Numerical Investigation of Influence of Extent and Position of a Bypass on the Performance of a Single Unit Liquid Desiccant Based Evaporative Cooler

Raja Naveen Pamu[1, 2], P. S. Kishore[1], Siva Subramanyam Mendu[3]*

[1]Department of Mechanical Engineering, Andhra University College of Engineering, Andhra University,Visakhapatnam, INDIA 530003
[2]Department of Mechanical Engineering, Raghu Engineering College (A), Visakhapatnam, INDIA 531162
[3]Department of Mechanical Engineering, MVGR College of Engineering (A), Vizianagaram, Andhra Pradesh, INDIA 535005
[*]rjnvn15@gmail.com

ABSTRACT

Air-conditioning of buildings is more energy-consuming with widely used vapour compression refrigeration systems in hot and humid areas. Alternatively, evaporative cooling does not show much effect in sultry regions because of increased humidity despite the minimum energy consumption. Indirect evaporative cooling is a promising alternative to direct evaporative cooling systems, but it suffers from low cooling effectiveness and is unsuitable for highly humid regions. Considering the above facts an alternative liquid desiccant-assisted modified M-cycle cooling system is proposed and examined numerically in the present work. The proposed cooling and dehumidification system is a simple unit incorporating chemical dehumidification and regenerative evaporative cooling in a single unit. A simultaneous heat and mass transfer model is developed for the numerical analysis. The numerical code is written in MATLAB and is validated with results available in the literature. Further, numerical analysis was carried out for the effect of various parameters on the performance of the proposed cooling system. The study shows an improvement of 19.2% in dew point effectiveness by incorporating an intermediate opening in the plate separating the product air and working air channels.

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Introduction

Due to rapid urbanization and improved living standards of people, there is an increased demand for human comfort and air-conditioning has become a necessity rather than a luxury. Commonly used Vapor Compression Refrigeration (VCR) systems consume a large quantity of power. These systems suffer from poor ventilation as they rely upon partial recirculation of indoor air. Because of certain drawbacks with conventional VCR systems such as reducing the air temperatures below Dew Point Temperature (DPT) for dehumidification, adding to high peak loads, complete reliance on electrical energy, use of environmentally harmful refrigerants, inability to perform cooling and dehumidification independently, poor ventilation and huge power demand, a necessity is created for its replacement with systems which consume less energy, provide better ventilation and rely more upon sustainable energy sources.

Dew point coolers are indirect evaporative systems using Maisotsenko cycle (M-cycle), wherein a portion of the primary air is diverted into an operating passage that provides necessary evaporative cooling to the primary air thereby attaining sub-wet bulb temperatures near to DPT. With direct and indirect evaporative coolers, air-cooling is possible to temperatures near Wet Bulb Temperature (WBT) and not below that. Using the M-cycle, temperatures below WBT and near to DPT can be achieved. Many researchers investigated its performance. Hsu et al. [1] conducted experimental and theoretical analysis of the M-cycle cooling system and observed the influence of significant parameters on its functioning. Hasan [2] proposed four types of indirect evaporative coolers, investigated their cooling performance using a computational model, and concluded that they are capable of attaining temperatures below WBT. Layeni et al. [3] analyzed improving indoor air quality in a lecture hall by using passive ventilation techniques and its potential energy savings using computational fluid dynamics. In order to shed light on the ideal length of dry channels, Fan [4] built an experimental system consisting of an M-cycle cooling tower with parallel counter-flow arrangement fills. Through numerical modeling, Pandelidis [5] carried out a thorough examination of the Maisotsenko cycle (M-cycle) cooling tower. A Number of Transfer Units (NTU) model of the M-cycle Cooling Tower (MCT) was created for the purposes of this study, and it was verified against experimental data that showed the M-cycle's strong practical potential in water cooling applications. The Maisotsenko cycle is used to power the air conditioning system that Khan et al. [6] devised and built. The system was able to condition the air in a small residential building, and under certain circumstances, it handled a cooling load of 2.0 TR. A mini-indirect evaporative cooler was conceptualized and proposed by Dizaji et al. [7]. The cooling characteristics of the cooler are examined through a thorough experimental and theoretical study under various thermal, fluid flow, and geometric conditions. The energy savings that may be achieved by combining a commercially available M-cycle evaporative cooling system with a traditional refrigeration cycle for the air conditioning of office buildings were examined by Zanchini and Naldi [8]. A cooling vest for employees' backs was proposed by Ragheb et al. [9], based on the M-cycle evaporative cooling method. Through simulation and testing, this method is contrasted with direct evaporative cooling vests. The M-cycle vest would result in a lower back temperature, according to the data.

