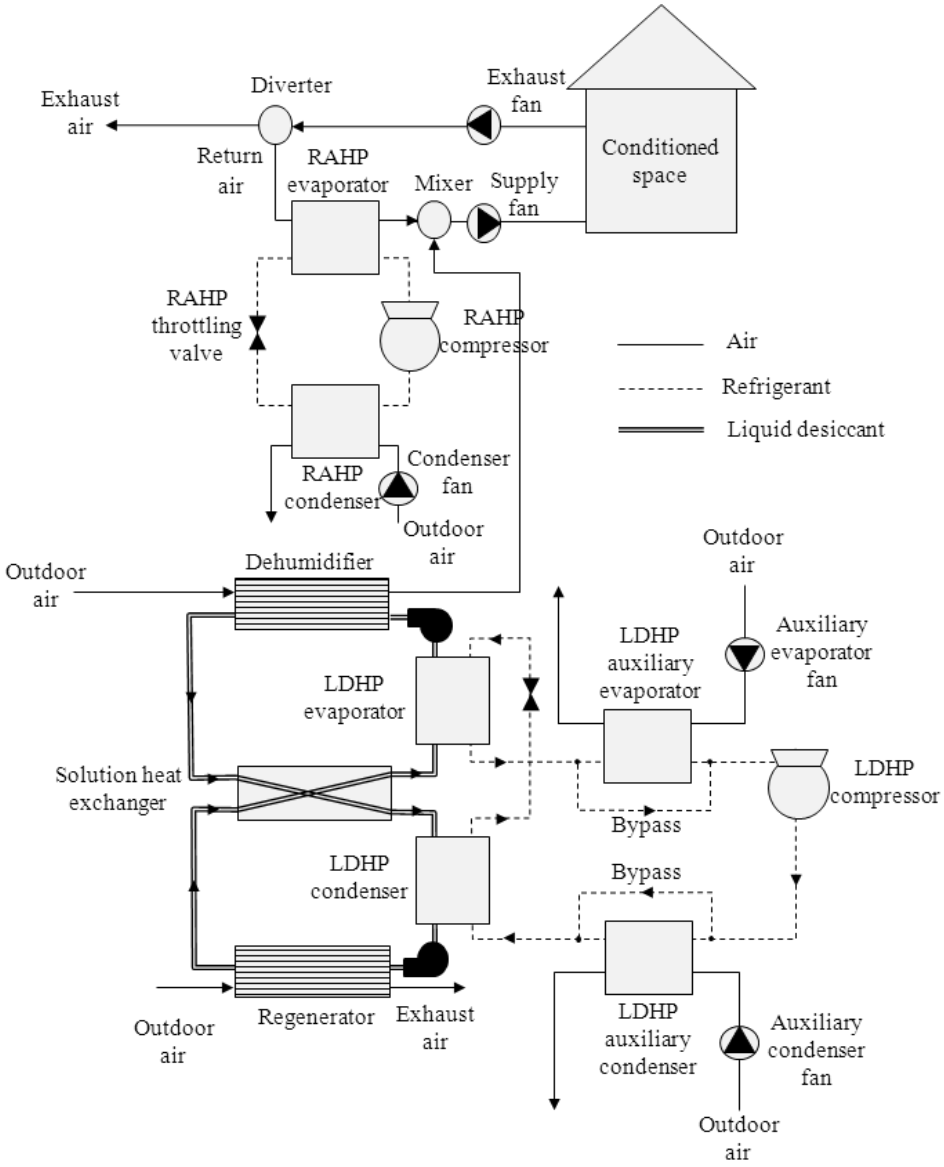

Fig. 2. Schematic diagram for the H-M-LDAC system

The H-M-LDAC system is controlled in the current study using an ON/OFF control methodology, and it operates at one of three operating modes as follows.

- Sensible Mode ($Mode_{sensible}$): This operating mode occurs when the indoor humidity ratio is $\leq$ 9.3 g/kg and the indoor air temperature is > 24°C. In this case, only the RAHP is in operation in order to cool the indoor air down to the set point temperature.

- Latent Mode ($Mode_{latent}$): This operating mode occurs when the indoor air temperature is ≤ 24°C and the indoor air humidity ratio is > 9.3 g/kg. In this case, only the LDHP is in operation in order to dehumidify the indoor air down to the set point humidity ratio.
- Total Mode ($Mode_{total}$): This operating mode occurs when the indoor air temperature is > 24°C and the indoor air humidity ratio is > 9.3 g/kg. In this case, both the LDHP and RAHP are in operation in order to cool and dehumidify the indoor air down to the set point temperature and humidity ratio.

3 Modeling Approach

The H-M-LDAC system proposed in the current study is annually simulated using the TRNSYS building energy software. The H-M-LDAC system is modelled for a small office building located in Miami, FL. The building model was developed in accordance with the ASHRAE Standards 90.1-2007 and 62.1-2004, and a typical meteorological weather data (TMY) file was used to describe the climate in Miami, FL. A detailed description for the building and weather data can be found in a previous study by the authors (Abdel-Salam and Simonson 2014). The modelling approach followed in this study to evaluate the performance of the H-M-LADC system is presented in the following three Sections.

