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Recent advances in semi-open ionic liquid absorption heat pump for separate sensible and latent cooling

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Abstract

Cooling systems are a necessity of modern civilization that have greatly enhanced our standard of living and enabled the development of large population centers in harsh climates. However, a key shortcoming of the existing prominent cooling system i.e., vapor compression system is the need for overcooling of air to condense its moisture content. Here, we report on recent advancements in absorption-based dehumidification using ionic liquids (ILs), focusing on their integration into a novel semi-open cycle architecture to achieve high energy efficiency in ventilation air treatment. Building on our recent work in ionic liquid design, we demonstrate how tailored ILs with high affinity for water vapor and low regeneration temperatures can significantly enhance moisture removal performance while simplifying system components. In parallel, we introduce a semi-open absorption cycle that eliminates the need for outdoor scavenging air in the regenerator—a longstanding inefficiency in traditional open desiccant systems. Experimental evaluation of this cycle under a range of outdoor conditions reveals marked improvements in thermal efficiency, including a dehumidification coefficient of performance (COP) reaching 0.9 and peak latent cooling capacity at elevated dew points. This architecture substantially reduces sensible heat loss during regeneration, enabling more effective decoupling of latent and sensible loads in the Separate Sensible and Latent Cooling (SSLC) framework. Together, these advancements in IL formulation and cycle design address critical challenges in humidity control and offer scalable, climate-resilient solutions for reducing the energy footprint of building ventilation. The findings reinforce the potential of ionic liquid-based absorption systems to redefine conventional dehumidification strategies and support sustainable indoor environmental management.

Keywords: Absorption; Dehumidification; Liquid desiccant; Ionic liquid.

1. Introduction

Cooling systems are responsible for nearly 10% of the global electricity consumption [1]. This energy use is steadily rising due to the increase in the number of installed cooling systems and climate temperature [2]. This trend presents significant challenges for decarbonization and the management of electrical grid loads, particularly in hot and humid climates, where cooling demand can exceed 70 % of the peak power demand [1]. The dehumidification (latent cooling) of outdoor ventilation air is a major contributor to the overall cooling demand [3]. Cooling systems, based on vapor compression cycles, must overcool air to saturation conditions to remove moisture from air. In certain applications, the air is reheated to maintain comfortable conditions. This subcooling-reheating process is responsible for significant energy consumption. The use of a more

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thermodynamically efficient process to remove moisture from air has the potential to improve energy efficiency in HVAC systems.

Liquid desiccant dehumidification technology provides an alternative approach to the energy intensive subcooling-reheating process by extracting moisture from the air without reducing its temperature. Although the liquid desiccant latent cooling (dehumidification) process is inherently more efficient than the subcooling-reheating process used in current vapor compression systems, its adoption within the market has been limited due to its relatively low energy efficiency, characterized by a low coefficient of performance (COP), defined as the ratio of the latent heat removed from the process air versus the heat supplied to the cycle. Conventional liquid desiccant systems consist of two primary components: an absorber and a regenerator, as depicted in Figure 1A. In the absorber, driven by vapor pressure differential, the desiccant solution absorbs water vapor from the process air. The resulting water-rich i.e., weak desiccant solution leaving the absorber is pumped to the regenerator. Heat is supplied to the desiccant within the regenerator to liberate the absorbed water while scavenging air, from ambient, is used to carry away the water vapor from the regenerator. The resulting strong desiccant solution exits the regenerator and returns to the absorber, while transiting through a heat recovery exchanger.

