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A Theoretical Analysis on Air-Falling Film Desiccant Dehumidifier

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Abstract – In the present study, the simultaneous heat and mass transfer between air and falling liquid desiccant film along an adiabatic vertical plate in a parallel flow configuration is investigated. The set of the governing equations describing the behavior of the air temperature, humidity ratio, solution temperature and concentration together with their boundary conditions were solved using explicit finite difference method. The effect of the dehumidifier height, air flow rate and air Reynolds number, air gap and the inlet liquid desiccant temperature on the dehumidification process. The present Sherwood and Nusslet number were compared with other researchers under the same condition. The results show that increasing the air flow rate reduce the dehumidification performance, but increases the exit solution concentration. The effect of the increasing air gap, decrease the exit humidity ratio and the absorption rate of water from the air by the solution concentration, where the exit Sherwood and Nusselt numbers increases. On the other hand, the liquid desiccant temperature has a negative impact on the dehumidifier performance. The higher the desiccant temperature, the smaller the change in humidity ratio between inlet and exit.

Keywords**:** Dehumidification; Water desalination; Air-Conditioning; Heat transfer

1. Introduction

The requirements for air-conditioning is rising during hot seasons in some places and in many others throughout the year, in particular in the hot humid climates such as African and Mediterranean countries. The increasing demands of air conditioning have a negative impact on the energy consumption and environment. The most common method in the air-conditioning industry is vapor compression refrigeration systems, which is driven by electrical power. Using the vapor compression system put the power plants at the highest loads during peak demand. Renewable energy such as solar and waste energy can be an appropriate alternative energy source to reduce the energy consumption in particular during the peak demand. Solar and waste energy require the use of different air-condition systems like evaporative cooling, vapor absorption and desiccant based air conditioning systems. Liquid desiccant based dehumidifier – humidifier is one of these technologies that has the great potential of using low-grade energy and is being investigated by many researchers. A desiccant is a sorbent material that has an affinity for water. The usefulness of desiccants in air conditioning systems is where there is a high latent to sensible load ratio and the cost of energy to regenerate the desiccant is low relative to the cost of energy for using a refrigeration cycle for dehumidification [1-3]. Researchers conducting numerical and experimental analysis on the application of liquid desiccant have shown significant improvements in the coefficient of performance of air conditioning systems [4 -7]. Armstrong and Brusewitz [4] conducted an experimental and theoretical studies. They found that the system can effectively dehumidify the air under the conditions tested. M.M. Bassuoni [7] investigated experimentally the performance of a desiccant based air conditioning system. They concluded that the coefficient of performance (COP) of the proposed system is found to be 54% greater than that of VCS with reheat at typical operating conditions.

Other researchers focused on the studies of the simultaneous process of heat and mass transfer between humid air and liquid desiccant, under different geometries and operating conditions [8-11]. Dong et al [8] conducted a technical review on falling film liquid desiccant air dehumidification system. The types of existing liquid desiccant and the existing simulation models were reviewed. They also concluded that the internally-cooled dehumidifiers could achieve better dehumidification efficiency due to the removal of latent heat. Park et al. [9] performed a numerical and experimental study of the steady-state coupled heat and mass transfer between a falling desiccant film and air in cross-flow. They considered both regeneration and absorption processes. Turgut and Çoban [10] investigated experimentally and numerically the performance of an internally cooled dehumidifier using lithium chloride and lithium bromide as liquid desiccants. They found that lithium chloride produced higher dehumidification rates than lithium bromide. Ali et al [11] analyzed the heat and mass transfer of air- falling film in a cross flow configuration using finite difference implicit method. Some of their conclusion was; low air Reynolds number provides better dehumidification and cooling for the exit air conditions, where the desiccant Reynolds number has a minimal effect and the channel dimensions have also an effect on the dehimidfication and cooling. Rahamah et al. [12] investigated numerically the heat and mass transfer process between air and falling liquid desiccant film in concurrent flow heat exchanger. They predicted the effect of inlet conditions, mass flow rates and channel geometry on the air cooling and dehumidification processes. Mesquita et al [13] developed and solved, for a parallel flow configuration, three mathematical models for internally cooled dehumidifiers. The first approach is a simplified model based on heat transfer and heat and mass transfer correlations whereas the second and third approaches are a numerical solution of the differential equations with a constant and variable film thickness using the finite-difference method. Their results show that constant thickness and simplified model under-predicted the dehumidification, especially for low desiccant flow rates. Ali [14] Developed models to investigate the enhancement of air and desiccant film for vertical, and inclined parallel, counter and cross flow. Results show that a low air Reynolds number provides better dehumidification and cooling for all the flow configurations where as a high air Reynolds number improves the regeneration process of desiccant film in both regular and inclined parallel and counter flow channels. Hajji and Worek [15] conducted a theoretical analysis of transient combined heat and mass transfer in a stagnant liquid layer absorbing water vapor with two different boundary conditions at the lower surface of the film. They derived an analytical expressions for the temperature and concentration in a stagnant film of finite depth. Si-Min Hang et al [16] studied the coupled heat and mass transfer in the membrane-formed parallel- plates channels for liquid desiccant air dehumidification. They assumed the flow to be hydrodynamically fully developed while developing in both thermally and concentration. By comparison at various conditions, they found that the Sherwood number is higher than the Nusselt numbers. The flow on the liquid desiccant side, heat transfer is developed shortly after the entry, whereas the mass transfer was found to be still in developing even at the outlet. Nakoryakov et al. [17] studied the heat and mass transfer during vapor absorption by a stagnant solution layer. They described the results of an experimental study of steam absorption by a stagnant water/LiBr solution. In addition, they reported a time dependences of temperatures at various heights inside the liquid layer and of the absorbed mass. Liu X et al [18] developed an analytical solutions of the air and desiccant parameters inside the parallel flow, counter flow and cross flow dehumidifier/regenerator under some reasonable assumptions based on the available heat and mass transfer models. They found good agreement between the analytical solutions of the air and desiccant parameters and the corresponding numerical results. Gandhidasan et al. [19] designed a packed bed column suitable for 5-ton hybrid cooling system to study the absorption of water vapor from moist air by contact with aqueous solutions of calcium chloride. The effects of different independent variables such as air inlet absolute humidity, desiccant inlet temperature, flow rate and its concentration on the performance of the dehumidifier have been investigated.

