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Energy performance of an innovative liquid desiccant dehumidification system with a counter-flow heat and mass exchanger using potassium formate

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Abstract

An innovative micro-scale liquid desiccant dehumidification system is numerically investigated. The liquid desiccant dehumidification unit employs a counter-flow low-cost and efficient heat and mass exchange core, improving the thermal performance and eliminating desiccant carryover with the process air. An environmentally friendly, non-corrosive, nontoxic and chemically stable HCOOK potassium formate liquid desiccant solution was employed in the unit. A set of governing differential equations was established for the dehumidification system operation allowing the development of a numerical model to predict and simulate the energy performance and various output parameters of the dehumidifier. A numerical case study was considered for a micro-liquid desiccant dehumidification system using potassium formate and a dehumidification and cooling capacity of around 11.32 kW was attained with about 59% humidity effectiveness and 62% enthalpy effectiveness. In addition, a parametric study was performed to investigate the effect of various operational parameters on the overall performance of the liquid desiccant dehumidifier. Utilizing the developed numerical model, it was shown that the dehumidifier effectiveness is directly proportional to the intake air temperature, intake air relative humidity and liquid desiccant flow rate where the effectiveness is inversely proportional to the intake air velocity and the heat exchanger air channel height.

Keywords: Dehumidification, liquid desiccant, counter-flow, potassium formate, heat and mass exchange, effectiveness

Introduction

Desiccant dehumidification is a technology that has been in use long time ago for both industrial and agricultural purposes including humidity control in textile mill and post-harvest crop drying in stores. In the recent years, a large body of research has been concentrated on desiccant materials and their applications in various fields especially dehumidification and cooling applications [1-5]. This is based on the reliability and simplicity provided by desiccant dehumidification and cooling systems and the desiccant materials ability to be regenerated with low temperature heat, solar energy, biomass heat, or any waste heat source. Such systems do not use ozone depleting fluids and can independently control temperature and humidity in a conditioned space. The capability of a desiccant material to absorb moisture is governed by its equilibrium vapour pressure and depends mainly on the difference in the vapour pressure between the process air and the surface of the desiccant material employed. As long as the vapour pressure of the desiccant surface is less than that of the process air, moisture absorption occurs and air dehumidification is attained until the equilibrium between air and desiccant is reached [6]. It is reported that desiccant-based dehumidification units use less than a quarter of the energy compared to traditional vapour-compression systems [7]. Desiccant materials are generally divided into two main categories, solid desiccants and liquid desiccants. Widely used solid desiccants include silica gel,

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natural and synthetic zeolites, polymers, activated alumina, hydratable salts and titanium silicate. Such desiccant materials are impregnated into a honeycomb wheel matrix, or called enthalpy plate/wheel exchanger, which is alternatively cooled and heated to allow water vapour sorption and desorption [8]. On the other hand, liquid desiccant based dehumidification is a relatively new technology compared to solid desiccant systems employing a deliquescent material to absorbmoisture from air, including lithium chloride (LiCl) and calcium chloride ($CaCl_2$) [9-12]. Compared to solid desiccants, liquid desiccant-based systems have higher utilization flexibility and mobility, lower regeneration temperature in addition to the ease of manipulation and lower pressure drop on the air side [9,13]. However, a common problem for conventional liquid desiccant dehumidification systems is the entrainment of the desiccant solution in the absorbers by the process air where the liquid desiccant flows over an extended surface or packing material [7]. This will have negative impacts on the indoor air quality and the comfort of the occupants. The liquid desiccant carryover can be reduced or eliminated by using a semi-permeable micro-porous membrane-based contactor instead of the conventional direct liquid desiccant-air contact core [14].

Liquid desiccant dehumidification system

A liquid desiccant dehumidification system consists of two major sub-units, a dehumidifier and a regenerator. In the dehumidifier, strong liquid desiccant absorbs moisture to dehumidify the hot and humid process air due to the difference in the water vapour pressure between the air and the desiccant surface. Therefore, the liquid desiccant concentration decreases and its moisture content increases to become saturated, where the air is dehumidified and its moisture content decreases. To allow a continuous operation of the liquid desiccant dehumidification unit, the weak liquid desiccant is transferred to the regenerator where it is re-concentrated, reducing its moisture content by means of an external heating source. A typical liquid desiccant dehumidification system is presented in **Figure 1** along with the system operation stages [15].

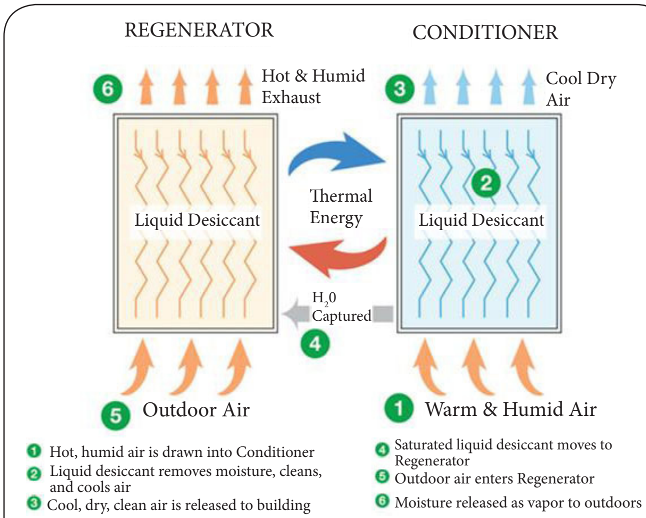

Figure 1. Liquid desiccant dehumidification system operation.