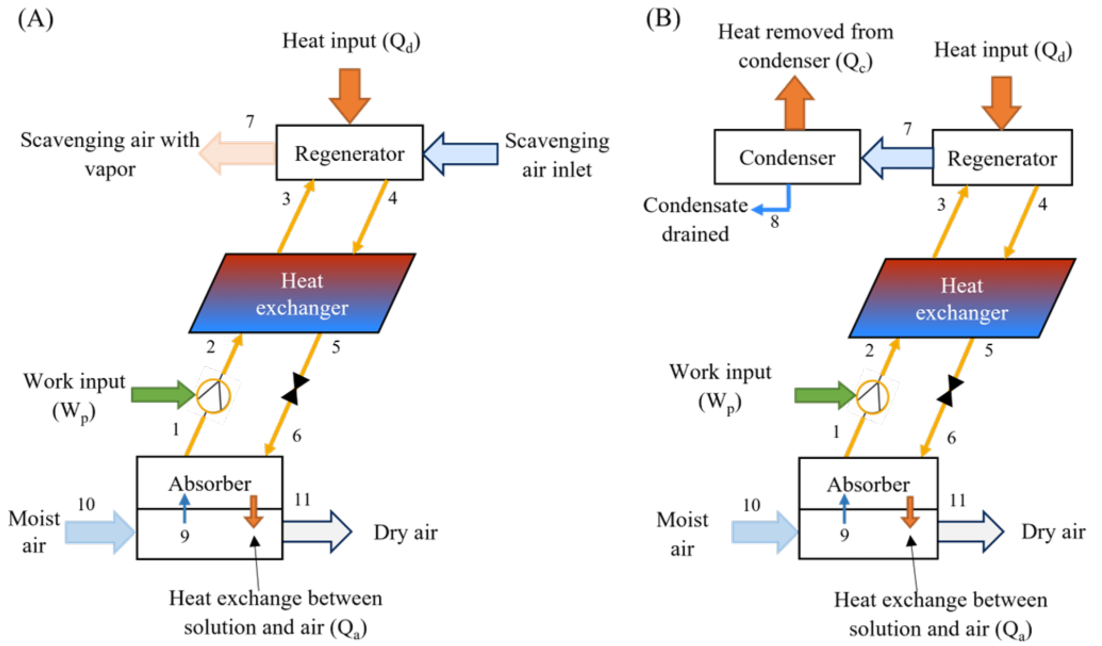

Figure 1. Schematic diagram showing an open liquid desiccant absorption cycle using scavenging air to extract moisture from the desiccant solution (A), semi-open liquid desiccant absorption cycle using a condenser to condense the extracted moisture from the desiccant solution (B).

When scavenging air is used to remove moisture from the desiccant solution, it also extracts a significant amount of sensible heat due to the temperature difference between the desiccant solution in the regenerator and the scavenging air. Under high humidity conditions, heat loss to the scavenging air becomes more pronounced. This is because the regenerator experiences a higher moisture load, since the absorption system has to remove more moisture from the ventilation air. Simultaneously, greater amounts of scavenging air are needed to regenerate the desiccant solution, as the differential water vapor pressure between the desiccant and scavenging air decreases with increasing inlet humidity. Consequently, for the same regeneration temperature and higher moisture removal demands, sensible heat loss to the scavenging air also increases. This sensible heat loss from the desiccant solution to the scavenging air increases the heat energy input required in the regenerator, thereby reducing the dehumidification efficiency per unit of moisture removed from the desiccant solution. A cycle utilizing scavenging air has several additional drawbacks. First, implementing the system in retrofit applications can be challenging due to the need for new ductwork. Second, because ambient air comes into direct contact with the liquid desiccant, filtration is necessary. Third, additional control equipment is required to modulate the system's heat input in response to changes in ambient temperature and humidity.

Our recently developed semi-open system (Figure 1B) [4], utilizes a combined regenerator-condenser module in which an indirectly cooled condenser is used to condense the water vapor emanating from the desiccant solution, thereby eliminating the need for scavenging air, alleviating its associated issues. The condenser heat is discharged to the ambient using a hydronic loop. Outdoor temperatures lower than the design conditions reduce the condensation pressure facilitating water vapor liberation from the desiccant.

In this study, we report on recent advancements in absorption-based dehumidification using ionic liquids (ILs), focusing on their integration into a novel semi-open cycle architecture to achieve high energy efficiency in ventilation air treatment. Building on our recent work in ionic liquid design, we demonstrate how tailored ILs with high affinity for water vapor and low regeneration temperatures can significantly enhance moisture removal performance while simplifying system components. In parallel, we introduce a semi-open absorption cycle that eliminates the need for outdoor scavenging air in the regenerator—a longstanding inefficiency in traditional open desiccant systems [5–9]. Experimental evaluation of this cycle under a wide range of outdoor conditions reveals marked improvements in thermal efficiency, including a dehumidification coefficient of performance (COP) reaching 0.9 and peak latent cooling capacity at elevated dew points [10–12]. This architecture substantially reduces sensible heat loss during regeneration, enabling more effective decoupling of latent and sensible loads in the Separate Sensible and Latent Cooling (SSLC) framework. Together, these advancements in IL formulation and cycle design address critical challenges in humidity control and offer scalable, climate-resilient solutions for reducing the energy footprint of building ventilation. The findings reinforce the potential of ionic liquid-based absorption systems to redefine conventional dehumidification strategies and support sustainable indoor environmental management.

2. Experimental System

The semi-open cycle (Figure 1B) uses compact plate and frame heat and mass exchangers which enhances the system's surface area to volume ratio, thereby reducing its overall size. The use of an ionic liquid as the desiccant solution mitigates the crystallization issue, as ionic liquids, which are hygroscopic molten salts, do not crystallize at any conditions within the US climate zones, thus expanding its operational envelope and reducing its control related cost. In dehumidification systems, a combined condenser/regenerator module is used so that the condenser indirectly condenses and removes the water vapor released from the regenerator within an enclosed module, thereby eliminating the heat loss associated with the use of scavenging air in the regenerator. Consequently, the condenser heat is discharged to the ambient using a hydronic loop and a conventional liquid-air heat exchanger. This approach isolates the regenerator from the impacts of change in the outdoor temperature and humidity thereby reducing the system complexity.

