

# A Theoretical Analysis of Hybrid Liquid Desiccant-Vapor Compression Air Conditioning Systems

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**Abstract** - In the present study, the falling film model is applied on the dehumidifier and regenerator in a hybrid system of a liquid desiccant-vapor compression system. The dehumidifier is internally cooled whereas the regenerator is adiabatic. The effect of the internally cooled desiccant on the air temperature and humidity ratio in the dehumidifier are analyzed. The effect of the air mass flow rate on the dehumidifier and regenerator effectiveness were studied. In all cases the effectiveness decreases with increasing the air mass flow rate. The hybrid system COP increases sharply with increasing the air mass flow rate for the range  $\dot{m}_a < 0.0125$  kg/ms and at  $\dot{m}_a > 0.0125$  kg/ms,  $COP_{h_{sys}}$  drops with a steep slope up to  $\dot{m}_a = 0.015$  kg/ms where at higher flow rate it drops at slower rate. The effect of the condenser temperature on the  $COP_{vcs}$ , and the  $COP_{h_{sys}}$ , the compressor work and the energy saving on heating the air prior to entering the regenerator. When the condenser temperature has increase by approximately  $17.5^\circ C$ , The  $COP_{vcs}$  as expected has dropped with increasing the condenser temperature significantly by 47.24% whereas the  $COP_{h_{sys}}$  increased by 49.84%.

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

## 1. Introduction

During the hot-humid seasons, the process air has excessive moisture, which causes discomfort for human and the major load on air conditioning systems. The requirements for air-conditioning is rising during hot seasons in some places and in many others throughout the year. 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 (VCS). Renewable energy can be an appropriate alternative energy source to reduce the energy consumption in particular during the peak demand. Renewable energy require the use of different air-condition systems. Liquid desiccant based dehumidifier – regenerator is one of these technologies that has the great potential of using low-grade energy and is being investigated by many researchers [1-3]. Researchers showed through the application of liquid desiccant significant improvements in the coefficient of performance [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 (COP) of the proposed system was 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 under different 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. Park et al. [9] performed a numerical and experimental study on a cross flow configuration between a falling desiccant film and air. Turgut and Çoban [10] investigated experimentally and numerically the performance of an internally cooled dehumidifier using lithium chloride and lithium bromide as liquid desiccants. Ali et al [11] analyzed the heat and mass transfer of air- falling film in a cross flow configuration using finite difference implicit method. They studied the effect of air Reynolds number, desiccant Reynolds number and the channel dimensions. Rahamah et al. [12] investigated numerically the heat and mass transfer process of a concurrent air-falling liquid desiccant film. 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 heat and mass transfer correlations whereas



Exchanger (H.E.) for further heating if needed prior entering the regenerator. The setting of both the regenerator inlet air and LD temperatures is vital in controlling the regenerator exit conditions

## 2.1. Hybrid LD/VC system

The hybrid system suggested in this work operates as follows (Figure 1); The liquid desiccant forms a falling film along a plate that is cooled by the evaporator keeping the plate isothermally cold while the air is blown in parallel along the film. The outsource energy (solar or waste) is to heat and control the air and LD temperatures before entering the regenerator. This current work is to examine such arrangement. The mathematical model and the assumptions are given below. The liquid desiccant used is Calcium Chloride (CaCl<sub>2</sub>) with properties given in Table 1. 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, (5) thermodynamic equilibrium exists at the interface between air and the aqueous solution, (6) the dehumidifier process is isothermal where liquid desiccant is cooled by the plate which kept at a constant temperature the evaporator. (7) the regenerator process is adiabatic.

Table 1: properties and values used in the current simulation

Parameters	Units	CaCl <sub>2</sub>	Air
$T_i$	$^{\circ}C$	20	35
$C_i$	$kg_w/kg_s$	0.6	-
$Wa_i$	$kg_v/kg_a$	-	0.02
$D$	$m^2/s$	$1.6(10^{-9})$	$2.5(10^{-5})$
$\alpha$	$m^2/s$	$1.55(10^{-7})$	$2.208(10^{-5})$
$\mu$	$kg/ms$	$20(10^{-3})$	$1.872(10^{-5})$
$\rho$	$kg/m^3$	1320	1.164
$k$	$W/mK$	0.54	0.02588
$\dot{m}$	$kg/sm$	$7.5(10^{-3})$	$15(10^{-3})$