Liquid desiccant systems are an effective way for dehumidification of air. Qu et al. [10] used building energy modelling software to conduct a case study on an ancient structure in order to examine the energy performance of systems implementing integration techniques in actual buildings. In the study done by Afzal et al. [11], an effort is made to maintain human thermal comfort by controlling the temperature and relative humidity inside the car cabin by impregnating an organic Phase Change Material (PCM)-coconut oil underneath the roof of the car and empty spaces in the interior of the doors. Gao et al. [12] used partial internal cooling and found it advantageous. Ge et al. [13] conducted a feasibility analysis of a self-cooled solid desiccant air-conditioner in which internal cooling is provided by evaporative cooling utilizing exhaust air and found that this configuration is capable of supplying air at a suitable condition. Cheng et al. [14] conducted an experimental study of an electro-dialysis regenerator for liquid desiccant which relies on the carrying of ions within relative members with the help of an electric field. Luo et al. [15] used a finned tube to provide internal cooling to the desiccant liquid and optimized the dimensions. Gommed and Grossman [16] conducted an experimental comparison between internally cooled and externally cooled liquid desiccant dehumidifiers and observed that internal cooling improved dehumidifying capacity. In a study done by Mokashi et al. [17], an analysis of the average Nusselt number, which indicates the heat removal from the battery pack cooled by flowing fluid is carried out considering coupled heat transfer conditions at the pack and coolant interface using Artificial Neural Network (ANN) modelling. Yang et al. [18] developed a unique metric called droplet suspension rate coupled with an optimisation design approach and carried out tests to validate it in order to solve this issue. An experimental study was done by Setiyo et al. [19] on the utilization of heat of vaporization of Liquefied Petroleum Gas (LPG) in a vehicle to take part in cooling in the cabin, which is otherwise taken from engine coolant. It is concluded from the test results that this setup can be useful to assist the primary Air-Conditioning (AC) system. The experimental research of the cooling effect obtained in LPG-fuelled automobiles was reported on by Waluyo et al. [20] in their article. Afzal et al. [21] conducted an experimental examination of the thermal performance of a helical tube three-fluid heat exchanger employing three different concentrations of graphene/ water nanofluid as the working fluid. For the triple-fluid helical tube heat exchanger, semi-empirical correlations are given to predict the Nusselt number and friction factor. Jiang et al. [22] examined the output and cost-effectiveness of $CaCl_2$ and LiCl mixture through experimentation and came up with an optimal ratio.

Some researchers worked on new desiccant liquids. Wen et al. [23] prepared an innovative liquid fluid (Li-Cl + hydroxy ethyl) and found that it has a higher dehumidifying capacity and less corrodibility. To reach the appropriate level of comfort, liquid desiccant dehumidification of the air is often followed by air cooling. The effectiveness of systems using liquid desiccant dehumidification followed by direct and indirect evaporative cooling systems was studied by Abdalla and Kamal [24]. Tu et al. [25] and Tu and Ren [26] investigated liquid desiccant dehumidification followed by evaporative cooling systems. Woods and Kozubal [27] conducted sensitivity and parametric analysis to find key design parameters using a numerical model and found an existing compromise between the coefficient of performance and dimensions of a liquid desiccant-based M-cycle air-conditioner. Woods and Kozubal [28] simulated and experimented with a liquid desiccant-assisted regenerative evaporative cooler. In the first step, liquid desiccant that has been evaporatively cooled by exhaust air dries the air; in the second stage, an indirect evaporative cooler cools the air without changing its moisture content. Investigation shows that an improvement in heat and mass characteristics is required. Cui et al. [29] designed a unit in which desiccant and evaporative systems are packed together. It has the advantages of both desiccant liquid and M-cycle systems. Fakhrabadi and Kowsary [30] proposed a desiccant liquid-assisted M-cycle cooling system, identified a set of key parameters and optimized them. By simulating and comparing a proposed liquid desiccant system to a conventional Variable Air Volume (VAV) system, Kim and Park [31] demonstrated the liquid desiccant system's potential for energy savings during indirect and direct evaporative cooler operations. According to simulation data, the suggested system uses 51% less cooling energy than the traditional VAV system. Park et al. [32] studied a liquid desiccant dehumidifier with internal cooling by evaporative method experimentally and found the optimal part of the working air as 0.5.

After an exhaustive literature survey, it has been found that few studies have been done on liquid desiccant-based evaporative cooling systems. Most of these kinds of systems have two separate units for evaporative cooling and liquid desiccant dehumidification and are clumsy. Very few addressed desiccant-based M-cycle indirect evaporative cooling systems in a single unit which is simple in construction, but their performance is limited. With the intention of improving the performance of single-unit desiccant-based indirect evaporative cooling systems, the present study aims to analyse the functioning of a proposed modified desiccant-based M-cycle indirect evaporative cooling system. To represent the physical system, a mathematical model is developed. Later the equations are discretized to carry out the simulation. The simulation was carried out by using a code written in MATLAB. The developed model is validated against the experimental and numerical results in the literature. Finally, the effect of various parameters on the performance of the proposed cooling system is studied. From the numerical study, it was observed that the extent and position of the bypass introduced in the plates separating the two channels influenced the dew point effectiveness of the system and a 19.2% improvement in dew point effectiveness is observed under certain conditions. However, dehumidification effectiveness was observed with the incorporated alterations.