Figure 2 shows a piping and instrumentation flow diagram of the experimental system. The system functions in a cyclical manner, where water vapor present in the airflow is collected in the absorber and then released in

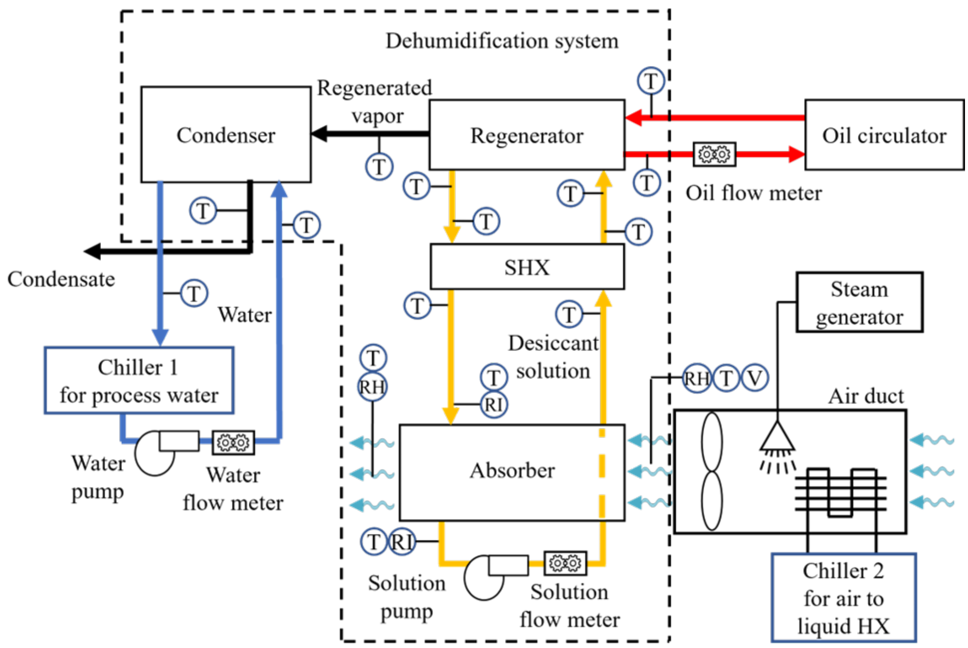

Figure 2. Diagram of the test system including all essential equipment with the desiccant flow shown in yellow, cooling water shown in blue and heating oil in red.[10]

the regenerator through heating, as previously described. The water vapor released from the desiccant solution condenses, undergoes metering, and is subsequently drained.

2.1. Ionic Liquids

The properties of the liquid desiccant profoundly impact the efficacy of open and semi-open cycles in absorption systems. Traditional choices like lithium bromide, lithium chloride, and calcium chloride, being brine-based solutions, are inherently prone to crystallization. It restricts system operational ranges, requiring the precise control of temperature and concentration parameters to prevent crystallization. This limitation results in underutilized desiccant capacity, necessitating elevated flow rates and larger surface areas within the absorber and desorber. ILs characterized as low temperature molten salts with hygroscopic properties, offer a solution to crystallization challenges of conventional brine desiccants. ILs are composed of large asymmetric organic cations and inorganic or organic anions that are moisture and air stable at room temperature [13,14] Incorporating ILs broadens absorption systems' operational ranges, enabling them to function within a significantly extended envelope, eliminating crystallization concerns. This eliminates the need for complex control systems, reducing system complexity and cost while enhancing efficiency.

ILs exhibit properties vital for absorption dehumidification, including thermal stability, low vapor pressure, high water solubility, and low metal corrosion, making them promising for absorption systems [15]. The desiccant's dehumidification capability, a critical thermo-physical property, is best represented by its water vapor pressure at specific concentrations and temperatures. Figure 3A shows a comparison of the vapor pressure between some ILs and conventional liquid desiccants, such as LiBr and LiCl. This comparison shows that [EMIM][OAc] and the IL used in this study can provide vapor pressures like LiCl and LiBr solutions but must be concentrated to a greater degree to achieve the desired low vapor pressure.