## 2.2. 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

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

where  $\mu_d$  is the desiccant viscosity,  $\rho_d$  is the liquid desiccant density and  $g$  is the gravitational acceleration.

$$u_d \frac{\partial T_d}{\partial x} = \alpha_d \frac{\partial^2 T_d}{\partial y^2} \quad (2)$$

$$u_d \frac{\partial C}{\partial x} = D_d \frac{\partial^2 C}{\partial y^2} \quad (3)$$

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

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

where  $\mu_a$  is the air viscosity,  $\rho_a$  is the air density and  $dp/dx$  is the pressure gradient along the channel which is constant due the fully developed air flow.

$$u_a \frac{\partial T_a}{\partial x} = \alpha_a \frac{\partial^2 T_a}{\partial y_a^2} \quad (5)$$

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

### 2.3. Initial conditions

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

$$T_a = T_{ai}; w_a = w_{ai} \quad \text{at } 0 \leq y_a \leq \delta_a; \quad T_d = T_{di} \text{ and } C = C_i \quad \text{at } 0 \leq y_d \leq \delta_d \quad (7)$$

### 2.4. Boundary condition

The boundary condition at  $x \geq 0$

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

$$\frac{\partial u_a}{\partial y} = 0; \quad \frac{\partial T_a}{\partial y_a} = 0 \text{ and } \frac{\partial W_a}{\partial y_a} = 0 \quad (8)$$

ii) At the liquid desiccant-wall boundary at  $y_d = 0$ , the dehumidifier is internally cooled and the regenerator is adiabatic

$$u_d = 0; \quad \frac{\partial C}{\partial y_d} = 0; (T_d = T_w)_{deh} \text{ and } \left( \frac{\partial T_d}{\partial y} = 0 \right)_{reg} \quad (9)$$

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

$$\frac{\partial u_d}{\partial y} = 0; \quad u_a = u_d; \quad T_a = T_d \quad (10)$$

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

$$\rho_d D_d \frac{\partial C}{\partial y_d} = -\rho_a D_a \frac{\partial w_a}{\partial y_a} \quad (12)$$

The falling film liquid desiccant velocity profile is expressed as film thickness can be obtained form

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

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

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

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

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

And the air pressure gradient is obtained as

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

The vapor pressure at the interface between air and Liquid desiccant is calculated using Bouzanada's correlations [17]

### 3. Method of Solution

An explicit finite difference scheme is used to solve the above set of equations for the dehumidifier and regenerator simultaneously. An upwind technique and central differences are used for first and second order derivatives, respectively of both air and liquid desiccant. At the boundary at air- liquid desiccant interface an iterative scheme is applied to calculate the system parameters. The solution was validated with numerical data found in the literature. After the effect of the mesh size is investigated, a mesh size in the y-direction of 41 nodes for both the air and the liquid desiccant was adopted. The above circuit in Figure 1 is simulated iteratively to find the operating balance point of the hybrid system. The liquid desiccant-to-liquid desiccant heat exchanger is assumed to have an effectiveness of 80% [21-22] and that for condenser and evaporator is 85%. The VCS coefficient of performance, COP is defined as

$$COP = \frac{Q_e}{W_c} \quad (17)$$

Where the  $Q_e$  is the refrigeration capacity required to cool the desiccant to the required temperature and  $W_c$  is the compressor Power.

Overall coefficient performance of the hybrid system may be defined as ratio of the useful latent and sensible cooling of the air in the dehumidifier to the sum of the compressor work and the outsource energy required to heat the air and/or the LD prior to entering the regenerator

$$COP_{hsys} = \frac{Q_{deh}}{W_c + W_{outsource}} \quad (18)$$

Where  $Q_{deh}$  is the energy removed from the process air during the dehumidification.

Since the effectiveness is a measure of the hybrid system capability it is defined for the dehumidifier and regenerator based on (1) the enthalpy effectiveness,  $\varepsilon_h$  [21-23], (2) the moisture effectiveness,  $\varepsilon_m$  as follows;

$$\varepsilon_h = \frac{ha_{out} - ha_{in}}{ha_{out-max} - ha_{in}}; \varepsilon_m = \frac{Wa_{in} - Wa_{out}}{Wa_{in} - Wa_{in,in}} \quad (19)$$