Compared to traditional vapour compression systems, liquid desiccant-based dehumidification and cooling systems have the following advantages [16-18]:

1. Using low-grade waste heat or solar energy instead of conventional boilers for desiccant regeneration at low temperature and thus increasing the system overall efficiency and allowing primary energy savings.
2. High reliability and flexibility of operation with the possibility of liquid pumping and transfer between units including dehumidifiers and regenerators.
3. Less air pressure drop on the process air side.
4. Environmentally friendly operation, limiting the use of HFC or HCFC refrigerants utilized and less greenhouse gas emissions.
5. Efficient humidity control especially in areas where healthy indoor air is a critical factor.
6. Cost effective with lower initial and operational costs since no or limited compression devices are employed.
7. The ability for thermal energy storage in the form of strong liquid desiccant especially in the intermittent operation cases.
8. Higher indoor air quality with the ability to filter moulds, microbial contamination, bacteria and viruses from the process air.

On the other hand, the disadvantages of using liquid desiccant-based dehumidification systems include the complex design, the corrosive nature of conventional liquid desiccant solutions and the issue of desiccant droplets entrainment by the process air which could affect the indoor air quality and could have negative impacts on the occupants' safety and health [2,6]. These issues concerning conventional liquid desiccant dehumidification systems could be overcome with proper system design and through using environmentally friendly liquid desiccant solution with attractive physical and thermodynamic properties. In this study, a numerical investigation is carried out for an innovative micro-scale liquid desiccant dehumidification system with a counter flow compact and efficient heat and mass exchanger core. A chemically stable potassium formate HCOOK liquid desiccant [19] is utilized offering physical and thermodynamic advantages compared to conventional desiccants, being less corrosive, environmentally friendly and cheaper to produce. In addition, a parametric study is provided to study the effect of different operational parameters on the system overall performance using the environmentally friendly potassium formate as a desiccant solution.

Methods

Figure 2 shows the structure of the desiccant dehumidifier core, consisting of a liquid desiccant tank, pump and cellulose

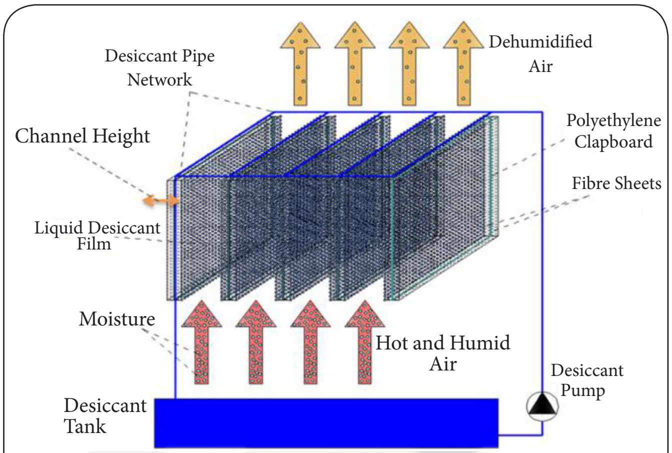

Figure 2. Schematic of the liquid desiccant dehumidifier core.

fibre membranes. Each fibre membrane is made of two fibre sheets attached to very thin polyethylene clapboard, and two adjacent fibre membranes form one air channel. The liquid desiccant is pumped from the solution tank at the bottom of the dehumidifier through the pipe network to be sprayed evenly at the top of the dehumidifier core. The vertical spraying of the liquid desiccant facilitates the design and configuration of the desiccant core as the desiccant solution flows downward by gravity.

The strong desiccant flows down along the fibre sheets and is soaked and held firmly by the cellulose long fibres having favourable hydrophilic properties and strong water absorptivity. Thus, humid air flows in an upward direction in a counter flow mode to get in contact with the liquid desiccant film on the fibre sheets without carrying out the desiccant droplets. As the humid air gets in direct contact with the liquid desiccant film, the water vapour pressure difference between the surface of the desiccant film and the process air allows moisture absorption by the strong liquid desiccant solution. In addition, the relatively lower temperature of the liquid desiccant film allows sensible cooling of the process air and both the sensible heat and the latent heat of condensation are transferred from the air side to the relatively cold desiccant solution. In this section, the mathematical formulation and the equations governing the liquid desiccant dehumidifier are presented. The regenerator unit is assumed to have a similar core as the dehumidifier one but with an inverse operation principle. The regeneration core allows reducing the moisture content of the liquid desiccant through heating the solution and transferring the moisture to the regenerator process air.

In the development of the dehumidification unit model, the following assumptions were considered:

1. The mass and heat transfer process in the counter-flow exchanger is considered in steady state.
2. The dehumidification process is adiabatic and the heat losses to the surrounding are negligible.
3. The desiccant solution is evenly distributed on the fibre sheets surfaces with complete saturation.
4. The resistance of the desiccant solution is ignored because of the very thin film thickness.
5. Air physical and thermal properties in addition to the desiccant solution properties are considered to be uniform within a single control volume.

The assumptions above were adopted in previous published numerical and analytical studies and were found to have insignificant effect on the overall dehumidification unit performance [20,21]. By applying heat and mass conservation on the air and liquid desiccant sides, a set of governing differential equations is established leading to the development of a one-dimensional model to predict the dehumidifier thermal performance and various output parameters. In order to solve for the governing heat and mass equations, a differential control volume is considered with its input and output parameters as shown in Figure 3 with a counter flow mode where the liquid desiccant is sprayed down and the process air is supplied upwards.