From the introduction, there are many studies on the performance dehumidifiers. However, it seems that there is still room to upgrade the investigation of the simultaneous heat and mass transfer process that can lead into further improvement of the dehumidifier performance. In this work, the heat and mass transfer between air and liquid desiccant of aqueous solution calcium chloride in a parallel flow configuration is investigated. The effect of the controlling parameters on the dehumidification and cooling processes of the air is analyzed.

2. Mathematical Formulation

A mathematical and numerical models were developed for internally cooled dehumidifiers previously by researchers [13, 20]. Mesquita et al [13] developed a mathematical model and solved numerically for parallel flow internally cooled dehumidifier. Nada [20] developed a model to conduct comparative study of parallel, counter and cross flow internally cooled arrangements. The current work is the study of an adiabatic parallel flow channel between air and falling liquid desiccant film as shown in Fig. 1 where the aqueous solution film falls freely and vertically over the plate and the air flows in concurrent with solution. The mathematical model is formulated under the following assumptions

(1) flow is laminar and steady state,
(2) thermal properties of the air and the aqueous solution are constant
(3) film thickness of the aqueous solution is constant,
(4) velocity profile is fully developed for both flow regimes, and
(5) thermodynamic equilibrium exits at the interface between air and the aqueous solution.

(6) The plate is adiabatic

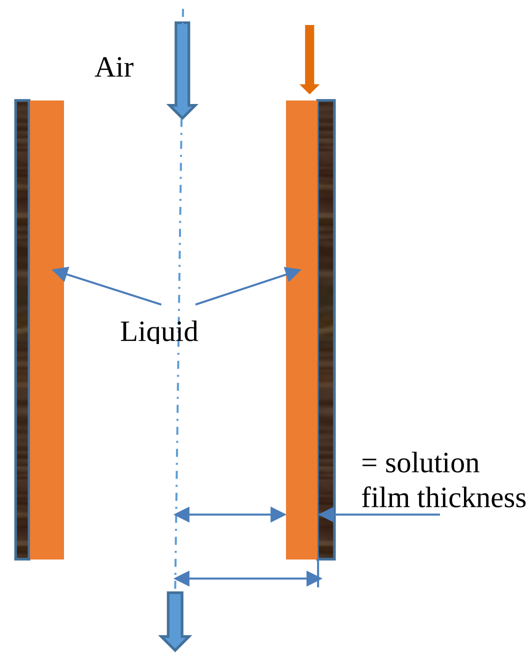

Fig. 1 dehumidifier

2.1 Analysis of the parallel flow channel between air and solution

The governing equations for momentum, energy and mass transfer on the liquid desiccant side are

$$\mu_d \frac{d^2 u_d}{dy^2} + \rho_d g = 0 \tag{1}$$

where μ_d is the desiccant viscosity, ρ_d is the liquid desiccant density and g is the gravitational acceleration.

$$u_d \frac{\partial T_d}{\partial x} = \alpha_d \frac{\partial^2 T_d}{\partial y^2} \tag{2}$$

$$u_d \frac{\partial C}{\partial x} = D_d \frac{\partial^2 C}{\partial y^2} \tag{3}$$

Similarly, the governing equations for momentum, energy and mass transfer on the air side are

$$\mu_a \frac{d^2 u_a}{dy^2} - \frac{dp}{dx} = 0 \tag{4}$$

where μ_a is the air viscosity, ρ_d is the air density and dp/dx is the pressure gradient along the channel which is constant due the fully developed air flow.

$$u_a \frac{\partial T_a}{\partial x} = \alpha_a \frac{\partial^2 T_a}{\partial y_a^2} \tag{5}$$

$$u_a \frac{\partial w_a}{\partial x} = D_a \frac{\partial^2 W_a}{\partial y_a^2} \tag{6}$$