## 4. Results and Discussion

### 4.1 Dehumidifier

In this research, the effect of the air mass flow rate on isothermally cooled dehumidifier is studied. The inlet condition are; the humidity ratio of  $0.02 \text{ kgv/kgd}$ , liquid desiccant concentration of  $0.06 \text{ kgw/kgd}$  and air of  $35^\circ\text{C}$  at inlet are applied. 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, 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  $\dot{m}_a = 7.50 (10^{-3}) \text{ kg/sm}$ ,  $T_a$  reaches it is minimum at approximately  $H = 0.5 \text{ m}$ , whereas at  $\dot{m}_a = 30(10^{-3}) \text{ kg/sm}$  and  $H = 0.5 \text{ m}$

the air temperature is still decreasing and far from reaching its minimum. (2) also increasing the air mass flow rate, increases the exit air temperature. Similarly Figure 3 shows that decreasing the air mass flow rate, decreases the exit humidity ratio and vice-versa,. Similarly figure 3 shows that decreasing the air mass flow rate, decreases the exit humidity ratio and vice-versa, i.e. the dehumidifier performance to cool and dehumidify the air drops with increasing the air mass flow rate. On the contrary, the aqueous solution concentration increases with increasing the air mass flow rate, i.e. the absorption of water by the solution is increased which must be the same amount of water removed from the air stream.

The effect of the air mass flow rate on the moisture and enthalpy effectiveness along the dehumidifier (x-direction) are shown in Figures 4 and 5. Both effectiveness behaves similarly and a close looking into the figures, the curves are quite close. However, when the mass flow rate drop below the base value (i.e.  $ma = 15 (10^{-3}) kg/sm$ ), the effectiveness tends to reaches the value of 1 at  $x = 0.5 m$ . At higher air flow rates the effectiveness drop below 1. At e.g.  $ma = 40 (10^{-3}) kg/sm$ , the exit effectiveness drop to 0.745. This agrees well with Figure 3 above.

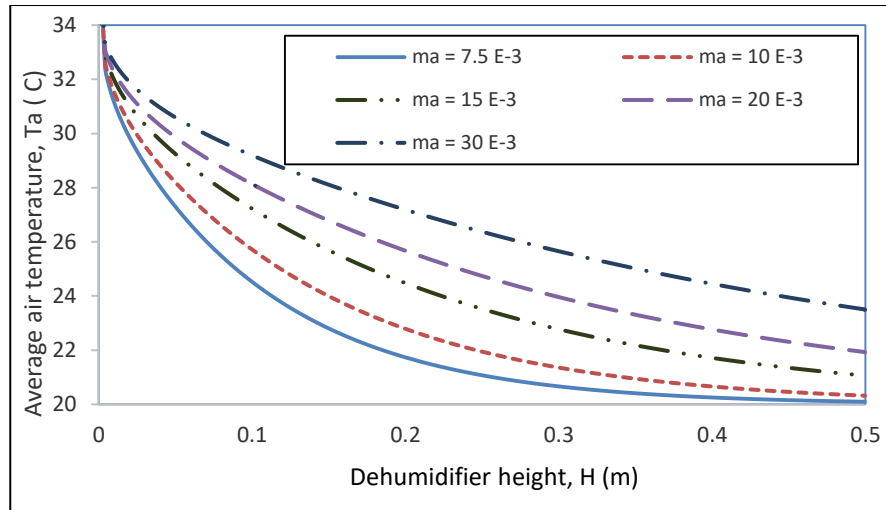


Fig. 2 Air temperature along the dehumidifier height at different air mass flow rate,  $\dot{m}_d = 7.5 g/sm$ ,  $T_{di} = 20 ^\circ C$ ,  $T_{ai} = 35 ^\circ C$ ,  $Wa_i = 20 g_v/kg$ ,  $T_{con} = 46.2 ^\circ C$

## 4.2 Regenerator

The regenerator in this hybrid system is an adiabatic humidification process. It is required to regenerate the liquid desiccant to obtain the same concentration as required at the inlet of the dehumidifier. Therefore the inlet air temperature and the liquid desiccant temperature must be controlled to achieve this condition. Figure 6 shows the regenerator moisture effectiveness varying with air flow rate. It should be noted here that the inlet LD was kept at  $60 ^\circ C$  for all flow air flow rates whereas the air temperature was varied to adjust the exit liquid desiccant concentration at  $0.6 kg_w/kg_s$ . From the Figure, it can be seen that the effectiveness increases along the regenerator but decreases with increasing the air flow rates. The maximum effectiveness in this adiabatic regenerator is 0.361 at an air flow rate of  $10 (10)^{-3} kg/ms$ . By increasing the flow to  $40 (10)^{-3} kg/ms$ , it drop down to 0.22. From Figure 7 it can be seen that the enthalpy effectiveness reaches its asymptote earlier at high mass flow rate. At  $40 (10)^{-3} kg/ms$ , the enthalpy effectiveness is 0.11 where it goes up to 0.353 at  $10 (10)^{-3} kg/ms$ .