The energy conservation equation for the process air in the considered differential control volume is given by:

$$\dot{m}_a \partial H_a = h_a L_{ch} (T_a - T_s) \partial x + H_{wv} \dot{m}_a \partial w_a \quad (1)$$

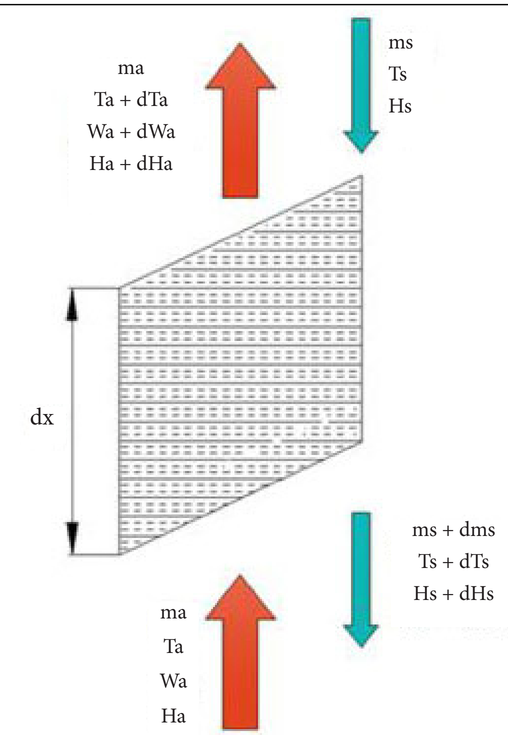

Figure 3. Numerical analysis differential control volume.

T_a and T_s are air and liquid desiccant solution temperatures in °C, h_a is the air convective heat transfer coefficient in the wet channel in W/m^2.°C, w_a is the air humidity ratio in kg_{H2O}/kg_{air}, L_{ch}

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is the length of the control volume in m, H_a is the moist process air specific enthalpy in J/kg and H_{wv} is the specific enthalpy of water vapour at the liquid desiccant film temperature in J/kg.

The mass balance equation on the liquid desiccant side and the process air side of the differential control volume is given by Equations (2) and (3):

$$\partial \dot{m}_s = \dot{m}_a \partial w_a \tag{2}$$

$$\dot{m}_a \partial w_a = h_{md} L_{ch} (w_{eq} - w_a) \partial x \tag{3}$$

$\dot{m}_a$ and $\dot{m}_s$ are the mass flow rate of the process air and the liquid desiccant in kg/s, h_{md} is mass transfer coefficient based on the air humidity ratio difference in kg/m^2.s and w_{eq} is the air humidity ratio in equilibrium with the liquid desiccant solution at the interfacial area in kg_{H2O}/kg_{air}. The overall energy balance for the differential control volume taking into account the liquid desiccant falling film and the process air flowing in a counter flow mode is given by:

$$\dot{m}_s cp_s \partial T_s + cp_s T_s \partial \dot{m}_s = \dot{m}_a \partial H_a \tag{4}$$

cp_s is the liquid desiccant film specific heat capacity in J/kg.°C. Regression functions for the potassium formate liquid desiccant thermo-physical properties, in terms of fluid temperature T_s and concentration X_s, were developed and implemented in the numerical model using SPSS statistics software (SPSS 2009, v. 18.0) based on thermo-physical properties provided in the literature [22,23].

The convective mass transfer coefficient between the process air and the falling liquid desiccant is given in terms of the convective heat transfer coefficient and the Lewis number by:

$$h_{md} = \frac{h_a}{\rho_a cp_a Le^{2/3}} \tag{5}$$

cp_a and ρ_a are the respective air specific heat capacity in J/kg.°C and density in kg/m^3. The variation in the liquid desiccant concentration at the outlet of the dehumidifier is given by the following equation:

$$X_{s,out} = \left(\frac{1}{1 + \frac{\dot{m}_a}{\dot{m}_s}(w_{a,in} - w_{a.out})} \right) X_{s,in} \tag{6}$$

$X_{s,in}$ and $X_{s,out}$ are the liquid desiccant concentration at the inlet and outlet of the dehumidifier respectively. The cooling capacity of the liquid desiccant dehumidification system can be represented as:

$$Q_{cooling} = \dot{m}_a (H_{a,in} - H_{a,out}) \tag{7}$$

$H_{a,in}$ and $H_{a,out}$ are the respective specific enthalpy of the intake air and the process air in J/kg. The performance of the liquid desiccant dehumidification system is evaluated by two main parameters, the humidity effectiveness and the enthalpy effectiveness. The humidity effectiveness, or called moisture effectiveness, is defined as the ratio of the actual change of air humidity ratio to the maximum possible change and can be calculated for both the dehumidifier and regenerator units by:

$$\varepsilon_{w,Deh/\mathrm{Reg}} = \frac{w_{a,in} - w_{a,out}}{w_{a,in} - w_{eq}} \tag{8}$$

In the same manner, the enthalpy effectiveness is defined as the ratio of the actual air enthalpy change to the maximum change at ideal conditions and can be given by:

$$\varepsilon_{h,Deh/\mathrm{Reg}} = \frac{H_{a,in} - H_{a,out}}{H_{a,in} - H_{eq}} \tag{9}$$

H_{eq} in J/kg, is the air enthalpy in equilibrium with the liquid desiccant solution and can be estimated using the air humidity ratio w_{eq} with an air temperature equal to that of the liquid desiccant solution.

Results

A set of governing differential equations was established based on the heat and mass transfer balances presented in the previous section for the dehumidification unit with a counter flow core. A one-dimensional computational numerical code was developed in Matlab R2011 employing a second order fully implicit accurate finite difference scheme to predict the thermal performance of the dehumidification unit including the temperature and humidity distribution of the process air and the liquid desiccant sprayed across the surfaces of the channels in addition to various system output parameters. The one-dimensional discretization scheme utilized is based on uniform grid spacing along the fibre sheet in the process air flow direction within the heat and mass exchanger. At each node, the governing set of coupled heat and mass balance equations were solved using the Newton Raphson method, and the iterative solution is considered to be converged when the maximum of the residual is less than 10^{-6} for temperature and humidity ratio at each node. It was shown that a maximum difference of 0.0463°C in the supply air temperature is attained as the grid is increased from 20 nodes to 50 nodes corresponding to a maximum temperature change of 0.23%. Thus, a 20 nodes grid was considered to be accurate enough to predict different system parameters with an acceptable computational memory and time.