Material and Methods

Description of the cooling system and mathematical model

The proposed sub-dew point cooling system consists of two channels, a product air channel and a working air channel separated by a very fine plastic sheet that restricts mass transfer but allows only heat transfer as shown in Figure 1. Many such arrangements are stacked with each other to suit the required air flow rate. Ambient air enters the product air channel at the bottom and flows upwards. At the left portion of the product air channel where ambient air enters, liquid desiccant is made to flow as a film over the separating plate perpendicular to the direction of flow of air i.e., from top to bottom (perpendicular to the plane of the line diagram of the system in Figure 1, in cross flow arrangement).

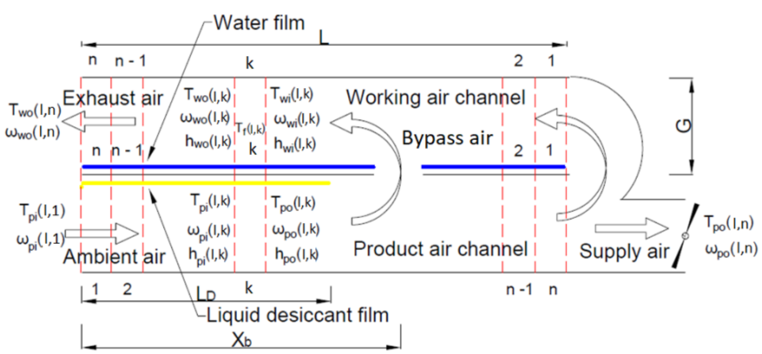

Figure 1: Schematic of the cooling system

After passing through the product air channel, the diluted desiccant liquid is collected at the bottom and delivered to the regenerator where it is concentrated by eliminating moisture and fed back to the unit. The rest of the portion of the product air channel is kept dry and acts as an indirect evaporative cooler. The LiCl-water solution of 40% concentration by mass is used as the desiccant liquid. A portion of the intake air is passed through a secondary air channel and the rest is sent into the room. This diverted part passing downwards through the working channel undergoes evaporative cooling by contact with the cooling pads. These cooling pads are made of porous ceramic panels that hold water. Makeup water is provided to the panels to balance the evaporative loss. Because the diverted part of the primary air is already cooled and dehumidified, evaporative cooling is more effective than cooling product air to temperatures below DPT. The functional air flowing through the secondary air channel is sent out of the system. This does not involve any energy loss because the working air exhausted is almost at ambient temperature and more humid than ambient air. Working air also provides internal cooling to the desiccant liquid by carrying away the latent heat of condensation. In this system, an intermediate opening is made in the wall separating the two channels, slightly above the desiccant liquid flow so that a part of air after dehumidification is bypassed into the working air channel called ‘bypass air’. The bypass air flowing through the liquid desiccant part of the system being drier, makes the air entering the indirect evaporative part colder thereby resulting in a lower temperature of the product air coming out of the system and in a higher dew point effectiveness than in the system without intermediate opening at higher overall working air to product air flow rate ratios. This enhancement in dew point effectiveness is not seen at lower overall working air-to-product air flow rate ratios. There is a slight reduction in dehumidification effectiveness at higher intake temperatures because of the higher temperature of the dehumidified bypass air. For the present analysis, the cooling system dimensions are taken as:

- Length of the plates, L = 1 m, gap in-between the plates, G = 5 mm
- Depth of the plates, w = 100 mm, inlet velocity, v = 1 m/s

A liquid desiccant of 40% aqueous Lithium Chloride solution is considered in the present work. Simultaneous heat and mass transfers take place when air flows over desiccant liquid and water films. The equations representing both the transfer phenomena are to be handled together for the analysis. The following assumptions are taken for the analysis:

- Since channel width is very small compared to the depth, the model is treated as one-dimensional.
- Owing to the very small heat resistance of the water film, the temperature of the water surface is taken the same as the water film temperature.
- The liquid desiccant flow rate is taken constant because the vapour absorption rate is smaller compared to its bulk flow rate.

- Local temperatures of the solution and the water are taken the same because the thickness of the wall separating them is very thin. However thermal resistance offered by the plate is taken into consideration in heat transfer calculations.
- Solution and water cover the entire surface.

A desiccant based evaporative cooling system is shown in Figure 1. The following governing equations are utilized for the computation [2]. Variations of temperature, specific humidity, and specific enthalpy of air along the product air channel are obtained by Equations (1), (2), and (3). Equations (1) and (2) are used in the dry part of the product air channel and Equations (2) and (3) are used in the bottom part where there is liquid desiccant flow because using all the three equations simultaneously makes the calculations over-constrained. The working air channel's specific humidity and enthalpy variations are measured using Equations (4) and (5).

Overall energy balance is used to determine the temperature distribution of water film and desiccant liquid film using Equation (6). The mass balance of water evaporated into working air is not considered as the evaporation rate is very small compared to the water quantity. However, because the absorption rate relies on solution concentration according to the equation, the mass balance of water vapour condensed into a desiccant solution is taken into account using Equations (7) and (8). The specific heat of humid air c_{pa} is taken as 1.026 kJ/kg-K.