3.1 Liquid-to-Air Membrane Energy Exchanger (LAMEE)

The performance of the LAMEE is evaluated in the current study using an analytical model that was developed by Zhang (2011) for a hollow-fiber LAMEE, and modified by Ge et al. (2013b) in order to be used for a counter-flow flat-plate LAMEE. The analytical model for the counter-flow flat-plate LAMEE was experimentally validated in a previous study by Ge et al. (2014), and its performance was found to be reliable in predicting the performance of the LAMEE. The analytical model is not presented in detail in the current study, and only a flow chart which shows how the performance of the LAMEE is evaluated using the model is presented in Fig. 3. A detailed description for the analytical model is presented in previous studies by the authors (Abdel-Salam and Simonson 2014; Abdel-Salam et al 2013b, 2014; Ge et al. 2014).

3.2 Return Air Heat Pump (RAHP)

The effectiveness of the evaporator/condenser ($\varepsilon_{cond/evap}$) is calculated using the ε-NTU method, Eq. (1), where the $NTU_{cond/evap}$ is assumed to be 2 (Tu et al., 2014). The evaporating temperature (T_{evap}) is calculated as shown in Eq. (3) after determining the operating conditions of the return air stream ($\dot{m}_{air}$, $T_{evap,in}$, $T_{evap,out}$). The condenser used in the RAHP is air cooled, and the condensing temperature (T_{cond}) is assumed to be 10°C higher than the ambient air temperature (i.e. ΔT_{cond}=10°C). After calculating the evaporating and condensing temperatures, the compressor power (W_{comp}) is calculated, Eq. (5), where the theoretical efficiency (η_{th}) of the RAHP is assumed to be 55% (Tu et al., 2014). In order to present a complete evaluation for the performance of the RAHP, the fan power ($W_{fan,cond}$) required to circulate the ambient air in the air-cooled condenser is calculated, Eq. (6), where the pressure drop in the condenser (ΔP_{cond}) is assumed to be 375 Pa and the efficiency of the fan is assumed to be 60%. The COP of the RAHP (COP_{RAHP}) is then calculated, Eq. (7), where the power of both the compressor and condenser's fan are included.

$$\varepsilon_{evap/cond} = 1 - e^{-NTU_{evap/cond}} \tag{1}$$

$$Q_{evap} = \dot{m}_{air,evap} \cdot c_{p,air} \cdot \left(T_{air,evap,in} - T_{air,evap,out}\right) \tag{2}$$

$$T_{evap} = T_{air,evap,in} - \frac{Q_{evap}}{\varepsilon_{evap} \cdot \dot{m}_{air,evap} \cdot c_{p,air}} \quad (3)$$

$$T_{cond} = T_{amb} + 10 \quad (4)$$

$$W_{comp} = \frac{Q_{evap}}{\eta_{th} \cdot \frac{273.15 + T_{evap}}{T_{cond} - T_{evap}}} \quad (5)$$

$$W_{fan,cond} = \frac{Q_{cond} \cdot \Delta P_{air,cond}}{\varepsilon_{cond} \cdot c_{p,air} \cdot \left(T_{cond} - T_{air,cond,in}\right) \cdot \eta_{fan} \cdot \rho_{air}} \quad (6)$$

$$COP_{RAHP} = \frac{Q_{evap}}{W_{fan,cond} + W_{comp}} \quad (7)$$

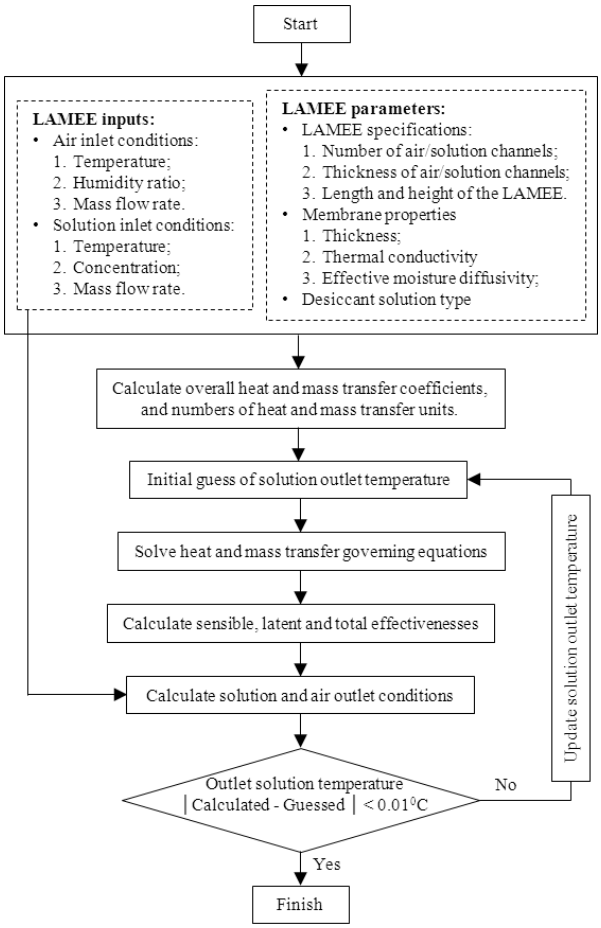

Fig. 3. Flow chart for the analytical model used to evaluate the LAMEE's performance. (Abdel-Salam et al. 2014)

3.3 Liquid Desiccant Heat Pump (LDHP)

Unlike the RAHP which only provides cooling for the return air stream, the LDHP provides simultaneous heating and cooling for the dilute and concentrated solution streams, respectively.