2.2 Initial conditions

The initial condition for the air flow and liquid desiccant at x = 0 and and

$$T_a = T_{ai}\ ;\ w_a = w_{a_i} \qquad at\ \ 0\ \le y_a\ \le\ \delta_a; \quad T_d = T_{di}\ and\ \ C = C_i \qquad at\ \ 0\ \le y_d\ \le\ \delta_d\ \rangle \tag{7}$$

2.3. Boundary condition

The boundary condition at x ≥ 0

i) At the air flow symmetry line at $y_a = 0$

$$\frac{\partial u_a}{\partial y} = 0\ ;\ \frac{\partial T_a}{\partial y_a} = 0\ \ and\ \frac{\partial W_a}{\partial y_a} = 0 \tag{8}$$

ii) At the liquid desiccant-wall boundary *at* $y_d = 0$ *for adiabatic flow*

$$u_d = 0\ ; \quad \frac{\partial C}{\partial y_d} = 0;\ \frac{\partial T_d}{\partial y} = 0 \tag{9}$$

iii) At the air-liquid desiccant interface $y_d = \delta_d$ and $y_a = \delta_a$ the following boundary conditions are applicable

$$\frac{\partial u_d}{\partial y} = 0\ ;\ u_a = u_d\ \ ; T_a = T_d \tag{10}$$

$$k_d \frac{\partial T_d}{\partial y_d} = -k_a \frac{\partial T_a}{\partial y_a} - \rho_a\, D_a\, \frac{\partial W_a}{\partial y_a}\ h_{fg} \tag{11}$$

$$\rho_d\, D_d\, \frac{\partial C}{\partial y_d} = -\rho_a\, D_a\, \frac{\partial w_a}{\partial y_a} \tag{12}$$

Integrating the momentum equation (1) by applying its boundary conditions at $y_d = 0$ $y_d = \delta_d$, the liquid desiccant velocity profile is derived using $\upsilon_d = \frac{\mu_d}{\rho_d}$ to give

$$u_d = \frac{\rho_d\, g}{\mu_d}\left(\delta_d y_d - \frac{1}{2} y_d^2\right) \tag{13}$$

And the film thickness in terms of the liquid desiccant flow rates can be obtained by integrating $\dot{m} = \int_0^{\delta_d} \rho_d u_d\, dy_d$ to give

$$\delta = \left(\frac{3\, \dot{m}_d\, \upsilon_d}{\rho_d\, g}\right)^{1/3} \tag{14}$$

Similarly, integrating the momentum equation (4) by applying its boundary conditions at $y_a = 0$; $y_a = \delta_a$, the air velocity profile is derived to give

$$u_a = u_d - \frac{1}{2\mu_a}\frac{dp}{dx}(\delta_a^2 - y_a^2) \tag{15}$$

And the air pressure gradient is obtained as

$$\frac{dp}{dx} = \frac{3\mu_a}{\delta_a^2}\frac{dp}{dx}\left(u_d - \frac{\dot{m}_a}{2\, \rho_a \delta_a}\right) \tag{16}$$

2.4 Vapor pressure

The vapor pressure for calcium chloride is required air-aqueous solution interface. Bouzanada et al [21] conducted experimental study. They suggested the following correlation for calculating the vapor pressure within a temperature range of 10–65°C and a concentration range of 20–50%.

$$Ln\ (Ps) = A(X) + \frac{B(X)}{T_s + 111.96} \tag{17}$$

$$A(X) = 10.0624\ + 4.4674\ X\ ;\ B(X) = 739.828 + 1450.96\ X \tag{18}$$

3. Method of Solution

An explicit finite difference scheme is used to solve the above set of governing equations. An upwind technique and central differences are used for first and second order derivatives, respectively of both air and desiccant. At the boundary at air- liquid interface an iterative scheme is applied to calculate the temperatures, the liquid desiccant concentration, and the air humidity ratio where a starting value for the humidity ratio is assumed to find all other parameters. After each iteration, the humidity ratio is replaced by the new calculated one. Iteration process continue until the difference between the new humidity ratio and the previous one is less than 0.002%. The solution was validated with numerical data found in the literature. The effect of the mesh size was investigated by using mesh size of 81, 41, 21 and 11 nodes in the y-direction for both air and liquid desiccant. The values of air and liquid desiccant temperatures using 81 nodes compared to their values using 11 nodes has produced 0.83% and 1.338 %, discrepancy respectively at x = 0.01 m.

4. Results and Discussion

In this research, the effect of various parameters on the adiabatic dehumidifier performance has been studied. The humidity ratio of 0.02 kgv/kga, liquid desiccant concentration of 0.06 kgw/kgs and air of $35^{o}C$ at inlet were applied. The effect of the liquid desiccant temperature, T_{di}, air mass flow rate and air Reynolds number on the dehumidification process is conducted. The operating parameters used in the current simulation are presented in table 1.