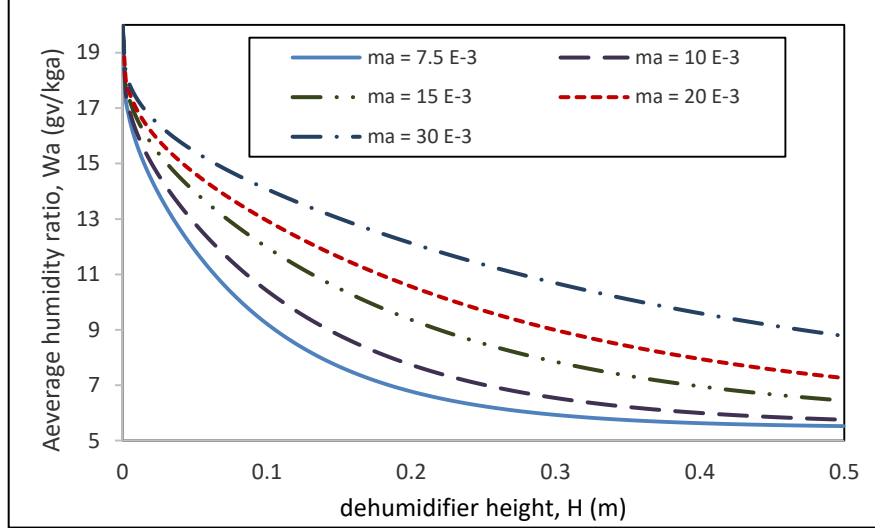


Fig. 3 Humidity ratio along the dehumidifier height at different air mass flow rate,  $\dot{m}_d = 7.5 \text{ g/sm}$ ,  $T_{di} = 20^\circ\text{C}$ ,  $T_{ai} = 35^\circ\text{C}$ ,  $Wa_i = 20 \text{ g}_v/\text{kg}$ ,  $T_{con} = 46.2^\circ\text{C}$

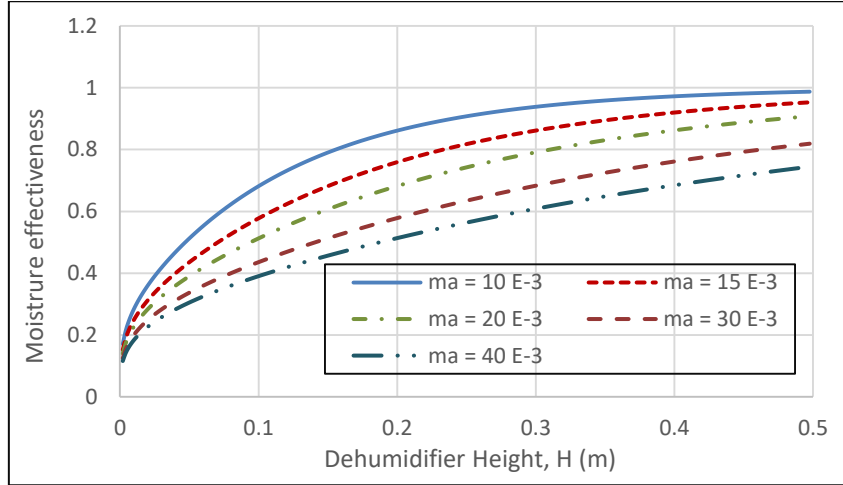


Fig. 4 Moisture effectiveness versus the dehumidifier height at different air mass flow rate,  $\dot{m}_d = 7.5 \text{ g/sm}$ ,  $T_{di} = 20^\circ\text{C}$ ,  $T_{ai} = 35^\circ\text{C}$ ,  $Wa_i = 20 \text{ g}_v/\text{kg}$ ,  $T_{con} = 46.2^\circ\text{C}$

#### 4.3 Hybrid system overall coefficient of performance

The effect of the air mass flow rate on the hybrid system overall coefficient  $COP_{hsys}$  has also been studied. Figure 8 shows the dependence of the  $COP_{hsys}$  on the  $ma$ . As mentioned above, to maintain the regenerator exit liquid desiccant concentration of  $0.6 \text{ kg}_w/\text{kg}_s$ , the regenerator inlet condition was controlled by keeping the liquid desiccant at  $60^\circ\text{C}$  and adjusting the air temperature. Other setting can also work. The corresponding air temperature are well shown in figures 6.