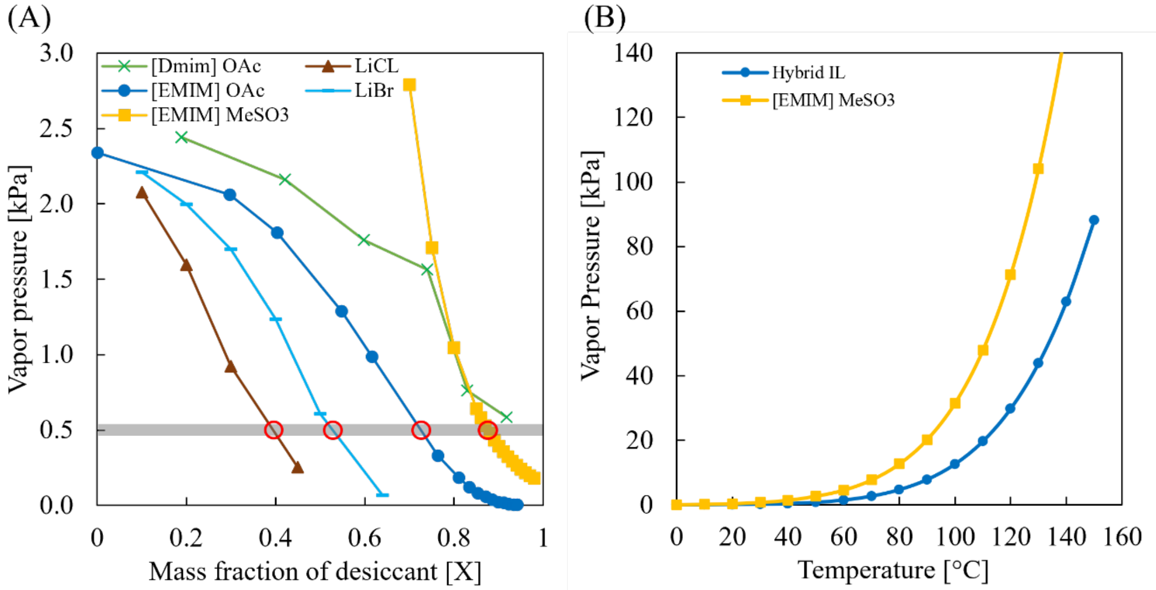

Figure 3. Comparison of equilibrium vapor pressure vs. concentration of various liquid desiccants at 20 °C (A)[15], and the vaper pressure depression of our newly developed hybrid IL (B).

Another key factor that dictates the dehumidification system design and whether an IL-based system can be successfully built is the IL viscosity. Our studies [6–9], have shown that the IL viscosity impacts the design of middle and low temperature components of the system the most, consisting of the absorber and solution heat exchanger. The high operating temperature of our desorber greatly reduces the IL solution viscosity such it is not the prominent factor in design considerations. Viscosity is a key constraint in design of the solution heat exchanger when the IL flow from the desorber to absorber is driven by gravity. Furthermore, design of the IL flow distribution within the absorber is impacted by the IL viscosity. Hence, while assessing utility of one IL versus another, their vapor pressure is considered in relation to their viscosity. This is illustrated in Figure 3A with its accompanying Table 1. The few example ILs shown in this graph meet a suitable vapor pressure of 0.5 kPa at different concentrations as shown with the red circles. Table 1 compares viscosity of the ILs that meet the water vapor pressure requirement, suggesting that while viscosity of the two example ILs is similar, it is substantially higher than that of the LiBr and LiCl. It is imperative to mention that water molecules diffusion coefficient through ILs is directly impacted by the IL viscosity, meaning that a higher viscosity leads to a lower mass transfer coefficient and consequently lower absorption (most relevant to closed systems) and desorption rates. Hence, past studies that have only replaced ILs in existing systems have observed significant deterioration in performance.

Table 1. Dynamic viscosity comparison for different liquid desiccants.

Desiccant	Viscosity (Pa-s)
LiCl (20 °C, 40%)	0.0096
LiBr (20 °C, 53%)	0.0051
[EMIM]OAc (20 °C, 73%)	0.0233
[EMIM] MeSO3 (20 °C, 87%)	0.0283
Hybrid IL	0.0333

Recently we have begun the hybridization of ILs, through the mixture of existing ILs, to lower the water vapor pressure at specific concentrations and temperatures, without sacrificing IL stability across the system operating temperature range. This effort has resulted in the development of a Hybrid IL that is stable, has a similar viscosity, and a lower water vapor pressure (Figure 3B). While [EMIM] MeSO3 has been used in this study, this new hybrid IL is supporting current testing.

2.2. Measurement System

The solution flow rate is measured using a positive displacement flow meter. Due to the elevated temperature of the oil flow loop, a high-temperature positive displacement flow meter is used to measure the oil flow rate. The water flow rate is measured using a turbine flow meter. All flow meters were calibrated in house using a bucket-stopwatch method. The airflow rate is measured using a pitot tube sensing element mounted within the air loop duct. Multiple sensing ports along the length of the sensing element are used, and internally averaged to ensure accurate airflow rate measurement. All liquid temperatures were measured using T-type thermocouples. The process air temperature and relative humidity are measured using thin-film capacitive sensors. The absorber solution inlet and outlet concentrations are measured using two inline refractive index (RI) sensors. RI instead of Coriolis effect is used because the variation in density for ILs is insignificant over a wide concentration range, making accurate measurement of the concentration infeasible. Curve fits have been developed to establish RI values as a function of IL concentration and temperature. The primary system temperature, flow rate, RI, and air velocity are recorded using a data acquisition system. A second data acquisition system is used to record the air temperature and relative humidity. Time stamps are used to combine the data collected by each data acquisition system to enable a system level data analysis.

2.3. Data reduction and uncertainty analysis

The system performance presented in the results and discussion section is calculated based on the measured data described above. The system latent capacity is calculated using Eq. (1).

$$Q_{latent} = \dot{V}_{air}\,\rho_{air}\,h_{fg}\,(\omega_{air,in} - \omega_{air,out}) \tag{1}$$

where $\dot{V}_{air}$ is the absorber airflow rate; $\omega_{air,in}$ and $\omega_{air,out}$ are the absorber air inlet and exit absolute humidity, respectively; ρ_{air} is the air density as calculated from standard tables using the measured air inlet conditions; and h_{fg} is the water latent heat of evaporation. The desorber heat input is calculated using Eq. (2).

$$Q_{oil} = \dot{V}_{oil}\,\rho_{oil}\,c_{p,oil}(T_{des\ oil,in} - T_{des\ oil,out}) \tag{2}$$

where $\dot{V}_{oil}$ is the desorber oil flow rate; $T_{des\ oil,in}$ and $T_{des\ oil,out}$ are the desorber oil inlet and outlet temperatures, respectively; ρ_{oil} is the oil density; and $c_{p,oil}$ is the oil thermal capacity calculated using curve fits derived from data provided by Clearco for their SIL 180 silicone oil. The system latent heat capacity along with the desorber heat input are used in Eq. (3) to calculate the system COP.