Table 1: properties and values used in the current simulation

Parameters	Units	CaCl2	Air
T_i	^{o}C	20	35
C_i	kg_w/kg_s	0.6	-
Wa_i	kg_v/kg_a	-	0.02
D	m^2/s	$1.6(10^{-9})$	$2.5(10^{-5})$
α	m^2/s	$1.55(10^{-7})$	$2.208(10^{-5})$
μ	kg/ms	$20(10^{-3})$	$1.872(10^{-5})$
ρ	kg/m^3	1320	1.164
k	W/mK	0.54	0.02588
$\dot{m}$	kg/sm	$7.5\ (10^{-3})$	$15\ (10^{-3})$

The effect of the air mass flow rate on the air temperature and humidity ratio along the dehumidifier (x-direction) are shown in Figures 2 and 3. From Figure 2, it can be seen, at the given inlet conditions, that (1) at lower air flow rate, the exit air temperature reaches it is corresponding minimum at shorter dehumidifier length. At ma = $2.50\ (10^{-3})$ kg/sm, Ta reaches it is minimum at approximately H = 0.1 m, whereas at ma = $20(10^{-3})$ kg/sm it reaches its minimum at a height H = 0.2 m,

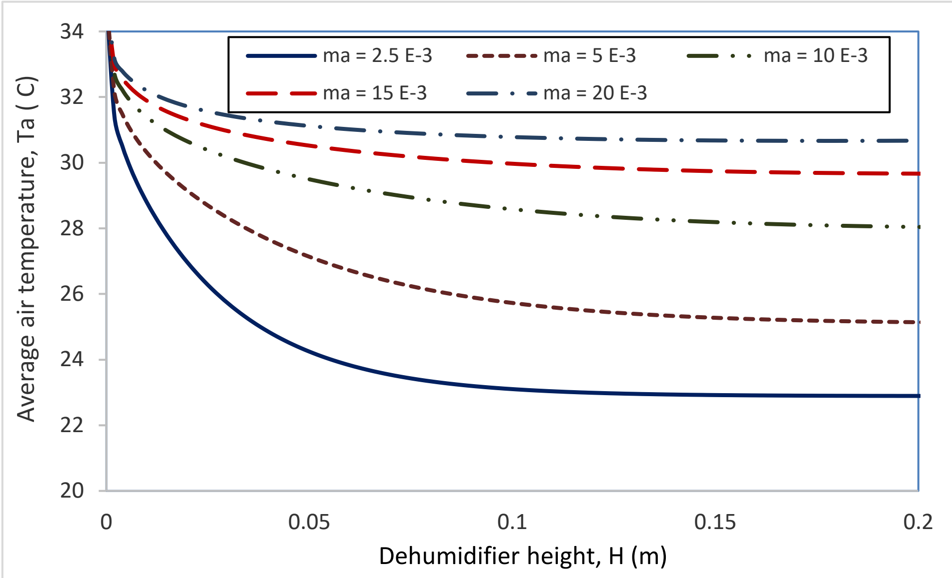

Fig. 2 Air temperature along the dehumidifier height at different air mass flow rate, $\dot{m}_d = 7.5\ \ g/sm, \dot{m}_a = 15,\ \ g/sm\ T_{di} = 20\,^oC, T_{ai} = 35\,^oC, Wa_i\ = \ 20\ g_v/kg$

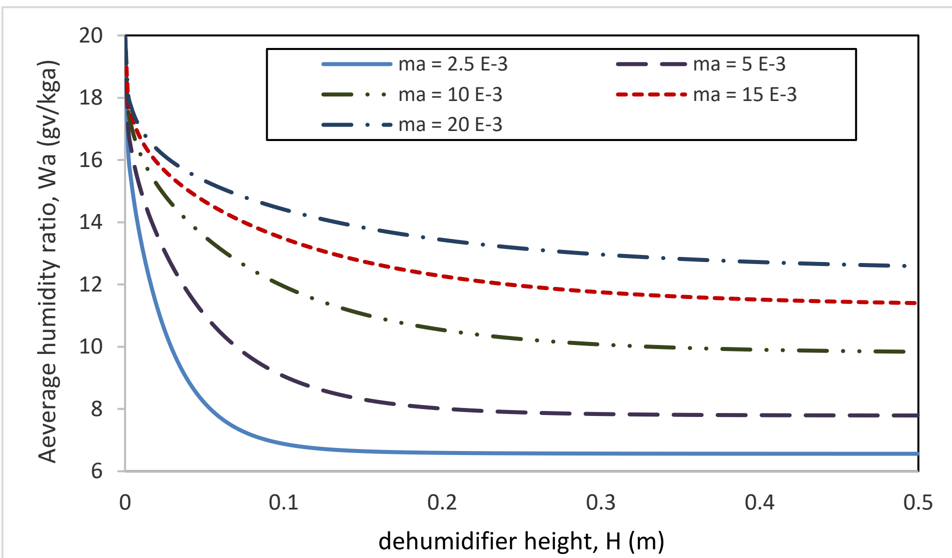

Fig. 3 Humidity ratio along the dehumidifier height at different air mass flow rate, $\dot{m}_d = 7.5\ \ g/sm, T_{di} = 20\,^oC, T_{ai} = 35\,^oC, Wa_i\ = \ 20\ g_v/kg$