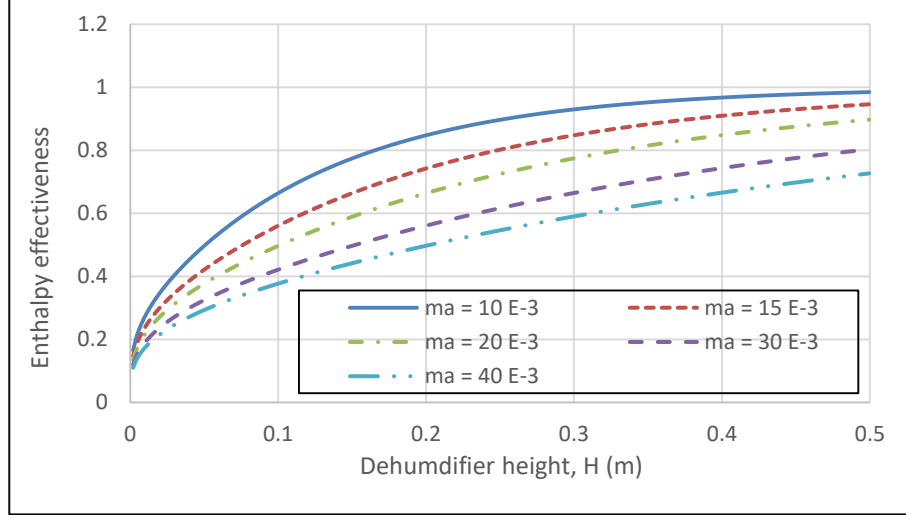


Fig. 5 Enthalpy effectiveness versus the dehumidifier height at different air mass flow rate,  
 $\dot{m}_d = 7.5 \text{ g/sm}$ ,  $T_{di} = 20^\circ\text{C}$ ,  $T_{ai} = 35^\circ\text{C}$ ,  $Wa_i = 20 \text{ g}_v/\text{kg}$ ,  $T_{con} = 46.2^\circ\text{C}$

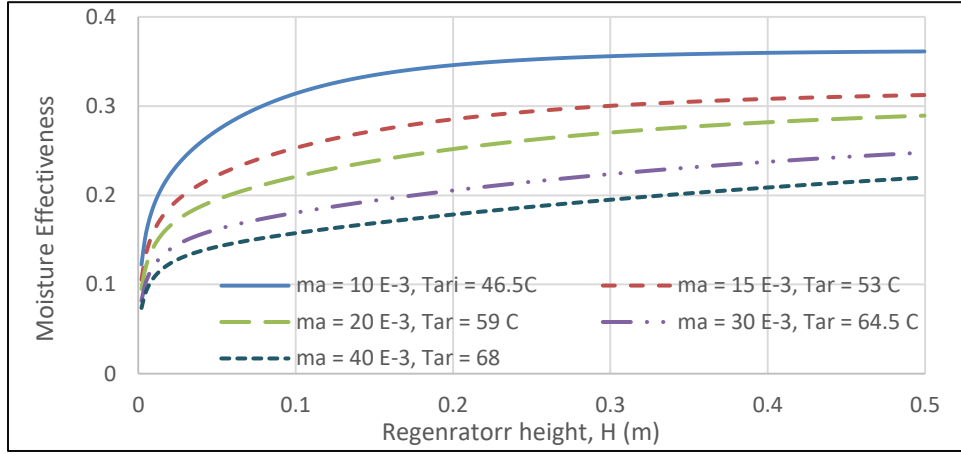


Fig. 6 Regenerator Moisture effectiveness versus the regenerator height at different air mass flow rate,  
 $\dot{m}_d = 7.5 \text{ g/sm}$ ,  $T_{di} = 60^\circ\text{C}$ ,  $Wa_i = 20 \text{ g}_v/\text{kg}$ ,  $T_{con} = 46.2^\circ\text{C}$

From Figure 8, the  $COP_{hsys}$  increases with increasing the air mass flow and reaches maximum at mass flow rate of  $0.0125 \text{ kg/sm}$ . The total power required (compressor power and the outsource heat) increases approximately linearly with air flow rate. The large increase in COP at  $ma < 0.0125 \text{ kg/ms}$  is due to the fact that the outsource energy required in this range is relatively low where the inlet air temperature is relatively small. When the air mass flow rate increases beyond  $0.0125 \text{ kg/ms}$ , the inlet air temperature increases at faster rate to balance the exit liquid desiccant concentration and so the energy required.