Specific enthalpy h of air is related to its temperature T and specific humidity ω with Equation (9). For product air with water, the overall heat transfer coefficient U is derived using Equation (10). The convective heat transfer coefficient is obtained by Equation (11). Nusselt number Nu is taken as 5.597 assuming a fully developed laminar flow with uniform temperature [33]. Thermal conductivity of air k_a is assumed constant as 0.026 W/m-K. Hydraulic diameter D_h is found by Equation (12). The mass transfer coefficient h_m is found by the Reynolds analogy given in Equation (13). Lewis number Le is taken as 1. Vapour pressure $P_{sat,f}$ in saturated air at the water surface is given by Equation (14).

$$\dot{m}_p . c_{pa} . dT_p = U . \left(T_f - T_p\right) . dA \tag{1}$$

$$\dot{m}_p . d\omega_p = h_m . \left(\omega_{p,sat\ at\ T_f} - \omega_p\right) . dA \tag{2}$$

$$\dot{m}_p . dh_p = h_m . \left(h_{p,sat\ at\ T_f} - h_p\right) . dA \tag{3}$$

$$\dot{m}_w . d\omega_w = h_m . \left(\omega_{w,sat\ at\ T_f} - \omega_w\right) . dA \tag{4}$$

$$\dot{m}_w.dh_w = h_m.\left(h_{w,sat\ at\ T_f} - h_w\right).dA \tag{5}$$

$$\dot{m}_w.dh_w + \dot{m}_p.dh_p = m_f.c_{pf}.dT_f \tag{6}$$

$$\dot{m}_p.d\omega_p + \dot{m}_s.dX = 0 \tag{7}$$

where X is $\frac{\text{kg water}}{\text{kg sol}}$

$$X = 1 - \xi \tag{8}$$

ξ is $\frac{\text{kg salt}}{\text{kg sol}}$

$$h = c_{pa}.T + 2500.\omega \tag{9}$$

$$U = \left[\frac{1}{\alpha} + \frac{p}{k}\right]^{-1} \tag{10}$$

where p and k represent the separating sheet's thickness and thermal conductivity, respectively.

$$\alpha = \frac{Nu.k_a}{D_h} \tag{11}$$

$$D_h = \frac{4A}{P} = \frac{4 \times depth \times gap}{2 \times (depth + gap)} \tag{12}$$

$$h_m = \left(\frac{\alpha}{c_{pa}}\right) \times Le^{\frac{2}{3}} \tag{13}$$

$$lnP_{sat,f} = \frac{X_1}{T} + X_2 + X_3.T + X_4.T^2 + X_5.T^2 + X_6.lnT \tag{14}$$

The vapour pressure of saturated air at the desiccant liquid surface is given by Equation (15).

$$P_{sat,s} = \pi.P_{sat,f} \tag{15}$$

where the value of π is found from equations [34].

$$C_{s,LiCl} = C_{H_2O} \times [1 - \varphi_1(T).\varphi_2(\xi)] \tag{16}$$

$$\text{Dew point effectiveness, } \varepsilon_{dew} = \frac{T_{in} - T_{out}}{T_{in} - T_{dew}} \tag{17}$$

where T is the water temperature in Kelvin [34].

$$\text{Dehumidification effectiveness, } \varepsilon_{\text{deh}} = \frac{\omega_{in} - \omega_{out}}{\omega_{in} - \omega_{sat,T_{sol,in}}} \tag{18}$$

Specific thermal capacity C_s of the solution depends on temperature T and salt content ξ and is calculated by Equation (16). $\varphi_1(T)$ and $\varphi_2(\xi)$ are found by the equations [34]. Equations (17) and (18) provide performance parameters for dew point effectiveness and dehumidification effectiveness, which are used to represent the proposed cooler's performance. Numerical computation is carried out using the following boundary conditions: the state of air at the entrance of the primary air passage is that of the ambient and the state of air at the entrance of the working air stream is that of the air leaving the product air channel.

Solution methodology

Computation is an iterative procedure using mass and heat balance equations. For the numerical analysis, the total length in the cooler is divided into n number of small parts as seen in Figure 1. In this study, it is divided into 100 elements to save computational time without compromising on accuracy. The analysis is done in 2 stages. The first stage involves a single iteration. In the first stage, the temperature of the water is taken to be uniform and fixed throughout the length. In this case, it is taken as 3 °C less than the ambient temperature to start with. Computation starts from the entrance of the primary air passage and is carried on up to the exit of the product air channel along it. A fraction of the intake air is directed into the functional air passage so the entering state for the working air channel will be the same as the leaving state of the product air channel. Computation will now be carried along the working channel up to the exhaust. In the primary air channel, air undergoes heat transfer with water through the separating plate and undergoes heat and mass transfer with the flowing desiccant solution. In the working air channel, air undergoes simultaneous heat and mass transfer with the same water stored in the panels. Energy balance is now used to compute the temperature of the water in each element of the panel. Computation is done for the entire length of the water panels starting from the top downwards. The local temperature of the water and desiccant liquid are taken to be the same because the separating plate is very thin. The difference in the product air's humidity ratio is used to compute the concentration of solution in each ingredient. Temperature, specific enthalpy and specific humidity of air at the outlet of each element are calculated from the temperature, specific enthalpy and humidity ratio of air at the inlet of that element. Calculations are done for the product air channel, working air channel, water film and desiccant liquid independently. For this purpose, Equations (1) to (18) are used. The computation procedure can be seen in the flow chart shown in Figure 2.