The numerical model was employed to investigate the thermal performance of the desiccant dehumidification unit with a counter-flow heat and mass exchange configuration, predicting the temperature, humidity and the specific enthalpy distribution along the air channel in addition to the system dehumidification capacity and the respective humidity and enthalpy effectiveness. Multiple numerical simulations were carried out and the thermal performance of an optimized dehumidification system design is presented in this section. The geometrical specifications of the air/liquid desiccant counter-flow heat and mass exchanger in addition to the operational conditions for the considered case study are presented in Table 1. The fibre sheets employed in the core are 500×500 mm area with a 5 mm air channel height between

Table 1. Dehumidification system geometrical specifications and operational conditions.

Parameter	Value
Fibre sheet length (mm)	500
Fibre sheet width (mm)	500
Air channel height (mm)	5
Wall thickness (mm)	0.5
Intake air flow rate (m^3/h)	1000
Intake air velocity (m/s)	2
Intake air temperature (°C)	30
Intake air relative humidity (%)	85
Intake air humidity ratio (kg_{H2O}/kg_{air})	0.023
Desiccant solution	HCOOK
Supply desiccant temperature (°C)	20
Supply desiccant mass concentration (%)	74
Supply desiccant flow rate/channel (l/s)	0.0067

two fibre membranes and 0.5 mm as sheet wall thickness between two adjacent channels. The velocity of the intake air was set to 2 m/s with a 1000 m^3/h as volumetric flow rate. The liquid desiccant is sprayed at the top of the exchanger in a vertical downward position at 20°C and 0.0067 l/s per channel. Simulations were carried out employing 20 nodes based on the finite difference discretisation **Figure 4** presents the drop in the process air temperature and specific enthalpy along the air flow direction in the air channel. Employing a liquid desiccant temperature of 20°C, the working air temperature decreases from 30°C to 24.3°C with a drop of around 5.7°C. This temperature drop is mainly due to the temperature difference between the relatively warmer air and the colder desiccant allowing air cooling as it gets in contact with the liquid desiccant film along the fibre sheets. In addition, the specific enthalpy of the process air drops through the channel from about 88.7 kJ/kg to 54.6 kJ/kg.

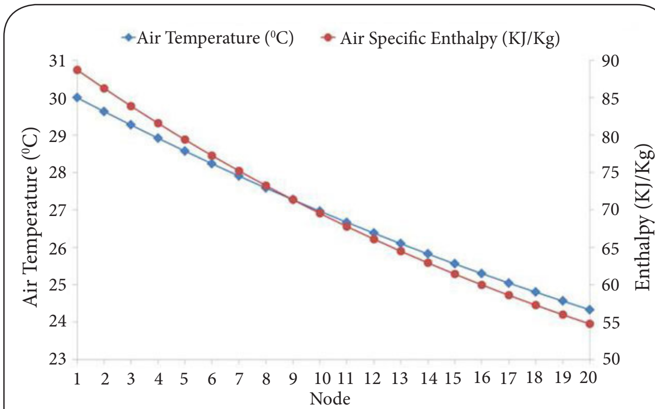

Figure 4. Air temperature and specific enthalpy variation in the dehumidifier.

In addition, **Figure 5** shows the variation in the process air moisture content and the corresponding relative humidity in the air flow direction in the channel. As the working air flows vertically upward, it gets in contact with the strong liquid desiccant sprayed along the fibre sheets. Due to the vapour pressure difference between the process air and the liquid desiccant film, water vapour is transferred from the air to the desiccant reducing the moisture content of air. As shown in **Figure 5**, process air humidity ratio drops from 0.023 kg_{H2O}/kg_{air} to about 0.0119 kg_{H2O}/kg_{air} with a corresponding drop in the air relative humidity from about 85% to about 62.7%. The variation in the liquid desiccant temperature in the direction of air flow is shown in **Figure 6**. In the dehumidification channel, both the sensible heat and the latent heat of condensation are transferred from the process air side to the relatively colder desiccant and thus increasing its temperature. The liquid desiccant temperature increases by 3°C from an initial temperature of 20°C to about 22.9°C. Due to the water vapour absorption by the liquid desiccant, the desiccant mass concentration has dropped from 74% to about 73.49%. In addition, the overall system dehumidification and cooling capacity is around 11.32 kW with 59.35% humidity effectiveness and about 62.18% enthalpy effectiveness.

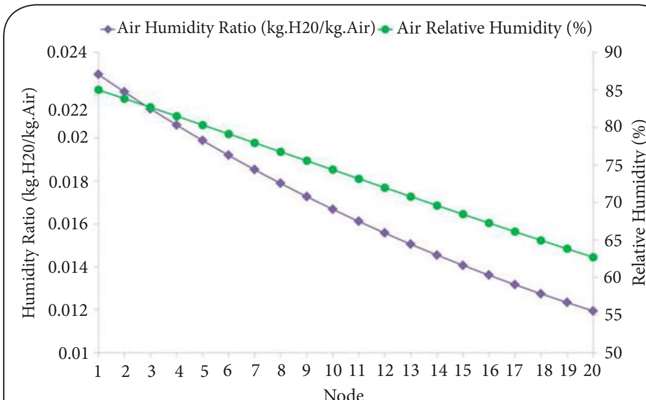

Figure 5. Air humidity ratio and relative humidity variation in the dehumidifier.