$$COP = \frac{Q_{latent}}{Q_{oil}} \tag{3}$$

The absorber moisture removal rate (MRR) is calculated using Eq. (4).

$$MRR = \dot{V}_{air}\,\rho_{air}\,(\omega_{air,in} - \omega_{air,out}) \tag{4}$$

The circulation ratio (CR) in the system is calculated using Eq. (5).

$$CR = \frac{\dot{m}_{sol}}{MRR} \tag{5}$$

The sensible heat loss to the condenser in the regenerator-condenser module is determined by adjusting the water flow rate in the condenser so that the heat gained by the cooling water matches the heat lost by the heating oil in the regenerator. The actual heat loss in then calculated using Eq. (6).

$$Q_{loss} = \dot{V}_{water}\,\rho_{water}\,C_{p,water}(T_{water,out} - T_{water,in}) \tag{6}$$

The uncertainty was calculated using an engineering equation solver (EES) uncertainty propagation subroutine. The subroutine is based on NIST guidelines[16] and assuming that the individual measurements are uncorrelated and random, the uncertainty in the calculated quantity can be estimated using Eq. (7).

$$U_Y = \sqrt{\sum_i \left[\frac{\partial Y}{\partial X}\right]^2 U_{X_i}^2} \tag{7}$$

Table 1 below lists all the relevant measurement errors and uncertainties in this experimental study.

Table 1. Measurement error and uncertainty propagation.

Variable	Uncertainty
Measured variables	
Fluid temperatures -All T-type TC	±0.8 °C
Desiccant flow rate	±0.25% reading
Desiccant RI	±0.5% scale range (0.0003nD)
Water flow rate	±1% full scale (0.5 LPM)
Air volumetric flow rate	±2% reading
Oil flow rate	±0.5% reading
Air temperature	±0.3% reading
Air relative humidity	±1.8% RH
Calculated variables	
Q_{latent}	±4.3%
Q_{oil}	±3.2%
$COP_{dehumid}$	±5.4%
MRR	±4.7%
CR	±7.1%

2.4. Test Conditions

Table 2 lists the wide range of operating conditions used to enable a comprehensive evaluation of the system performance. This includes all ISMRE conditions [17], one IEER condition [18], and various other inlet air conditions, at an overall dew point temperature range of ~10-35 °C. The IEER and ISMRE conditions provide a standardized benchmark and are used extensively in the HVAC industry for comparing the performance of different dehumidification systems. The desiccant solution, process air flow, and heating oil flowrates along with the inlet temperature of the heating oil have been maintained at constant values of ~300 mLPM, ~150 CFM, ~2.80 LPM, and ~150 °C, respectively.

Table 2. Inlet air test conditions

Test	Inlet air conditions

condition	Dry bulb temp. (°C)	Relative humidity (%)	Dew point temp. (°C)
1	35.0	90.0	33.1
2	32.0	80.0	28.1
3	26.8	84.5	24.0
4	30.0	70.0	23.9
5	35.0	47.3	22.1
6	26.7	72.0	21.2
7	21.1	81.4	17.8
8	26.7	51.1	15.8
9	17.2	79.4	13.6
10	20.0	54.64	10.3

3. Results and Discussion

3.1. Dehumidification Capacity

As shown in Figure 4A the system's dehumidification capacity (i.e. latent cooling) increases as the inlet air dew point temperature increases. This rise in performance is attributed to the growing differential water vapor pressure, stemming from the elevated outdoor air water vapor pressure. This increase in the absorption rate is supported by the desiccant solution's ability to continue absorbing more water. The linear increase in the absorber latent capacity associated with tests at high dew points i.e., 24, 28.1, and 33.1 °C clearly suggests that the solution is yet to be saturated. A dew point temperature of 33.1°C is the maximum achievable in our experimental system. It is important to mention that in the United States, the highest dew point ever recorded is 32 °C, measured at the New Orleans Naval Air Station, on July 30, 1987, and Melbourne, Florida on July 12, 1987.