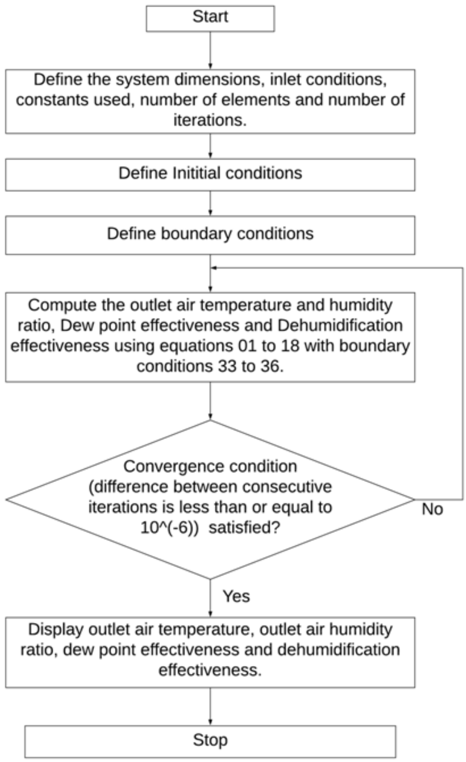

Figure 2: Flow diagram of the computation

Validation

For validating the developed code, the experimental results of Hsu et al. [1] were considered. They conducted experiments with intake air DBT 34.2 ℃, WBT 15 ℃, length of the channel 0.508 m, a working air ratio of 0.5, gap in between the plates 3 mm, air intake velocity 0.96 m/s, dimensionless numbers λ_1 = 3.81 and λ_2 = 0.01. The present computational scheme was applied to the above-mentioned system with the length of the system divided into 100 elements for the analysis. It was observed that any further increase in grid size did not show any significant variation in the results. Hence the grid size of the 100 scheme is validated with the experimental results of the published data [1] and the differences are within ±5% as shown in Figure 3.

Further to verify the effectiveness of the results, the present computational model was validated against the numerical simulation results of Cui [29]. The simulation methodology of the present study using MATLAB was applied to the above study and the values are validated through the outcomes of the above simulation with a maximum discrepancy of ±5% as shown in Figures 4 and 5. Hence the present simulation methodology is considered to predict the performance with reasonable accuracy.

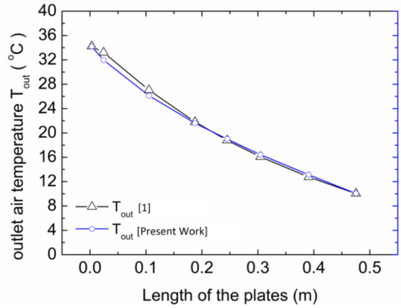

Figure 3: Experimental validation

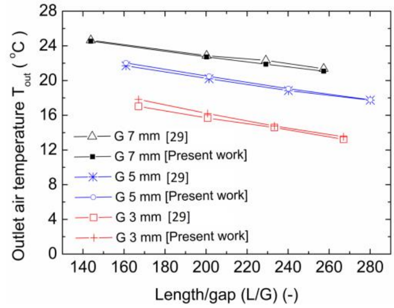

Figure 4: Validation of the model temperature

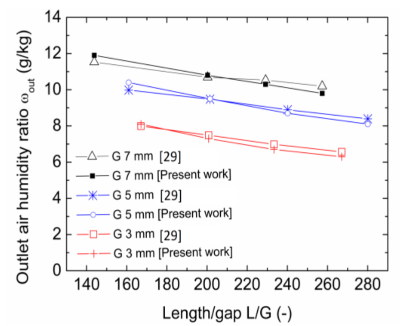

Figure 5: Validation of the model humidity ratio

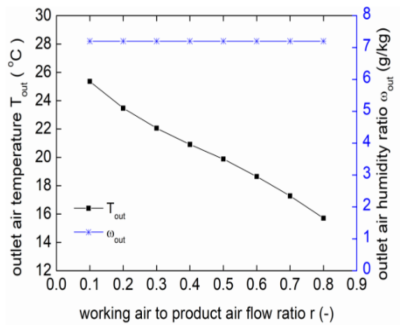

Figure 6: Effect of working air ratio on the leaving air condition

Results and Discussions

In this work, the effect of various influencing parameters on the outlet air condition, dew point effectiveness, and dehumidification effectiveness is studied numerically. The parameters influencing the performance of the proposed cooling system are identified as working air to product airflow ratio r, liquid desiccant flow length to total length ratio L_D/L, input air temperature T_{in}, input air specific humidity ω_{in}, the gap in between plates G, length of the plates L, the ratio of bypassed air to total working airflow ratio r_b, the position of the intermediate opening from the inlet X_b and another additional opening provided. In all the investigations, $r = 0.7$, $L_D/L = 0.5$, $r_b = 0.65$, $X_b = 0.6$ L are taken which give the lowest outlet air temperature as seen in Figures 6, 7, 13, and 14, respectively wherever they are constant.