After dehumidifying the humid and hot air, the strong desiccant moisture content increases and its concentration decreases form 74% to 73.49%. Thus, to allow the continuous operation of the liquid desiccant dehumidification unit, the moisture content of the desiccant should be reduced through proper regeneration of the weak desiccant in the regenerator core of the unit. In this case study, the regenerator was assumed to have the same counter-flow heat and mass exchanger core as the dehumidifier described above. Therefore the operation of the regenerator unit was simulated in order to allow the continuous operation of the dehumidification system. It was shown that the temperature of the liquid desiccant exit from the dehumidifier, at about 23°C, should be raised to about 63°C to allow the liquid desiccant regeneration and restore the

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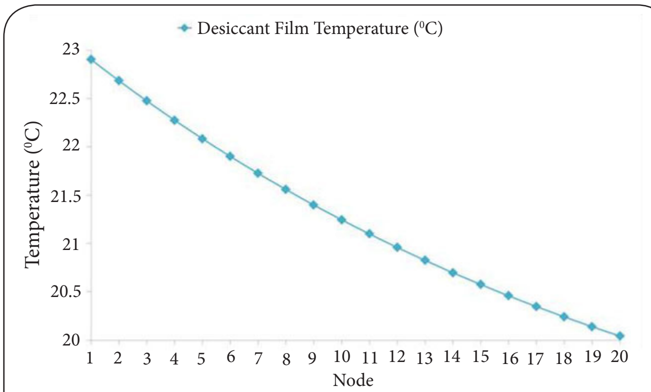

Figure 6. Liquid desiccant temperature variation in the dehumidifier.

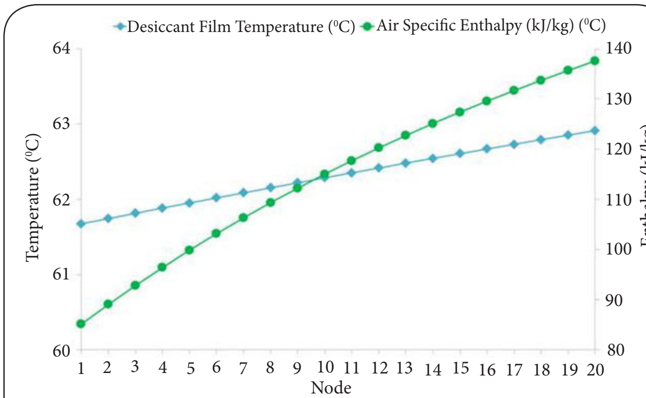

Figure 8. Air specific enthalpy and liquid desiccant temperature variation in the regenerator.

74% mass concentration of the strong desiccant. This could be attained using a low-grade waste heat source or other alternative renewable heating resource as solar energy or biomass heat. Employing the same operational and geometrical parameters presented in **Table 1**, **Figure 7** shows the variation in the air temperature and humidity ratio in the direction of air flow throughout the regeneration process. The process air temperature exhibits an increase from 30°C to about 49°C as it gets in contact with the relatively warmer liquid desiccant at 63°C. In addition, mass transfer of water vapour from the weak liquid desiccant to the process regeneration air allows increasing the air humidity content from 0.021 kg_{H2O}/kg_{air} to about 0.033 kg_{H2O}/kg_{air}.

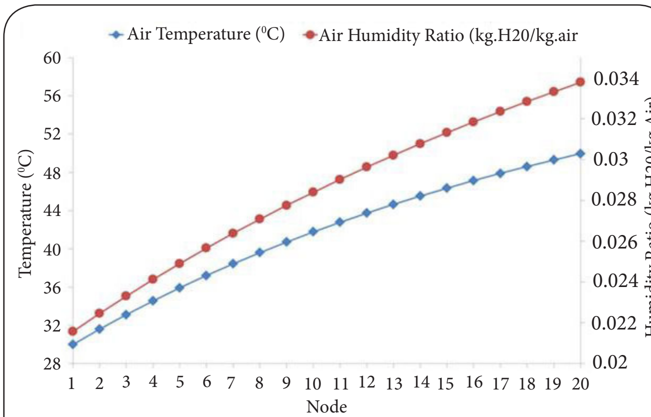

Figure 7. Air temperature and humidity ratio variation in the regenerator.

Figure 8 presents the variation in the liquid desiccant temperature and the overall air specific enthalpy in the direction of process air flow during the regeneration process. With an increase in the dry bulb temperature and humidity ratio, the process air specific enthalpy increases by about 52 kJ/kg accompanied by a decrease in the liquid desiccant temperature. In addition, the regenerator unit humidity effectiveness and enthalpy effectiveness are 60.08% and 61.11% respectively.

In this case study, an extreme case study has been chosen with a high air relative humidity to demonstrate the capability of the liquid desiccant dehumidification unit to reduce the air humidity content. Based on the numerical study results provided, it should be noted that the attained exit air conditions with 24°C would be suitable for space air ventilation and improving the indoor air quality, however the dehumidification system should be coupled with a cooling system to provide the additional cooling capacity required by reducing the supply air temperature further to less than 15°C including electric and absorption chillers or dew point cooling systems. Regarding the dehumidification capability of the proposed system, the obtained humidity ratio drop of about 0.011 kg_{H2O}/kg_{air}, or even a less value, is suitable for air conditioning purpose.

Discussion

Utilizing the developed numerical simulation model, a parametric study was carried out for a liquid desiccant dehumidifier with a counter-flow heat and mass exchanger core. Multiple simulations were performed to investigate the effect of different operational parameters on the overall performance of the dehumidifier including intake air temperature and relative humidity, intake air velocity, liquid desiccant mass flow rate and air channel height. The initial set operational conditions for the parametric study are 30°C intake air temperature, 80% air relative humidity, 2 m/s air velocity, 20°C liquid desiccant temperature, 0.01 kg/s desiccant flow rate, 0.5 m channel length and 5 mm channel height.