(2) also increasing the air mass flow rate, increases the exit air temperature. Similarly Figure 3 shows that decreasing the air mass flow rate, degreases the exit humidity ratio and vice-versa, i.e. the dehumidifier performance drops with increasing the air mass flow rate. On the contrary, the aqueous solution concentration increases with increasing the air mass flow rate. Figure 4, i.e. the rate of water absorbed by the solution is increased with increasing air mass flow rate. However, in practice, since the objective is to dehumidify and cool the air, the dehumidifier must be designed to supply the required state of air to the air-conditioned space. Figure 5 shows the effect of air Reynolds number on the change in the aqueous solution concentration (Co – Ci), exit humidity ratio, the change in humidity ratio between inlet and exit ($Wa_i - Wa_o$) and the rate of water transferred from the air to the liquid desiccant. From figure 5, it can be clearly seen that the change in the solution concentration and the humidity ratio increase with increasing Reynolds number whereas the change in humidity

ratio decreases and this agrees well with results of Figures 3 and 4. However, the rate of water absorbed by the solution increases.

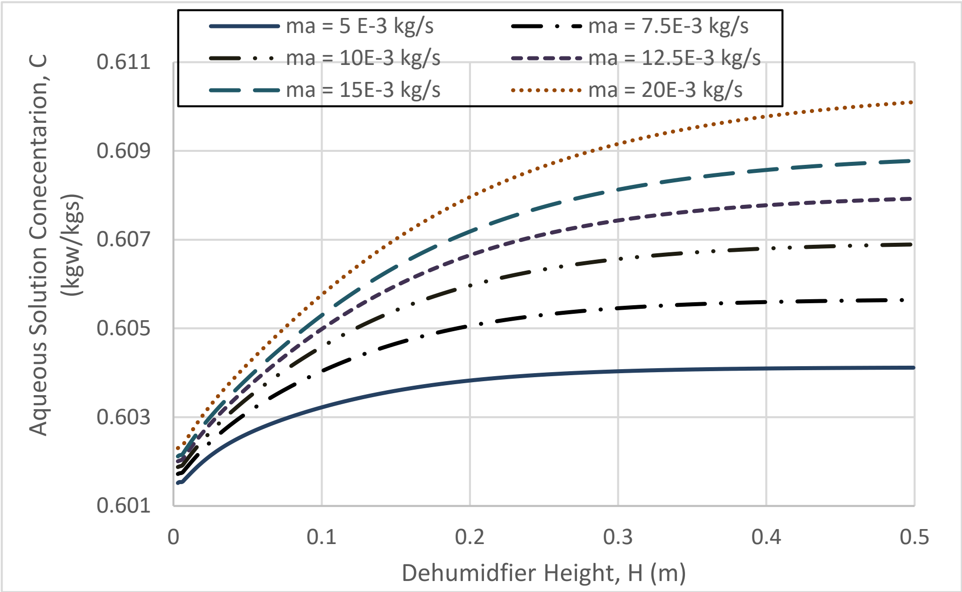

Fig.4 Aqueous solution concentration along the dehumidifier length at different air mass flow rate ratio.
$\dot{m}_d = 7.5\ g/sm, T_{di} = 20\ ^oC, T_{ai} = 35\ ^oC, Wa_i = 20\ g_v/kg$

The effect of the air gap on the dehumidifier performance is also of research interest. Figures 6, 7 and 8 illustrate the effect of the air gap ratio (AR = air gap/base air gap) on the humidity ratio, aqueous solution concentration. Increasing the Air Gap relative to the base Air Gap (Figure 6), decreases the absorption of water by the aqueous solution and also reduces the dehumidification of the air stream. By tripling the air gap, the solution concentration drops by 1.19% and the humidity ratio increase (i.e. it weakens the dehumidification performance of moisture movement from the airstream) by 9.26%. This indicate that, increasing the air gap relative to the base air gap will require more height to complete the dehumidification. Sherwood number (Figure 7) and Nusselt number (Figure 8) are plotted versus dehumidifier height, H at various air gap. The Sherwood number and Nusselt number are defined as

$$She = \frac{4\ \delta_a}{H\ (wa_i - Wa_{si})} \int_0^H \left(\frac{\partial W}{\partial y}\right)_{y=\delta_a} dx \tag{19}$$

$$Nu = \frac{4\ \delta_a}{H\ (Ta_i - Tw)} \int_0^H \left(\frac{\partial T}{\partial y}\right)_{y=\delta_a} dx \tag{20}$$