Influence of working air ratio

The working air stream undergoes evaporative cooling which is essential for effective cooling. A part of product air is used as working air for this purpose. Figure 6 shows the effect of secondary air to product air fraction *'r'* on leaving air temperature and specific humidity, investigated for $r = 0$ to 0.8. Input conditions are $T_{in} = 30$ °C, $\omega_{in} = 16$ g/kg, $L = 1$ m, $L_D/L = 0.5$, $G = 5$ mm, $r_b = 0.65$, $X_b = 0.6$ L. The system showed lower leaving air temperatures and thus improved dew point effectiveness at higher working air to product air fraction. It can be attributed to better indirect cooling due to an increase in the working air flow rate. It has no influence on the leaving air moisture content and therefore on the dehumidification effectiveness due to unvaried solution film length.

Influence of the length of the liquid desiccant flow

In the present work, the solution liquid used helps to achieve higher dew point effectiveness and dehumidification effectiveness. Dehumidified air provides more effective evaporative cooling in the secondary air passage due to low vapour pressure. Figure 7 shows the effect of the ratio of the length of solution flow to total length L_D/L on outlet air temperature and leaving air moisture content. L_D/L values from 0 to 0.6 are taken for the study. Input conditions are $T_{in} = 30$ °C, $\omega_{in} = 16$ g/kg, $L = 1$ m, $r = 0.7$, $G = 5$ mm, $r_b = 0.65$, $X_b = 0.6$ L. By this analysis, it is seen that outlet humidity ratio continuously reduces with increased L_D obviously because of larger area of contact of air with the solution. Outlet temperature seems to reduce initially but rises thereafter. The reason may be with an initial increase in L_D/L ratio, greater dehumidification makes evaporative cooling more effective but with a further increase in L_D/L ratio, heat released by absorption of water vapour in air overtakes this effect resulting in a rise in outlet air temperature. Figure. 8 highlights the influence of L_D/L on the variation of dew point and dehumidification effectiveness. They reflect the same effects seen in Figure 8. Minimum outlet temperatures are shown at L_D/L ratio of 0.5. Dehumidification effectiveness improved continuously with increased L_D/L ratio obviously due to more area of contact with desiccant liquid.

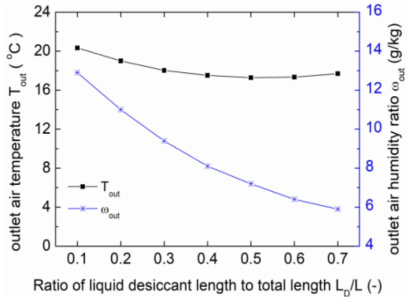

Figure 7: Effect of desiccant length on outlet condition

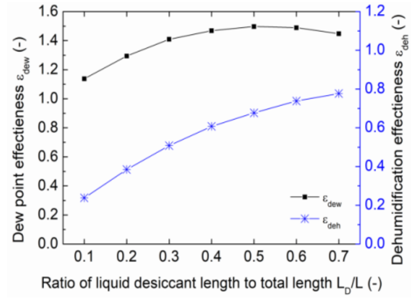

Figure 8: Effect of desiccant length on effectiveness

Influence of inlet temperature and specific humidity

Outdoor condition of air depends on location and season. This section of the study investigates the effect of inlet air condition on the functioning of the proposed cooling system. Inlet temperatures varied from 27.5 °C to 37.5 °C and inlet humidity ratios of 12, 16, and 20 g/kg of dry air were taken for the study. Figure 9 shows the effect of intake air condition on leaving air temperature and Figure 10 shows the effect of intake air condition on the outlet air humidity ratio. Input conditions are L = 1 m, r = 0.7, G = 5 mm, r_b = 0.65, X_b = 0.6 L and L_D/L = 0.5. Outlet temperatures are reduced with decreased inlet temperatures which are obvious. Lower outlet air temperatures resulted in lower inlet air humidity ratios. This may be because of more effective evaporative cooling resulting from lower humidity ratios. Outlet air humidity ratios are lower at lower inlet air humidity ratios which is obvious. Outlet air humidity ratios increased with increased inlet air temperatures. This may be because of the higher temperature of bypassed air which has a lower ability to absorb heat of condensation from the liquid desiccant and to cool the liquid. Desiccants are less effective at dehumidification at rising temperatures.