Intake air relative humidity effect

Figure 9 shows the effect of the intake air relative humidity on the overall performance of the counter-flow liquid desiccant dehumidifier. As shown in **Figure 9a**, increasing intake air relative humidity leads to an increase in the supply air relative humidity in addition to a slight increase in the supply air temperature. **Figure 9b** shows that the exit desiccant temperature is directly proportional to the intake air relative

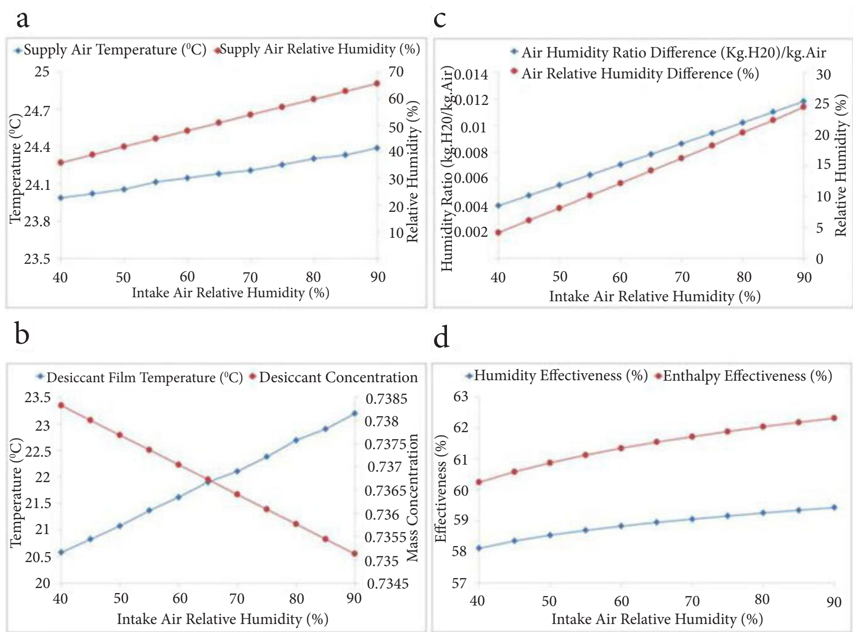

Figure 9. Intake air relative humidity effect on the dehumidifier overall performance.

humidity where the exit desiccant concentration is inversely proportional to the relative humidity of air inlet to the dehumidifier. In addition, the drop in both air humidity ratio and the corresponding air relative humidity is proportional to the increase in the inlet air relative humidity as shown in **Figures 9c** and **9d** shows that as the intake air relative humidity increases, the humidity and enthalpy effectiveness of the dehumidifier increase. This is mainly because increasing the air relative humidity will result into an increase in the air partial vapour pressure, and thus providing larger vapour pressure difference between the process air and the liquid desiccant surface. This increase in the vapour pressure difference is accompanied by an increase in the desiccant moisture absorption capacity and dehumidifier effectiveness.

Intake air temperature effect

Figure 10a shows that the supply air temperature and relative humidity are proportional to the intake air temperature. Due to the contact between the process air employed and the liquid desiccant on the channels surfaces, the desiccant film temperature increases where the exit solution concentration decreases as the intake air temperature increases as shown in **Figure 10b**. In addition, **Figure 10c** shows that the drop in the relative humidity across the dehumidifier is inversely proportional to the intake air temperature where the drop in the air humidity ratio across the dehumidifier increases as the intake air temperature increases. As presented in **Figure 10d**, there is a slight increase in the dehumidifier humidity and enthalpy effectiveness as the intake air temperature increases from 25°C to 40°C. Based on this investigation, it is shown that the dehumidification unit performance is more dominated by the intake air relative humidity than the air temperature.

Intake air velocity effect

Figure 11 presents the effect of the intake air velocity on the overall performance of the counter-flow liquid desiccant dehumidifier. As shown in **Figure 11 a**, increasing the velocity of the intake air leads to an increase in both the supply air relative humidity and temperature. Increasing the air velocity and keeping the same geometrical specifications of the exchanger results in an increase in the air mass flow rate. With an intake air temperature of 30°C and desiccant temperature of 20°C, larger air mass flow rate allows increasing the temperature of the liquid desiccant as shown in **Figures 11b** and **11c** shows a decrease in both the air humidity ratio and relative humidity drop across the dehumidifier as the intake air velocity increases. Moreover, the overall dehumidifier humidity and enthalpy effectiveness exhibit a drop as the air velocity is increased from 0.5 m/s to 5 m/s as presented in **Figure 11d**. This is basically due to the larger amount of air introduced to the dehumidification unit and thus the capability of the dehumidifier to reduce the humidity of the air decreases.

Desiccant flow rate effect

Figure 12 shows the effect of the liquid desiccant flow rate on the dehumidifier performance. It is shown that both the supply air temperature and relative humidity decrease as the desiccant flow rate increases as shown in **Figure 12a**. In addition, the increase in the desiccant temperature at the dehumidifier outlet is less significant for higher desiccant mass flow rates as shown in **Figure 12b**. Moreover, the drop in the air humidity ratio and relative humidity across the dehu-