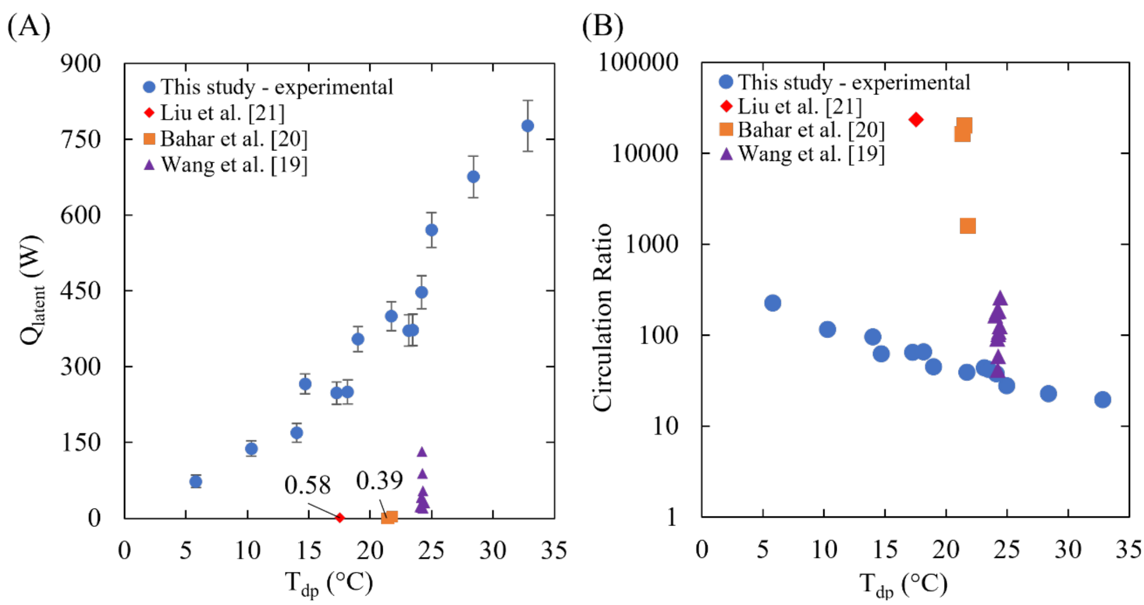

Figure 4. Experimental latent capacity of the system (A), and experimental circulation ratio of the system (B) as a function of the inlet air dew point temperature. The desiccant solution, process air, and heating oil flowrates along with the inlet temperature of heating oil and condenser cooling water have been maintained at constant values of 311.8 mLPM, 156.7 CFM, 2.78 LPM, 147.6 °C, and 25.2 °C, respectively.

Figure 4A also compares the dehumidification capacity of our system with the recent experimental results[19–21] involving membrane-based absorbers and ILs. These studies are component level studies and do not operate as a continuous cycle. Figure 4B compares the circulation ratios (eq 5) obtained in our study with the recent experimental results[19–21]. The lower circulation ratio of the system used in this study (Figure 4B) suggest it absorbs more moisture per unit of desiccant mass flow and would therefore have a higher dehumidification efficiency.

Wang et al.[19] and Bahar et al. [20] used hollow fiber ionic membranes in tube bundle configuration whereas Liu et al. [21] used a semi-permeable membrane, as in this study. All three studies used [EMIM] OAc IL at different concentrations. Wang et al. [19] varied the IL concentration from 62% to 80% whereas Bahar et al. [20] and Liu et al. [21] used 80% and 90% concentrated solutions, respectively.

Liu et al.[21] reported a latent heat capacity of 0.58 W. They maintained a constant inlet air condition of 28 °C and 53% RH in their tests while maintaining all other system parameters at a constant value. Bahar et al. [20] reported a maximum latent heat capacity of 4 W. They performed studies at an inlet air temperature of 25 °C and 80% RH and measured the dehumidification performance at three different inlet air flow rates of 2.7, 6.3, and 8.5 CFM. They reported an increase in dehumidification capacity from 0.39 to 4.15 W with increasing the flowrate from 2.7 to 6.3 CFM, but further increase in the air flow rate from 6.3 to 8.5 CFM led to a decrease in dehumidification capacity, from 4.15 to 0.34 W. Wang et al.[19] reported a latent heat capacity of 20 to 132 W.

The impact of air and solution inlet flow rates, along with solution inlet temperature and concentration, was assessed while maintaining a constant dew point temperature of 24.3 °C (equivalent to an inlet air temperature of 32 °C and 63% RH). They varied the air flowrate from 20 to 65 CFM, solution mass flow rate from 0.8 g/s to 3.15 g/s, inlet solution temperature from 21.6 to 30.5 °C, and inlet solution concentration from 62 to 72%.

3.2. Heat Loss Analysis

To provide a comparison of the sensible heat loss from the desiccant solution to the scavenging air in a conventional open system or to the condenser of our semi-open system, a survey of the existing literature on the heat loss associated with the scavenging air was conducted [19,22–27]. While the component design and experimental conditions detailed in these studies [19,22–27] widely vary, in terms of the solution and air flow rates, air inlet conditions, solution concentration, and desiccant, their empirical findings consistently suggest that the sensible heat loss predominantly depends on the desiccant and scavenging air temperatures and water vapor pressures as well as the regenerator heat and mass transfer characteristics. A large difference between the temperature of the two medium results in a high heat loss while a high water vapor pressure difference is associated with a lower heat loss. Notably, Fumo et al. [22] observed sensible heat losses of 28 % and 17 % at 40 °C and 25 °C temperature differences, respectively. Yin et al. [9] reported heat losses of 38 % and 32 % in adiabatic and internally heated regenerators, respectively. This reduction in heat loss is attributed to the elevated water vapor pressure in the desiccant solution induced by internal heating. Kumar et al. [26] reported a drop in heat loss from 51 % to 43 % when the inlet solution and scavenging air temperature difference was reduced from 43.2 °C to 20.6 °C. Yin et al. [27] lowered the sensible heat loss at relatively high temperature differences of 43.5 °C to 54.7 °C by employing a low scavenging air inlet dew point range of 7.3 °C to 16.5 °C. This is due to the fact that increasing the solution and scavenging air water vapor pressure difference leads to an increase in the moisture removal rate per unit mass of the scavenging air. In their tests, Cao et al. [23] document a high heat loss exceeding 60 % at a temperature difference of 56 °C even at a very low dew point of –0.3 °C. Liu et al. [25] report a significant sensible heat loss reaching 50 % even with a low temperature difference of 5.2–8.6 °C and a low average dew point temperature of –0.4 °C because of a relatively low mass transfer coefficient. Wang et al. [19] observed a much higher heat loss relative to those of both Yin et al. [27] and Cao et al.[23] at a similar temperature difference level primarily due to a very limited mass transfer coefficient and a substantial heat transfer area in a hollow fiber membrane regenerator configuration.