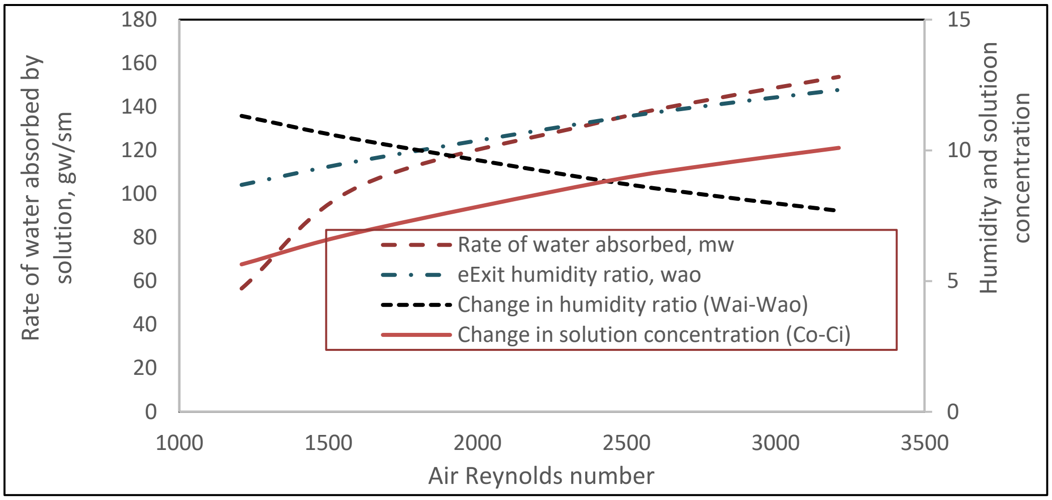

Fig. 5 effect of air Reynolds number on the humidity ratio, solution concentration and the rate of water absorbed. $\dot{m}_d = 7.5\ g/sm, \dot{m}_a = 15\ g/sm, T_{di} = 20\ ^oC, T_{ai} = 35\ ^oC, Wa_i = 20\ g_v/kg$.

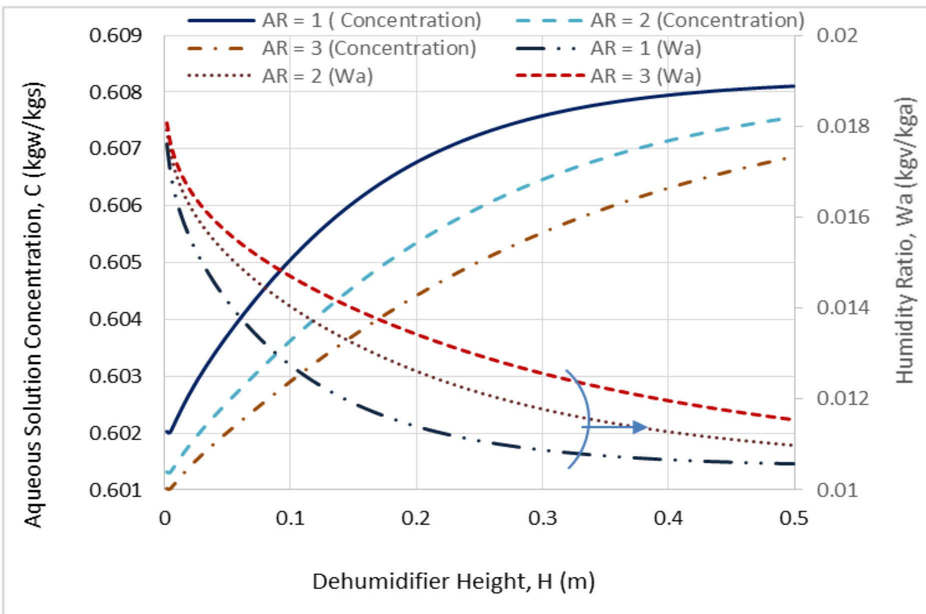

Fig.6 effect of dehumidifier air gap ratio, AR, on the humidity ratio and aqueous solution concentration along the dehumidifier height, $\dot{m}_d = 7.5\ g/sm, \dot{m}_a = 15\ g/sm,\ \ T_{di} = 20\ ^oC, T_{ai} = 35\ ^oC, Wa_i = 20\ g_v/kg$

The Sherwood and Nusselt numbers decrease with increasing the dehumidifier height and increase with increasing the air gap. By doubling the air gap relative to the base one increases the Sherwood number by 82% and the Nusselt number by 108% at H = 0.5 m. This indicate that the dehumidifier will have more potential for heating and cooling with increasing the air gap, which also means larger height to reach the required air cooling and dehumidifying.

A comparison of Sherwood and Nusselt numbers with Rahamah et al [12] and Mesquita et al [13] is conducted and shown in Figure 9 at various dehumidifier heights using the same condition as reported by Mesquita [13]. The Sherwood and Nusselt numbers are higher at the present work. At H = 0.8 m, the present Sherwood and Nusselt numbers, are higher by 17% and 7.58% compared to Mesquita's values respectively, and by 19% and 11.85% compared to Rahamah [12] values respectively. The Sherwood numbers decreases with increasing the dehumidifier height for all three curves and the discrepancy is approximately the same at all heights. The Nusselt number behaves similarly except at H = 0.7, The Nusselt number discrepancy between present work and Mesquita drops down 3%.