The loads of the condenser and evaporator of the LDHP have to be efficiently matched with the cooling and heating solution loads, respectively, in order to avoid either the overheating of dilute solution stream or the overcooling of concentrated solution stream. For this reason, a matching index is defined in the current study in order to show the relation between the thermal loads of the condenser and evaporator of the LDHP, and the heating and cooling energy required by solution streams. The matching index is defined as follows.

$$HR_{LDHP} = \left[\frac{\frac{Q_{sol,cool}}{Q_{sol,heat}}}{\frac{Q_{evap}}{Q_{cond}}} \right] \tag{8}$$

where, $Q_{sol,cool}$ and $Q_{sol,heat}$ are the cooling and heating energy requirements of the desiccant solution, respectively, Q_{evap} is the evaporator load, and Q_{cond} is the condenser load. In order to assure that the LDHP is operating at the proper operating mode, the HR_{LDHP} is checked at every time step. A flow chart which shows how the operating mode of the LDHP is determined at every time step is presented in Fig. 4. The next two Sections show how the COP_{LDHP} is evaluated at each operating mode (i.e. *Case A or Case B*) for the LDHP.

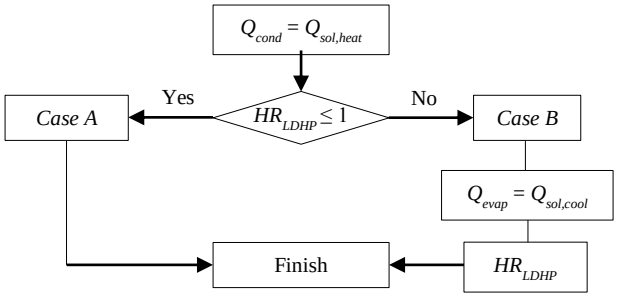

Fig. 4. Flow chart of the operating modes of the LDHP.

Case A:

If $HR_{LDHP} \leq 1$, the condenser load matches the solution heating requirements and the evaporator provides larger cooling energy than that required for the concentrated solution cooling. Thus, an auxiliary evaporator is used in order to reject the additional cooling energy. The required evaporator and condenser loads are calculated using the operating conditions of the concentrated and dilute solution streams, Eqs. (9) and (10). The evaporating temperature is then calculated using Eq. (11), in order to assure that the evaporating temperature does not exceed the ambient air temperature at any operating condition when the air-cooled auxiliary evaporator is used. The condensing temperature is then calculated, Eq. (12), followed by the power of the compressor, Eq. (13). The amount of cooling energy which should be rejected from the auxiliary evaporator is calculated as shown in Eq. (14), and the fan power required to circulate the ambient air through the auxiliary evaporator is calculated using Eq. (15).

$$Q_{evap,sol} = \dot{m}_{sol,evap} \cdot c_{p,sol} \cdot \left(T_{sol,evap,in} - T_{sol,evap,out} \right) \tag{9}$$

$$Q_{cond} = \dot{m}_{sol,cond} \cdot c_{p,sol} \cdot \left(T_{sol,cond,in} - T_{sol,cond,out} \right) \tag{10}$$

$$T_{evap} = Min \left\{ T_{sol,evap,in} - \frac{Q_{evap,sol}}{\varepsilon_{evap} \cdot \dot{m}_{sol,evap} \cdot c_{p,sol}}, T_{amb} - 10 \right\} \tag{11}$$

$$T_{cond} = T_{sol,cond,in} + \frac{Q_{cond}}{\varepsilon_{cond} \cdot \dot{m}_{sol,cond} \cdot c_{p,sol}} \quad (12)$$

$$W_{comp} = \frac{Q_{cond}}{\eta_{th} \cdot \frac{273.15 + T_{cond}}{T_{cond} - T_{evap}}} \quad (13)$$

$$Q_{evap,aux} = \left| Q_{cond} - W_{comp} - Q_{evap,sol} \right| \quad (14)$$

$$W_{fan,evap,aux} = \frac{Q_{evap,aux} \cdot \Delta P_{air,evap,aux}}{\varepsilon_{evap,aux} \cdot c_{p,air} \cdot \left(T_{amb} - T_{evap}\right) \cdot \eta_{fan} \cdot \rho_{air}} \quad (15)$$

Case B:

If $HR_{LDHP} > 1$, the heat pump loads will not be matched with the solution heating and cooling requirements under the aforementioned operating condition (i.e. condenser meets the solution heating load). Thus, the LDHP is operated so that the evaporator load matches with the solution cooling requirements and an additional condenser will be used in order to meet the condensation requirements in the LDHP. The condensing and evaporating temperatures are calculated using Eqs. (12) and (16), respectively. The power of the compressor is then calculated, Eq. (17), and the fan power required to circulate the ambient air in the secondary condenser is evaluated, Eq. (19).

$$T_{evap} = T_{sol,evap,in} - \frac{Q_{evap}}{\varepsilon_{evap} \cdot \dot{m}_{sol,evap} \cdot c_{p,sol}}$$