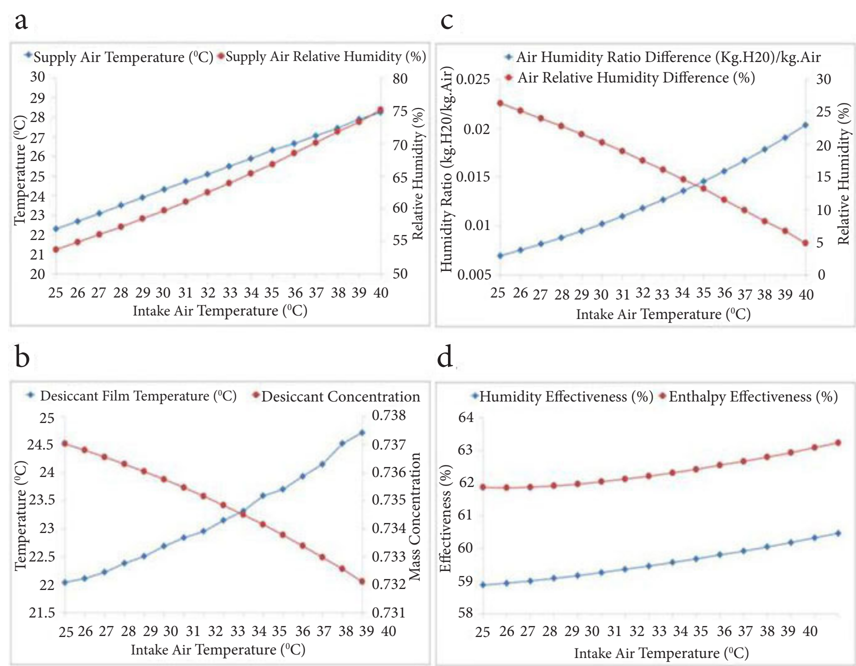

Figure 10. Intake air temperature effect on the dehumidifier overall performance.

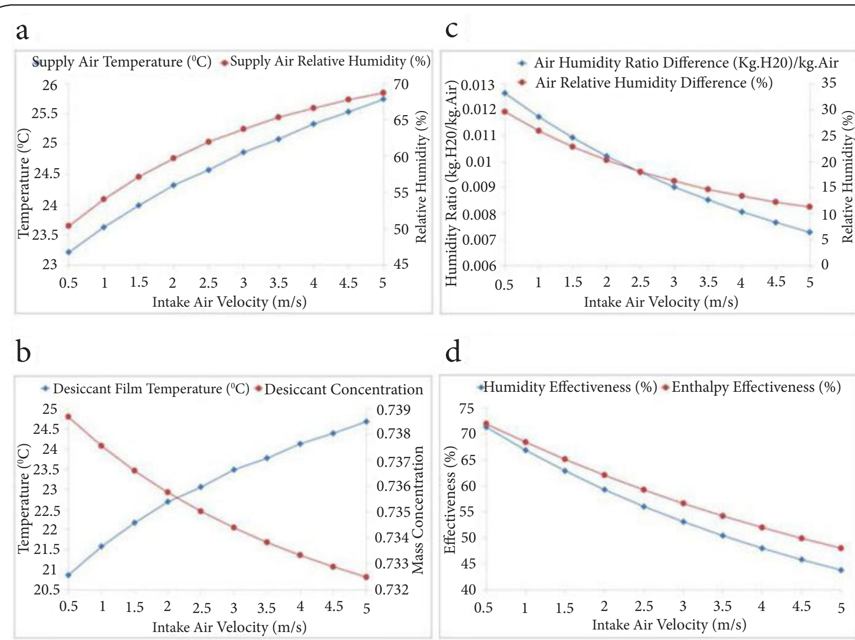

Figure 11. Intake air velocity effect on the dehumidifier overall performance.

midifier increases as the desiccant flow rate increases. This increase is accompanied by an increase in the humidity and enthalpy effectiveness as shown in **Figures 12c** and **12d**.

Channel height effect

Figure 13 shows the effect of the exchanger channel height on different dehumidifier operational parameters. A range of channel height between 3 and 15 mm was considered and simulations were carried out to investigate the effect of increasing the height of the channel on the dehumidifier performance. As shown in **Figure 13a**, both the supply air temperature and relative humidity are inversely proportional to the process air channel height. **Figure 13b** shows that increasing the channel height is accompanied by an increase

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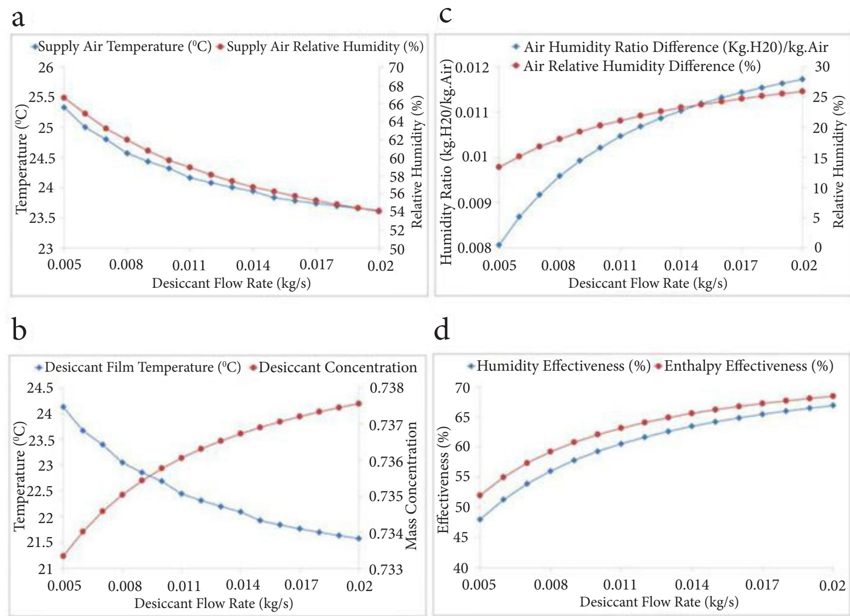

Figure 12. Liquid desiccant flow rate effect on the dehumidifier overall performance.