Influence of channel dimensions

Channel dimensions are the significant variables that influence performance and are also vital due to space restrictions and flow conditions. This section focuses on the effect of channel dimensions on the outlet air condition. Figure 11 highlights the effect of the ratio of length of the plates to the gap in between the plates L/G on outlet air temperature and Figure 12 shows the influence of L/G on outlet air humidity ratio. Lengths of plates L from 0.5 m to 1.2 m and gaps in between the plates G of 3 mm, 5 mm, and 7 mm are considered for the analysis. Input conditions are T_{in} = 30 °C, ω_{in} = 16 g/kg, r = 0.7, r_b = 0.65, X_b = 0.6 L and L_D/L = 0.5. Increased length showed lower outlet temperatures. This may be because of a larger area of contact with a water film. Increased length also resulted in lower humidity ratios. The reason may be the larger area of contact with the liquid desiccant film. However, longer channels need more fan power. Similarly smaller gaps in-between the plates showed lower outlet temperatures and lower outlet air humidity ratios. This seems to be because of lower resistance to heat and mass transfer associated with narrower gaps in between plates. However, this requires additional power.

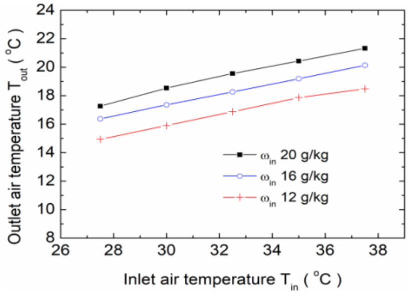

Figure 9: Effect of intake condition on outlet temperature

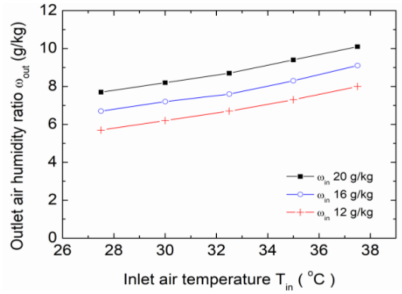

Figure 10: Effect of intake condition on outlet humidity

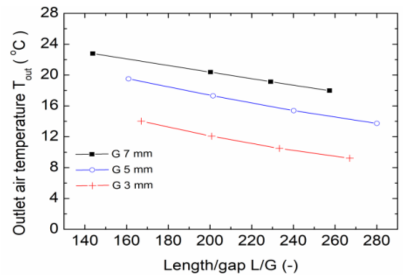

Figure 11: Effect of channel dimensions on outlet air temperature

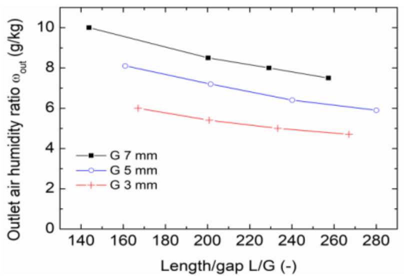

Figure 12: Effect of channel dimensions

Influence of bypass airflow rate

The proposed desiccant assisted regenerative indirect evaporative cooler is provided with an opening in the wall separating the two channels. A fraction of the primary air is bypassed into the secondary air passage through this opening which depends on the size of the opening. The effect of bypass air on outlet air condition is studied in this section. Figure 13 shows the effect of the fraction of bypassed airflow rate to total working airflow rate r_b on outlet air temperature and outlet air humidity ratio for r = 0.6 and r = 0.7. The analysis was done for various values of r_b from 0 to 1. Input conditions are T_{in} = 30 °C, ω_{in} = 16 g/kg, L = 1 m, L_D/L = 0.5, G = 5 mm, r = 0.6 and 0.7, X_b

= 0.6 L. Outlet temperatures reduced with an increase in this ratio up to some value in between 0.6 and 0.7. Beyond that, the leaving air temperature started to increase. The bypassed air is drier and at a higher temperature because air is bypassed from passing through the indirect evaporative cooling part of the product air channel to the right of the desiccant part in Figure 1. The dryness results in a lower temperature of the air entering the indirect evaporative part due to better evaporative cooling in the evaporative channel that takes more heat from the desiccant side and the higher temperature of bypassed air results in a higher temperature of the product air leaving the system. The trend seen in the curve is due to the relative effects of the two. This ratio does not show any effect on dehumidification effectiveness with an unchanged outlet humidity ratio because of unchanged liquid desiccant film length. For working air ratio r = 0.6, the improvement in dew point effectiveness is observed as 8.74% and at r = 0.7, it is observed as 18.89% as seen in Figure 13. So, it is noted that the incorporation of an intermediate opening in the plate separating the two channels improved the dew point effectiveness of the system.

Effect of position of bypass opening

The bypass opening in the wall separating the two-channels influences the performance of the cooling system. This section studies the influence of the position of the bypass opening from the inlet X_b as shown in Figure 1 on outlet air condition. Figure 14 shows the influence of the position of intermediate opening X_b on leaving air temperature and outlet air moisture content. The analysis was done for different positions of the opening with values of ratio X_b/L ranging from 0.5 to 1.0. Input conditions are T_{in} = 30 °C, ω_{in} = 16 g/kg, L = 1 m, L_D/L = 0.5, G = 5 mm, r = 0.7, r_b = 0.65. Since liquid desiccant flows downward, the aperture can only be created beyond the liquid desiccant flow. Instead of flowing downward, water is retained in the panels. Lower outlet temperatures and higher dew point effectiveness were seen with opening placed closer to the liquid desiccant, whereas the latter decreased with opening located at farther positions. This might be a result of the bypass air being dryer at lower places of the intermediate opening, which is assumed to be the explanation for increased dew point effectiveness. The position of the bypass aperture does not affect the specific humidity of the outgoing air. This may be as a result of the constant L_D/L ratio.