(16)

$$W_{comp} = \frac{Q_{evap}}{\eta_{th} \cdot \frac{273.15 + T_{evap}}{T_{cond} - T_{evap}}} \quad (17)$$

$$Q_{cond,aux} = \left| Q_{evap} + W_{comp} - Q_{cond,sol} \right| \quad (18)$$

$$W_{fan,cond,aux} = \frac{Q_{cond,aux} \cdot \Delta P_{air,cond,aux}}{\varepsilon_{cond,aux} \cdot c_{p,air} \cdot \left(T_{cond} - T_{amb}\right) \cdot \eta_{fan} \cdot \rho_{air}} \quad (19)$$

Eventually, the COP of the LDHP (COP_{LDHP}) is calculated as follows.

$$COP_{LDHP} = \frac{Q_{sol,cool} + Q_{sol,heat}}{W_{fan,aux,evap/cond} + W_{comp,LDHP}} \quad (20)$$

4 Performance Indices

The performance of the H-M-LDAC system is evaluated using the following indices. (1) The annual COP_{RAHP}, which is calculated as shown in Eq. 7, Section 3.2. (2) The annual HR_{LDHP} and COP_{LDHP} which are calculated as shown in Eqs. 8 and 20, respectively, Section 3.3. (3) The annual COP of the H-M-LDAC system ($COP_{H\text{-}M\text{-}LDAC}$), which is calculated as follows.

$$COP_{H-M-LDAC} = \frac{Q_{air}}{\left(\frac{W_{comp,LDHP} + W_{comp,RAHP} + W_{fans}}{0.33}\right)} \quad (21)$$

where, Q_{air} is the total (sensible and latent) cooling load of the supply air stream, and W_{fans} includes the fans power for the supply and exhaust ducts, the air-cooled condenser of the RAHP, the auxiliary condenser/evaporator of the LDHP. It is worth mentioning that the electrical-thermal conversion coefficient is assumed to be 0.33 in the current study (Abdel-Salam and Simonson, 2014).

5 Results and Discussions

5.1 Annual Performance of LDHP and RAHP

The annual hourly temperature lifts for the RAHP and LDHP are shown in Fig. 5. The temperature lift for the LDHP ($T_{lift,LDHP}$) is found to be higher than that for the RAHP ($T_{lift,RAHP}$) year round. This is because although the evaporating temperature for the LDHP (~17°C) is higher than that for the RAHP (~13°C), the condensing temperature for the LDHP is significantly higher than the RAHP. This is because the LDHP simultaneously covers the heating and cooling requirements of the desiccant solution, which means that the LDHP is operated at high condensing temperatures in order to heat the diluted desiccant up to 50°C before being supplied to the regenerator. In addition, it can be seen from Fig. 5 that unlike the $T_{lift,RAHP}$ which fluctuates depending on the ambient conditions, the $T_{lift,LDHP}$ is almost constant during all operating conditions. The fluctuation of the $T_{lift,RAHP}$ is because the condenser of the RAHP is air cooled, which means that the condensing temperature depends on the ambient air temperature. On the other hand, the evaporating and condensing temperatures of the LDHP are mainly determined based on the desiccant solution set points which are maintained constant, as previously mentioned, except for the startup of the system and the few cool operating hours where the evaporating temperature is determined based on the ambient temperature as shown in Eq. 11.

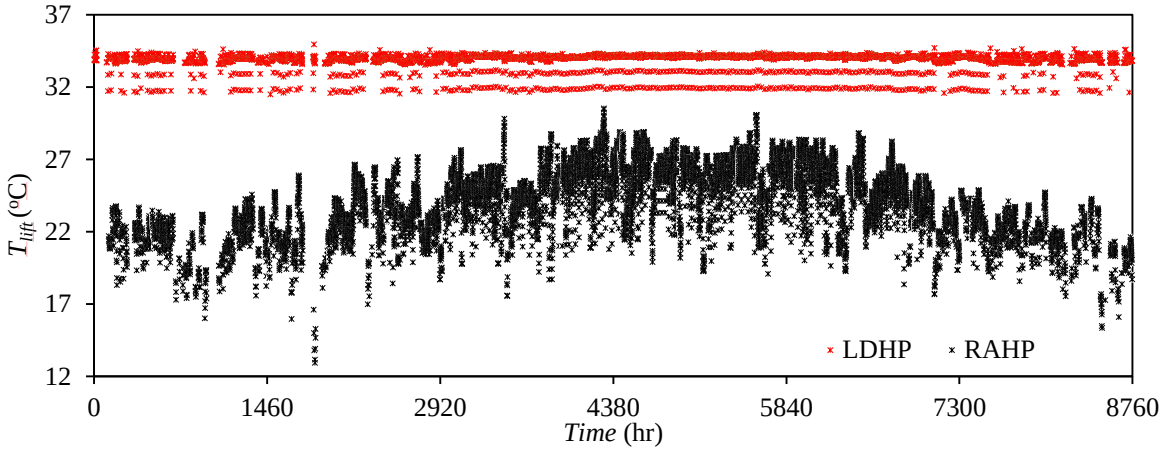

Fig. 5. The annual hourly temperature lifts for the RAHP and LDHP.