Figure 5 shows the comparison of the sensible heat loss as a percentage of the total heat transferred (both sensible and latent) from the desiccant solution to the scavenging air in a conventional open system or to the condenser of our semi-open system, as a function of the average scavenging air dew point or condenser saturation temperatures (Tsat). This temperature represents the lower limit of the water vapor pressure at the scavenging air or condenser and dictates the differential water vapor pressure that drives the water desorption process. A better comparison would have been between the heat loss versus the difference between the water vapor pressures of the source (i.e., regenerator) and the sink (i.e., scavenging air or condenser). Unfortunately, these studies do not provide the desiccant solution concentration which is required to calculate the regenerator vapor pressure.

Comparison of the results at a similar dew point suggests that on average the heat loss of the semi-open regeneration system is only about 13% of the total regenerator heat input while 38% of the regeneration heat is lost in the conventional regenerator studies conducted by Fumo et al. [22], even at a much lower average solution temperature in the regenerator (61.3 °C) relative to that of this study (112.6 °C). It is important to note that a higher regeneration temperature enhances the effectiveness of the moisture removal from the desiccant i.e., making it more concentrated. A more concentrated desiccant improves the system's ability to better dehumidify the air during the adsorption phase and handle higher moisture loads, ultimately improving the overall cycle COP. An optimal regenerator design should maximize the desiccant solution concentration while minimizing its heat loss. The data suggests that the semi-open system achieves this objective at the most important conditions i.e., high dew points, where the heat input to the regenerator is at its peak.

3.3. Effect of Air Moisture Content

Figure 6 shows the system's COP as a function of the inlet air dew point. The desiccant solution flowrate was maintained at 0.0065 kg/s and the air flowrate was 0.0856 kg/s. Figure 5 also shows the heat loss as a percentage of total heat transfer for the absorber and regenerator versus the inlet air dew point. Heat loss in the absorber refers to the fraction of latent heat that returns to the process air, while in the regenerator, it is the

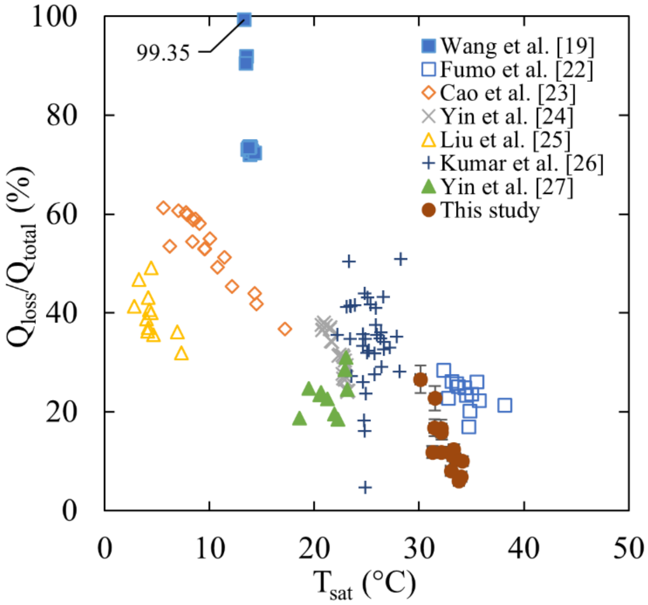

Figure 5. Regenerator sensible heat loss as a percentage of the total heat transfer in relation to the saturation temperature of the scavenging air or condenser. The desiccant solution, process air, and heating oil flowrates along with the inlet temperature of heating oil and condenser cooling water have been maintained at constant values of 311.8 mLPM, 156.7 CFM, 2.78 LPM, 147.6 °C, and 25.2 °C, respectively.[10]

portion of total heat input released to the scavenging air as sensible heat. The COP increases progressively with the dew point, peaking at a dew point of 33.1 °C, as the heat loss ratio substantially drops in the regenerator. The maximum COP corresponds to the highest latent load, when the regenerator heat loss ratio drops to only ~5%. At low dew points, the absorber heat loss is responsible for a greater decline in COP. This can be explained using Figure 6 that shows the CR and the changes in solution concentration (Δx) as a function