Influence of multiple openings on outlet air temperature

In this section, the effect of two openings on the functioning of the cooling system is presented. Figure 15 shows the influence of additional openings provided apart from the bypass opening on leaving air temperature and specific humidity. An additional opening is provided 0.1 L apart from the bypass opening. A comparison was done between performances with one opening and two openings in the wall separating the two channels for r = 0 to

0.8. Input conditions are T_{in} = 30 °C, ω_{in} = 16 g/kg, L = 1 m, L_D/L = 0.5, G = 5 mm, r_b = 0.65, X_b = 0.6 L. Figure 16 shows the influence of an additional opening provided apart from the bypass opening on outlet air temperature and specific humidity ratio at different liquid desiccant flow lengths, L_D/L = 0 to 0.7. Other parameters being T_{in} = 30 °C, ω_{in} = 16 g/kg, L = 1 m, r = 0.7, G = 5 mm, r_b = 0.65, X_b = 0.6 L. A single opening in the wall separating the two air channels shows an improvement in the dew point effectiveness. Multiple openings provided do not show any further improvement in dew point effectiveness or dehumidification effectiveness as seen in Figures 15 and 16.

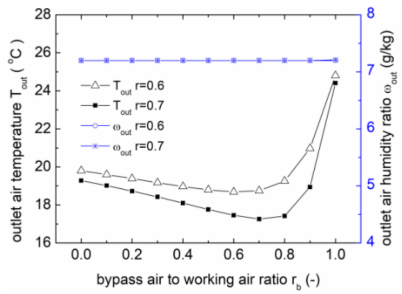

Figure 13: Effect of bypass airflow ratio

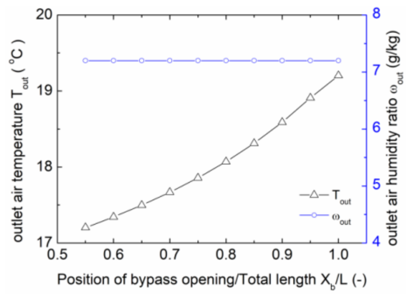

Figure 14: Effect of position of bypass opening on outlet air condition

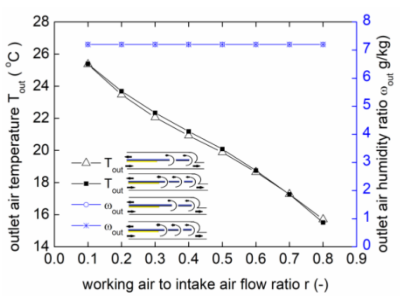

Figure 15: Effect of multiple openings on performance

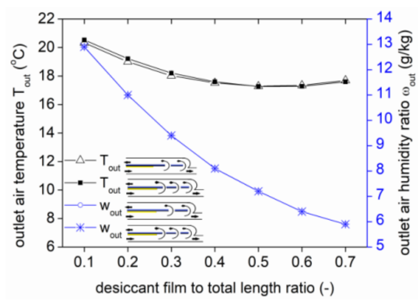

Figure 16: Influence of multiple openings on performance

Conclusion

With the intention to enhance the performance of liquid desiccant assisted single-unit indirect evaporative cooling systems, A simple liquid desiccant based modified regenerative evaporative cooling system is proposed and a mathematical model is developed for simulating its performance. Cooling and dehumidifying performances are investigated using a code executed in MATLAB. The proposed cooling system is found to achieve sub-dew point temperatures under varying conditions of inlet air, while simultaneously dehumidifying the air in a single unit. The proposed air cooling system is suitable as an air-conditioning unit or can be used as a pre-cooler that can reduce the cost of running. The following important points can be gleaned from the present work.

- Increased working air ratio improved ‘dew point effectiveness’ and ‘dehumidification effectiveness’.
- With the increased length of desiccant flow, a reduction in outlet air temperature is observed followed by an increase in the same and a continuous reduction in the humidity ratio of outlet air is observed.
- Increased channel length and decreased channel gap are observed to improve dew point effectiveness and dehumidification effectiveness.
- Dew point effectiveness can be improved with an increased working air flow rate ratio in the combined heat and mass exchanger.
- To improve further, a part of product air is bypassed through an intermediate opening instead of making it flow up to the end of the primary air passage. This modification improved the dew point effectiveness by 19.2% at a working airflow ratio of 0.7.
- Dehumidification effectiveness is unaltered with the introduction of intermediate openings.

Contributions of Authors

The authors confirm the equal contribution in each part of this work. All authors reviewed and approved the final version of this work.

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Conflict of Interests

All authors declare that they have no conflicts of interest.

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