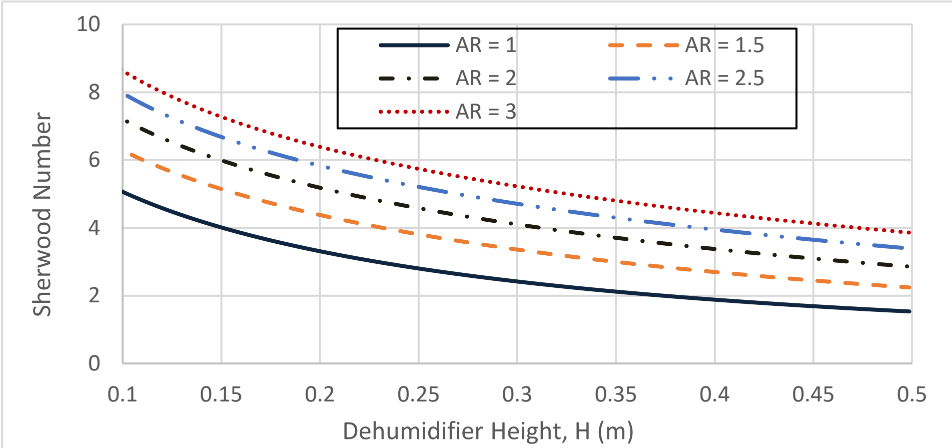

Fig.7 effect of dehumidifier air gap ratio, AR, on the Sherwood number along the dehumidifier height, $\dot{m}_d = 7.5\ g/sm, \dot{m}_a = 15\ g/sm\ , T_{di} = 20\ ^oC, T_{ai} = 35\ ^oC, Wa_i\ =\ 20\ g_v/kg$

The effect of the inlet aqueous solution temperature on the dehumidification process is shown in Figure 10. Obviously as expected, by increasing the inlet solution temperature, Td_i, the air exit air temperature increases, the humidity ratio increases so that the change in humidity ratio between inlet and exit ($Wa_i - Wa_o$) decreases. In other words, the colder the solution temperature, the higher the performance of the dehumidifier.

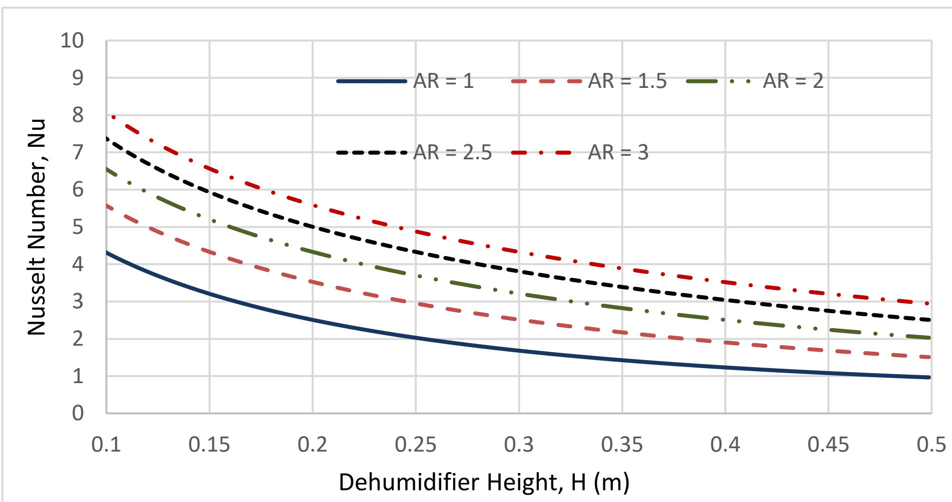

Fig. 8 effect of dehumidifier air gap ratio, AR, on the Nusselt number along the dehumidifier height, $\dot{m}_d = 7.5\ g/sm,\ \dot{m}_a = 15\ g/sm,\ T_{di} = 20\ ^oC, T_{ai} = 35\ ^oC, Wa_i\ =\ 20\ g_v/kg$

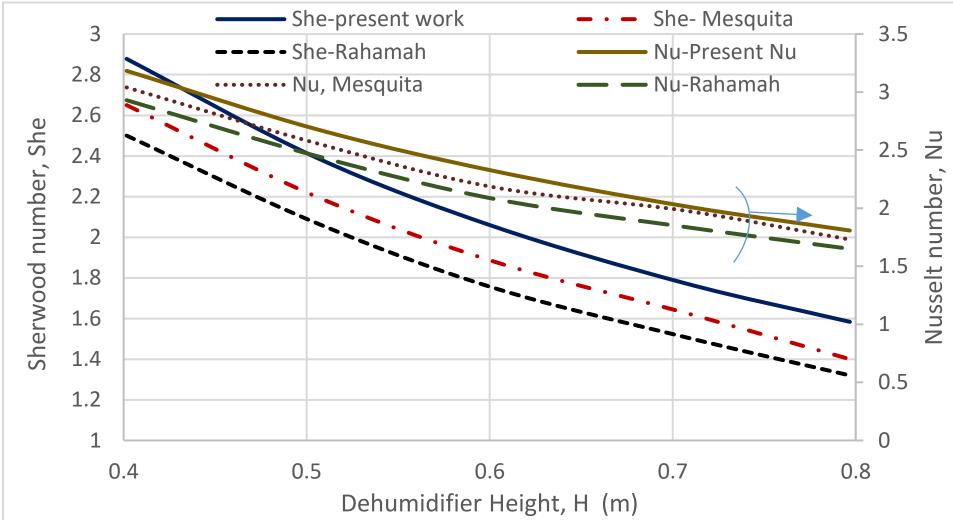

Fig.9 Comparison of Sherwood and Nusselt numbers between present work and Rahamah [13] and Mesquita [14], under the conditions specified in Mesquita [14]