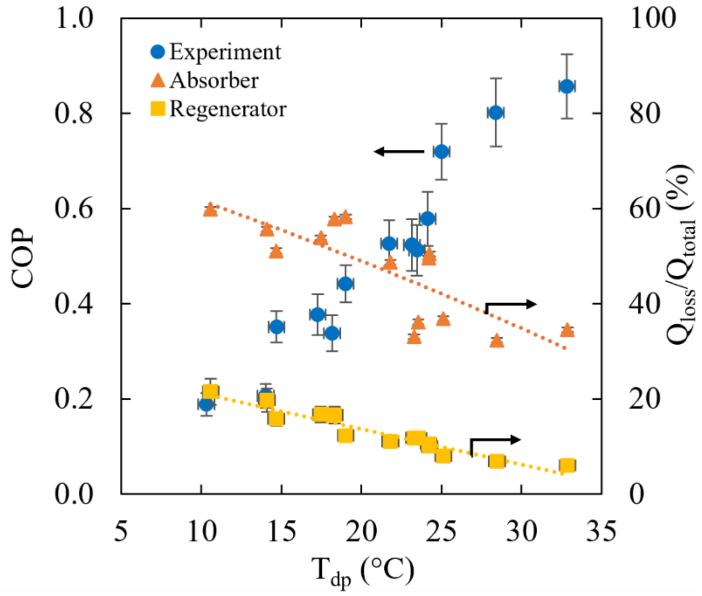

Figure 6. System experimental COP and heat loss as a percentage of total heat transfer for the absorber and regenerator versus the inlet air dew point. The desiccant solution, process air, and heating oil flowrates along with the inlet temperature of heating oil and condenser cooling water have been maintained at constant values of 311.8 mLPM, 156.7 CFM, 2.78 LPM, 147.6 °C, and 25.2 °C, respectively.

of the inlet air dew point. The CR, along with Δx, reflects the effectiveness of the moisture absorption process as the desiccant cycles through the absorber. As the water vapor potential driving the moisture to the desiccant declines, the amount of moisture absorbed to the desiccant relative to the desiccant flow (i.e., Δx) decreases, leading to a high CR. A high CR means excessive cycling of the desiccant solution through the absorber, and heat loss to the process air, without picking up a substantial amount of latent load. In essence, at extremely high CR the system operates as a heater, rapidly shuttling the regenerator heat to the process air. Ideally, an absorber would remove moisture from the process air without returning sensible heat to the air. The dehumidification COP of the system was also calculated analytically using Eq. 3 and is plotted in Figure 7.

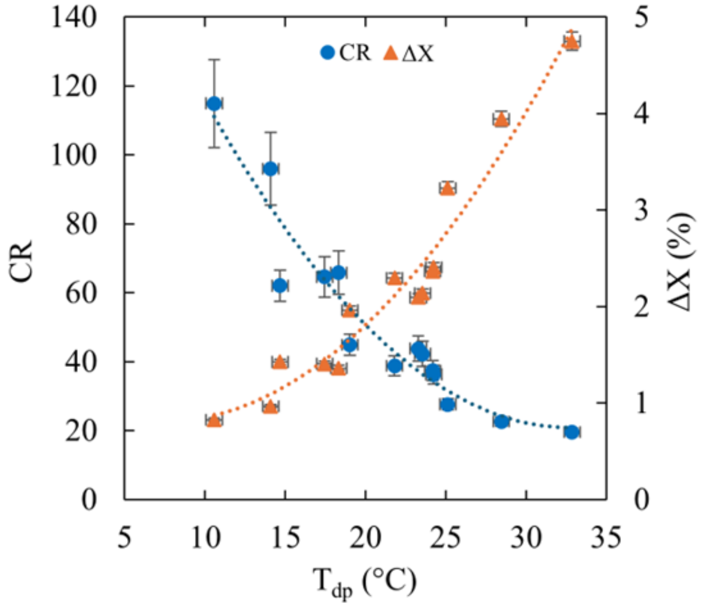

Figure 7. Circulation ratio (CR) and change in solution concentration versus the inlet air dew point. The desiccant solution, process air, and heating oil flowrates along with the inlet temperature of heating oil and condenser cooling water have been maintained at constant values of 311.8 mLPM, 156.7 CFM, 2.78 LPM, 147.6 °C, and 25.2 °C, respectively.[10]

At low dew points, the higher CR leads to a higher sensible heat loss in the absorber and a lower COP. Therefore, it is anticipated that reducing the solution flowrate leads to a reduction in heat loss and an improvement in COP.

4. Conclusions

This study highlights the potential of the semi-open absorption cycle for efficient dehumidification, particularly in applications where high latent cooling is required. The elimination of scavenging air through the integration of a regenerator-condenser module not only addresses a key limitation of conventional liquid desiccant systems but also significantly reduces heat losses associated with the regeneration process, which have been a major contributor to inefficiency in traditional designs. Our experimental results indicate that the semi-open cycle achieves superior performance, with heat losses as low as 5% of the total heat input. This reduction in heat loss, coupled with the ability to concentrate the desiccant more effectively, leads to a marked improvement in the system's coefficient of performance (COP), particularly at higher dew points where the dehumidification demand is highest. Future research could explore increasing the condenser temperature at partial load conditions to significantly reduce regeneration heat loss in proportion to the reduction in latent load.

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