Fig. 6 shows the annual hourly values and the frequency of occurrence for the COP_{LDHP} and COP_{RAHP}. Although $T_{lift,RAHP}$ is lower than $T_{lift,LDHP}$, it is found that the COP_{LDHP} is higher than the COP_{RAHP}. The COP_{RAHP} lies between in the range of 3.8-6.8, while the COP_{LDHP} lies in the range of 4.2-10.6. This is expected due to the fact that the LDHP provides simultaneous heating and

cooling for the desiccant solution, which means that the LDHP is operated in a more energy efficient manner compared to the RAHP which provides only cooling for the return air. It is found that the ambient air conditions have an opposite influence on the COP_{LDHP} compared with the COP_{RAHP}, as can be seen from Fig. 6a. The COP_{RAHP} is found to decrease during the hot summer months, and improves during the mild winter months. This is expected for the RAHP, as the higher the ambient air temperature the higher the condensing temperature, which decreases the COP_{RAHP}. On the other hand, the COP_{LDHP} is found to have an opposite trend, where it improves during the hot summer months and decreases during the mild winter months.

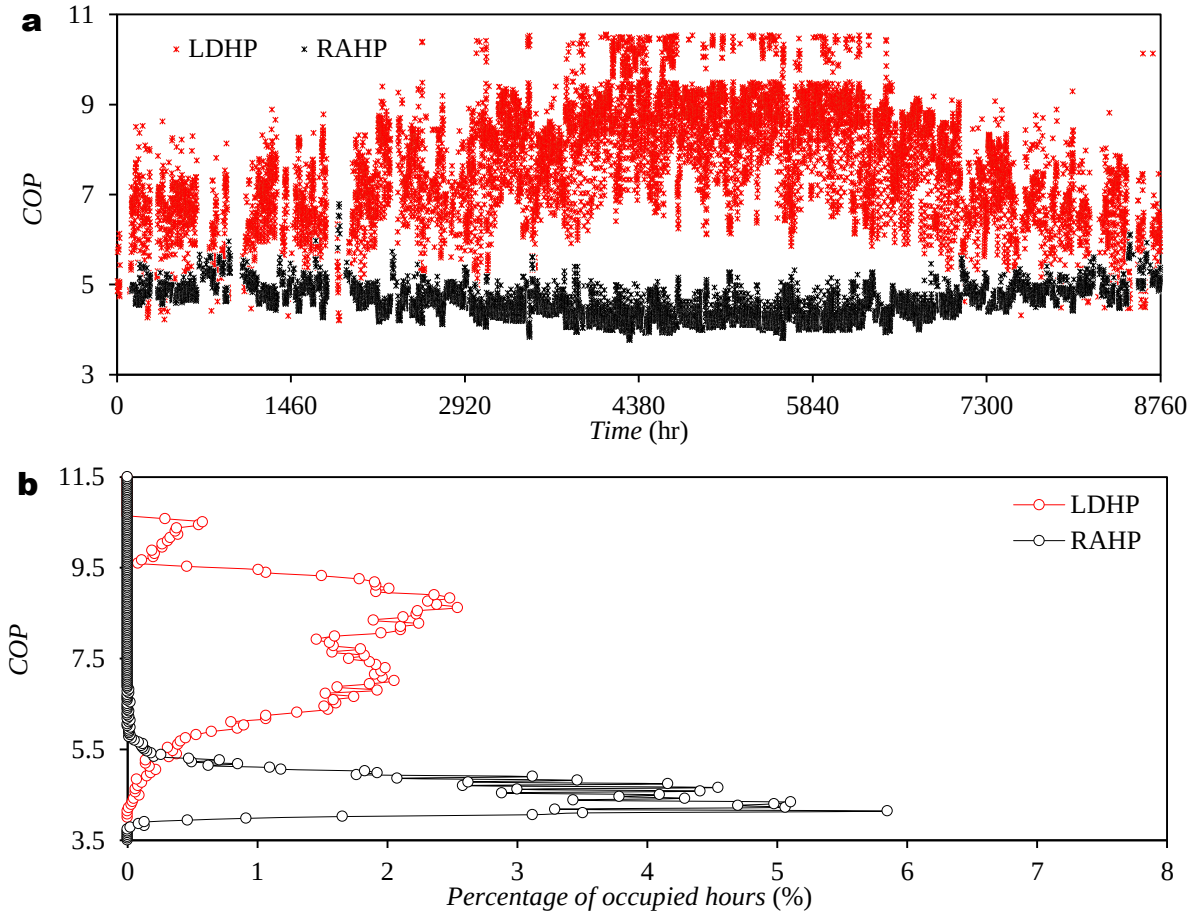

Fig. 6. The (a) annual hourly values and (b) frequency of occurrence for the COP_{LDHP} and COP_{RAHP}.