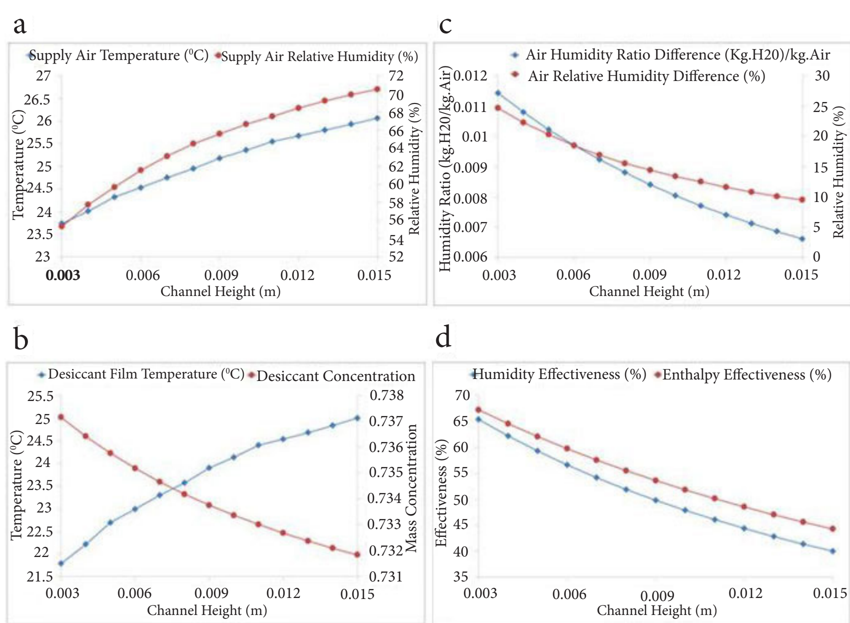

Figure 13. Process air channel height effect on the dehumidifier overall performance.

in the liquid desiccant film temperature where the capability of the unit to reduce the air humidity ratio decreases with the increase of the channel height. In addition, the dehumidifier unit humidity and enthalpy effectiveness are inversely proportional to the channel height as shown in **Figure 13d**. Therefore, it is shown that the heat and mass transfer efficiency decreases when the air channel height increases, **Figure 13c**. Also it is shown that for large channel heights, the heat and mass transfer in the channels is poor resulting into higher air supply temperatures and lower effectiveness.

Based on the similar trends observed in **Figures 9** and **12** and in both **Figures 11** and **13**, it was shown that the dehumidifier effectiveness and the humidity ratio drop in the dehumidifier increase with the increase in the intake air temperature, intake air relative humidity and liquid desiccant flow rate where the effectiveness is inversely proportional to the intake air velocity and the heat exchanger air channel height.

Conclusions

High humidity levels have different negative impacts on the thermal comfort and the indoor air quality in occupied spaces including mould growth, bad smells and odours and evaporation restriction from the body. The conventional way of latent heat removal is through water vapour condensation by reducing air temperature below the dew point temperature using vapour compression devices and cooling coils. However, these conventional techniques are usually associated with high electrical energy consumption with major financial and environmental drawbacks. Among the alternative solutions, desiccant dehumidification is one of the attractive technologies allowing significant savings on the primary energy consumption with an efficient and environmentally friendly operation mode. Compared to solid desiccant units, liquid desiccant systems allow higher flexibility and mobility in the utilization with lower regeneration temperature and higher coefficient of performance.

In this work, an innovative micro-scale liquid desiccant dehum-idification unit is numerically investigated. The liquid desiccant dehumidification unit employs a counter-flow compact and efficient heat and mass exchanger core enhancing the thermal performance and eliminating desiccant carryover with the process air. In addition, a chemically stable, non-corrosive, nontoxic and environmentally friendly HCOOK potassium formate liquid desiccant solution was employed in the unit. A set of governing differential equations are established for the dehumidification system operation allowing the development of a one-dimensional model to predict and simulate the dehumidifier thermal performance and various output parameters. A numerical case study was considered for a micro-liquid desiccant dehumidification system using potassium formate and a dehumidification and cooling capacity of around 11.32 kW was attained with 59% humidity effectiveness. Based on the results attained, it was shown that the liquid desiccant dehumificiation unit has the capacbility to provide fresh air for space air ventilation and improving the indoor air quality, however a cooling system is needed to be integrated with the dehumidification unit to provide the additional cooling capacity required and reduce further the supply air temperature, including electric and absorption chillers or dew point cooling systems. In addition, a parametric study was performed to investigate the effect of various operational parameters on the liquid desiccant dehumidifier performance. It was reported that the dehumidifier effectiveness and the humidity ratio drop in the dehumidifier are directly proportional to the intake air temperature, intake air relative humidity and liquid desiccant flow rate where the effectiveness is inversely proportional to the intake air velocity and the heat exchanger air channel height.

List of abbreviations

$CaCl_2$: Calcium Chloride
cp: specific heat capacity at constant pressure (J/kg.°C)
h: heat convection coefficient (W/m^2.°C)
hmd: mass transfer coefficient (m/s)
H: specific enthalpy (J/kg)
Heq: specific air enthalpy in equilibrium with the liquid desiccant solution (J/kg)
Hwv: specific enthalpy of the water vapour at the water film temperature (J/kg)
HCFC: hydrochlorofluorocarbon
HCOOK: Potassium Formate
HFC: hydrofluorocarbon
L: length (m)
Le: Lewis number
LiCl: Lithium Chloride
m: mass flow rate (kg/s)
Q: cooling capacity (W)
T: Temperature (°C)
w: humidity ratio (kg_{H2O}/kg_{air})
weq: humidity ratio in equilibrium with liquid desiccant solution (kg_{H2O}/kg_{air})
X: solution concentration

Greek

ε: Effectiveness (%)
ρ: Density (kg/m^3)

Subscripts

a: Air
ch: Channel
Deh: Dehumidification
in: input
Md: Moisture dehumidification
Out: Output
Reg: Regenerator
S: Solution

Competing interests

The authors declare that they have no competing interests.

Authors' contributions

Authors' contributions	MJ	SR
Research concept and design	✓	✓
Collection and/or assembly of data	✓	--
Data analysis and interpretation	✓	--
Writing the article	✓	--
Critical revision of the article	✓	✓
Final approval of article	✓	✓
Statistical analysis	✓	--

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