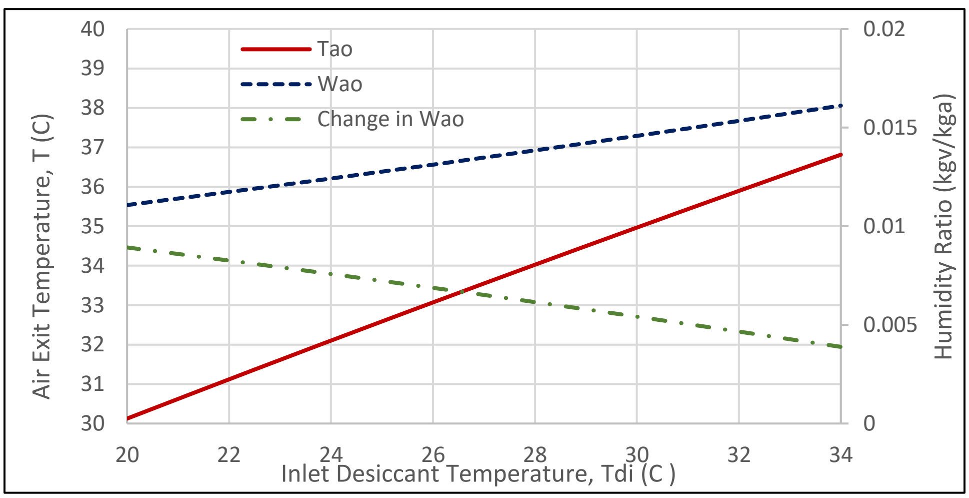

Fig. 10 effect of inlet aqueous solution temperatureTd_i on exit air, aqueous solution temperature, exit humidity ratio and the change in humidity ratio between inlet and exit. $\dot{m}_d = 7.5 \ g/sm, \dot{m}_a = 15 \ g/sm, T_{ai} = 35\,^oC, \ Wa_i \ = \ 20 \ g_v/kg$

5. Conclusions

A mathematical model is solved numerally using finite difference for simultaneous heat and mass transfer in parallel-plate falling film dehumidifiers. The study focused on the effect of air mass flow rates, the inlet aqueous solution temperature and the air gap on the dehumidifier's paramters. A comparison of Sherwood and Nusselt numbers with previous researchers is conducted. Fromm the results, the followings can be concluded; (1) at the base inlet conditions the air temperature reaches it minimum at H = 0.2 m whereas the humidity ratio reaches its minimum at H = 0.5 m. (2) increasing the air mass flow rate, the dehumidifier performance to cool and dehumidify the air drops where the aqueous solution concentration increases with increasing the air mass flow rate, (3) The dehumidifier performance decline significantly with increasing the air gap. Both the cooling and dehumidification of air and the water absorption by the solution drops. By tripling the air gap, the solution concentration drops by 1.19% and the humidity ratio increases by 9.26%, (4) Reducing the inlet solution temperature, improves greatly the dehumidifier performance, (5) the Sherwood and Nusselt numbers are compared with values of Mesquita and Rahamah values. The present work showed discrepancies of 14.72% and 19.2% of Sherwood numbers with values of Mesquita and Rahamah respectively and of 758% and 11.85% of Nusselt numbers with Mesquita's and Rahamah's respectively.

Nomenclatures

AR	Air Gap ratio (air gap/base air gap) (-)
C	aqueous solution concentration ($kg_w\ kg_s^{-1}$)
D	mass diffusivity (m2 s_1)
g	gravitational acceleration ($m\ s^{-2}$)
H	dehumidifier height (m)
h_{fg}	enthalpy of evaporation ($J\ kg^{-1}$)
k	thermal conductivity ($W\ m^{-1}\ K^{-1}$)
m	mass flow rate per unit width, ($kg\ s^{-1}m^{-1}$)
Nu	Nusselt number (-)
Ps	vapor pressure (mmHg)
p	partial pressure, (Pa)
She	Sherwood number (-)
T	temperature, (^oC)
Ts	is the aqueous solution temperature in (°C)
u	fluid velocity in the x-direction ($m\ s^{-1}$)
W	humidity ratio ($kg_v\ kg_a^{-1}$)
X	liquid desiccant concentration ($kg_d\ kg_s^{-1}$)
x	x-coordinate (m)

Greeks

α	thermal diffusivity ($m^2 s^{-1}$)
Δ_a	air gap (m)
δ	Thickness (m)
μ	Viscosity ($kg\ m^{-1}s^{-1}$)
ρ	density, ($kg\ m^{-3}$)
υ	kinematic viscosity ($m^2 s^{-1}$)

Subscripts

a	air
d	desiccant
i	inlet

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