As previously discussed, there are two operating modes for the LDHP (*Case A* and *Case B*). It is necessary to track the operating mode of the LDHP in order to assure that the heating and cooling requirements of the desiccant solution are efficiently matched with the condenser and evaporator loads of the LDHP at different operating condition. As can be seen from Fig. 7, it is found that the LDHP operates at *Case A* operating mode for 95% of the operating hours, and operates at *Case B* for 5% of the operating hours. In *Case A*, the solution cooling requirements range from 30% to 100% of the evaporator load, depending on the operating conditions, with an average value of 77%. While in *Case B*, the condenser load ranges from 100% to 130% of the solution heating requirements, depending on the operating conditions, with an average value of 106%. It is worth mentioning that this shows a potential for improving the COP_{LDHP} by using the available

cooling energy when the LDHP operates at *Case A* to provide a portion of the sensible cooling required for the return air. This is expected to cause an observable improvement in the energy efficiency of the LDHP, and consequently in the energy efficiency of the H-M-LDAC system. This point will be investigated in detail in future work by the authors.

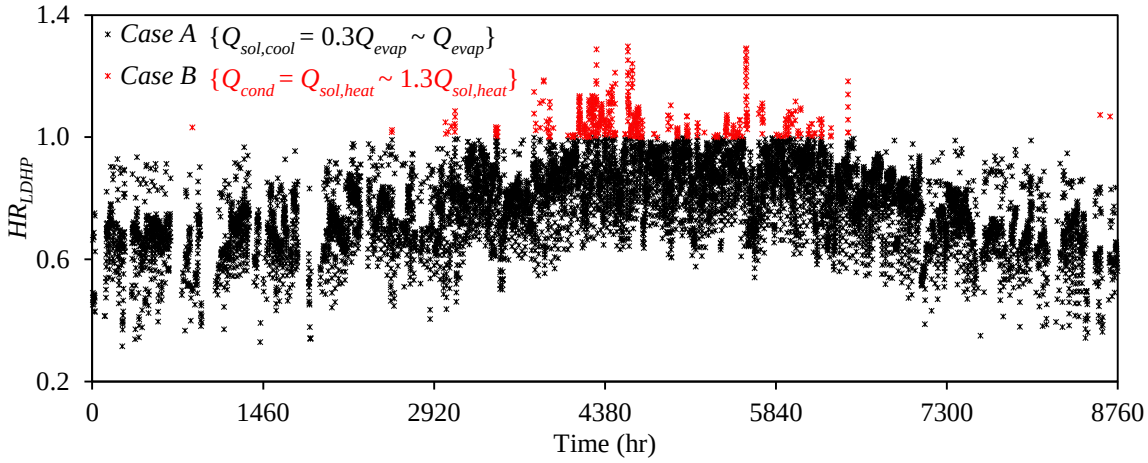

Fig. 7. The hourly annual heat ratio between the solution heating and cooling requirements, and the evaporator and condenser loads of the LDHP (HR_{LDHP}).

5.2 Annual Performance of H-M-LDAC System

After evaluating the performance of the LDHP and RAHP, the performance of the whole system is analyzed in the current Section. Fig. 8 shows the annual hourly values and frequency of occurrence for the COP of the H-M-LDAC system ($COP_{H\text{-}M\text{-}LDAC}$). It is found that the $COP_{H\text{-}M\text{-}LDAC}$ is not maintained within specific range year around. The H-M-LDAC system operates for approximately 15%, 21% and 63% of the operating hours at $Mode_{sensible}$, $Mode_{latent}$ and $Mode_{total}$, respectively (i.e. the three operating modes are defined in Section 2). It is clear from Fig. 8 that the operating mode has a significant impact on the $COP_{H\text{-}M\text{-}LDAC}$, where the $COP_{H\text{-}M\text{-}LDAC}$ is the lowest during $Mode_{latent}$ and the highest during $Mode_{sensible}$.

Since the $COP_{H\text{-}M\text{-}LDAC}$ is dependent on both the outdoor air conditions, and the cooling and dehumidification requirements of the conditioned space, the $COP_{H\text{-}M\text{-}LDAC}$ is plotted versus the cooling load in order to have a better understanding of the performance of the H-M-LDAC system, see Fig. 9. It can be seen that the $COP_{H\text{-}M\text{-}LDAC}$ is strongly correlated with the cooling load when the H-M-LDAC system operates at either $Mode_{latent}$ or $Mode_{total}$, with larger slope during $Mode_{latent}$ compared with $Mode_{total}$. It can be seen that the $COP_{H\text{-}M\text{-}LDAC}$ is high during $Mode_{sensible}$ which is believed to be due to the following. The H-M-LDAC system is controlled using an ON/OFF control methodology, as previously mentioned. During $Mode_{sensible}$, the LDHP is turned off; however, the desiccant solution stream might still cool and dehumidify the fresh air stream for a few time steps, depending on the operating conditions, but at lower capacity due to the quick variations that would occur in the dilute and concentrated desiccant solution streams when the LDHP is turned off. Thus, although there is no energy consumed by the LDHP during $Mode_{sensible}$, the desiccant cycle causes a cooling effect for the fresh air stream, which when added to the sensible cooling caused by the RAHP will result in high $COP_{H\text{-}M\text{-}LDAC}$.

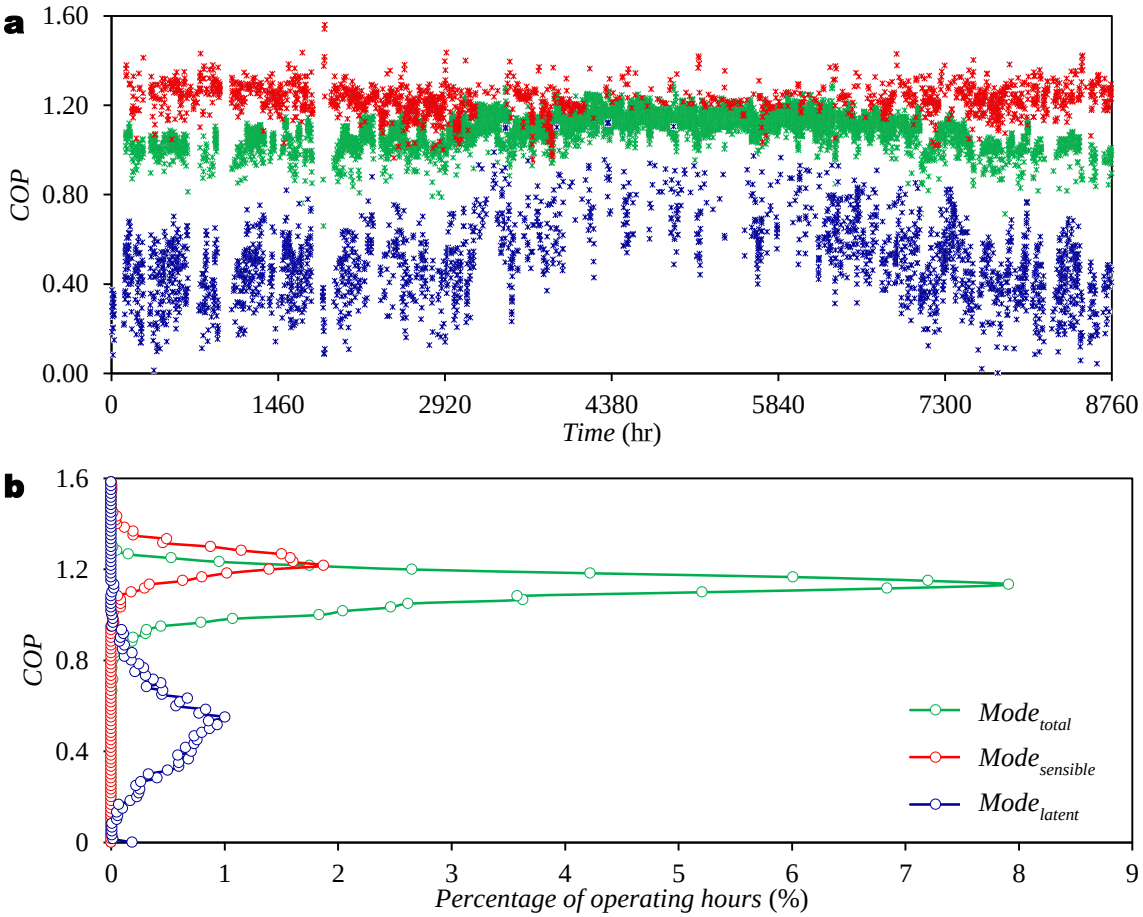

Fig. 8. The (a) annual hourly values and (b) frequency of occurrence for the $COP_{H\text{-}M\text{-}LDAC}$.

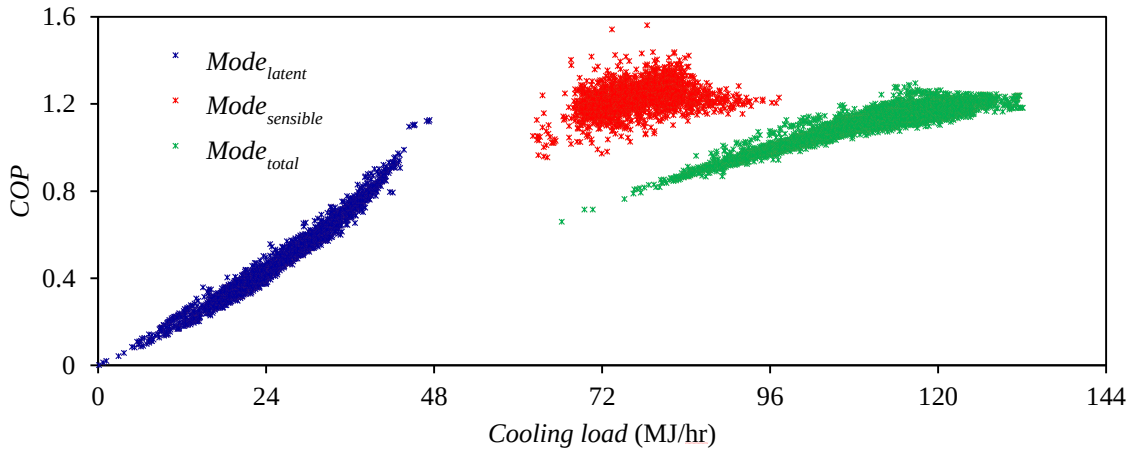

Fig. 9. The annual hourly $COP_{H\text{-}M\text{-}LDAC}$ versus the annual hourly cooling loads.

5.3 Sensitivity Analysis of LDHP and RAHP Parameters

In this Section, the influences of the $NTU_{cond/evap}$, $\Delta P_{cond/evap}$, η_{th} and ΔT_{cond} on the $COP_{H\text{-}M\text{-}LDAC}$ are evaluated. Each of the investigated parameters is varied by up to ±20% of its base value in ±5% intervals, and the equivalent percent change in the $COP_{H\text{-}M\text{-}LDAC}$ is evaluated. Fig. 10 shows

the variation of the $COP_{H\text{-}M\text{-}LDAC}$ due to the different variations performed in the $NTU_{cond/evap}$, $\Delta P_{cond/evap}$, η_{th} and ΔT_{cond}. It is found that ΔT_{cond} has the lowest influence on the $COP_{H\text{-}M\text{-}LDAC}$, while η_{th} has the largest influence. It is clear that the assumptions used to evaluate the performance of the heat pumps have considerable influences on the results. Thus, future work will include developing more advanced heat pump model.

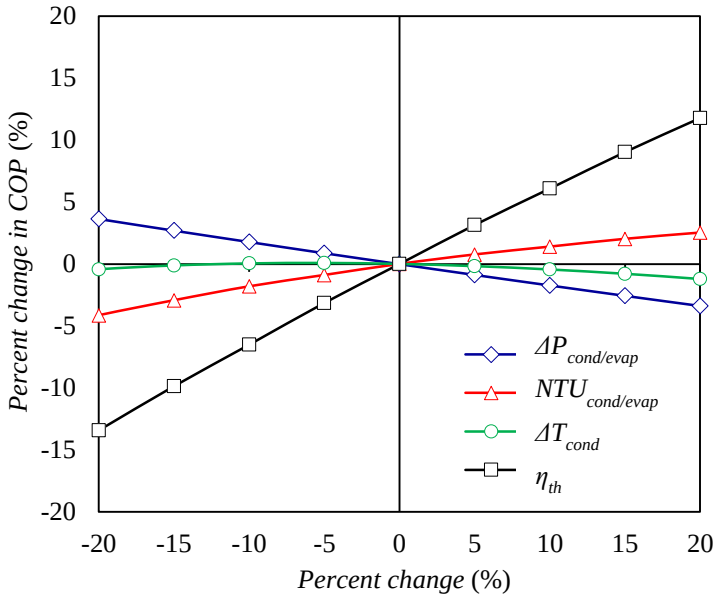

Fig. 10. The influence of the $NTU_{cond/evap}$, $\Delta P_{cond/evap}$, η_{th} and ΔT_{cond} on the $COP_{H\text{-}M\text{-}LDAC}$.

6 Conclusions

The annual COP of a hybrid membrane liquid desiccant air conditioning (H-M-LDAC) system was evaluated in this study using the TRNSYS building energy simulation software for a small office building located in Miami, FL. The main advantages of the proposed H-M-LDAC system are: (1) the elimination of the desiccant droplets carryover problem through the use of liquid-to-air membrane energy exchangers (LAMEEs) as the dehumidifier and regenerator, and (2) the simultaneous heating and cooling of the desiccant solution using a hybrid liquid desiccant heat pump (LDHP).

It is found that the annual hourly COP of the LDHP ranges between 4.2 and 10.6, with an annual average value of 7.8. The annual hourly COP of the return air heat pump (RAHP) ranges between 3.8 and 6.8, with an annual average value of 4.5. A new matching index was defined for the hybrid LDHP in this study, where it was found that there is auxiliary cooling energy which is provided by the LDHP for 95% of the operating hours. This is promising, because considerable reductions is expected to be achieved in the return air cooling requirements if the auxiliary cooling energy provided by the LDHP is used to precool the return air before it enters the RAHP. Based on the evaluated COPs for the RAHP and LDHP, the average annual COP of the H-M-LDAC system is found to be 1. The COP of the H-M-LDAC system is found to be strongly influenced by the cooling loads and the mode of operation of the system. The influence of four heat pump parameters was investigated on the COP of the H-M-LDAC system, where the theoretical efficiency of the heat pump is found to be the most influential parameter.

The authors strongly recommend that future research on LDAC systems focus on two main points. First, the development of novel conditioners (dehumidifier/regenerator), in order to

improve the performance of LDAC systems, and eliminate the problems associated with the use of available conditioners. Second, since LDAC system includes several components, it is important to keep investigating the performance of novel energy efficient LDAC systems configurations, which improves the energy efficiency for these systems.

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Nomenclature

ΔP	pressure drop (Pa)
COP	coefficient of performance
c_p	specific heat capacity(J/(kg.K))
HR	matching index
$\dot{m}$	mass flow rate (kg/s)
NTU	number of heat transfer units
Q	rate of energy consumption (J/s)
T	temperature (^{o}C)

Greek letters

η_{th}	thermal efficiency of heat pump
ε	effectiveness
ρ	density

Subscripts

amb	ambient air
aux	auxiliary
cond	condenser
cool	cooling
evap	evaporator
heat	heating
in	inlet
out	outlet
sol	desiccant solution

Abbreviations

H-M-LDAC	hybrid membrane liquid desiccant air conditioning
LAMEE	liquid
LDAC	liquid desiccant air conditioning
LDHP	liquid desiccant heat pump
RAHP	return air heat pump

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