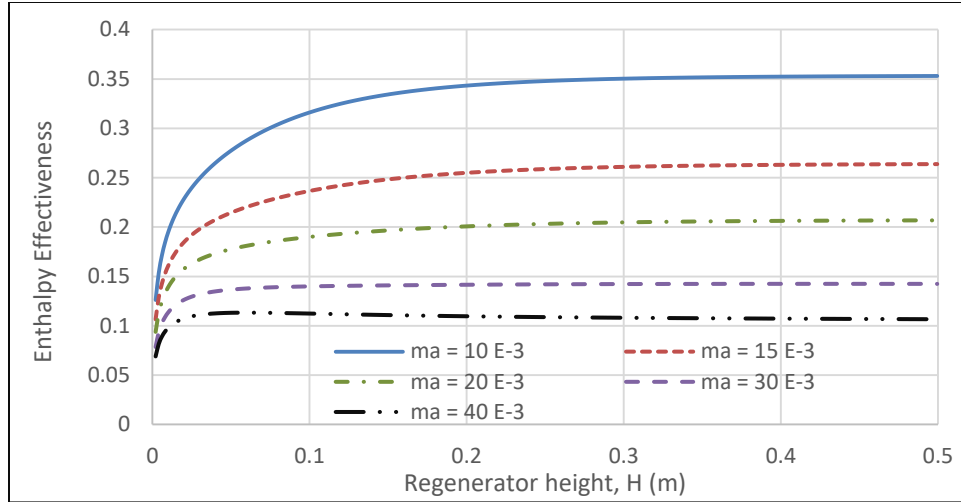


Fig. 7 Regenerator enthalpy effectiveness versus the regenerator height at different air mass flow rate  
 $\dot{m}_d = 7.5 \text{ g/sm}$ ,  $T_{dri} = 60 \text{ }^\circ\text{C}$ ,  $Wa_i = 20 \text{ g}_v/\text{kg}$ ,  $T_{con} = 46.2 \text{ }^\circ\text{C}$

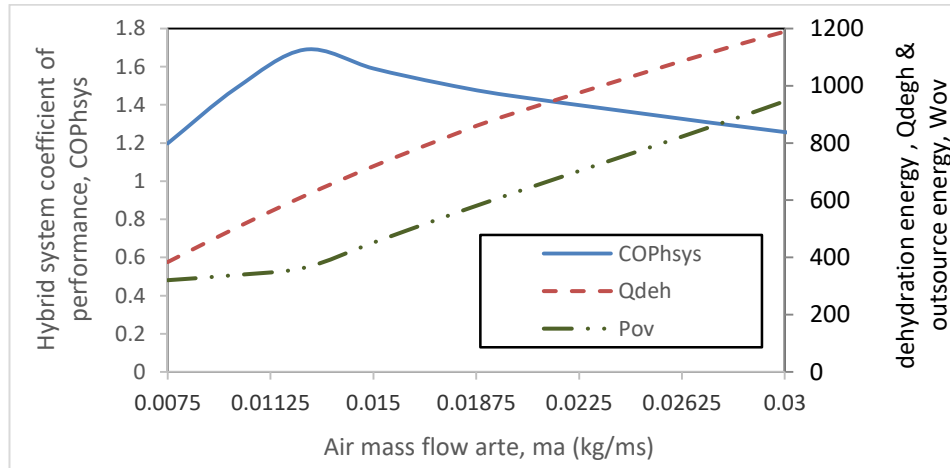


Fig. 8 the effect of air mass flow rate on the hybrid system COP,  
 $\dot{m}_d = 7.5 \text{ g/sm}$ ,  $T_{dri} = 60 \text{ }^\circ\text{C}$ ,  $Wa_i = 20 \text{ g}_v/\text{kg}$ ,  $T_{con} = 46.2 \text{ }^\circ\text{C}$

From Figure 9, we can see under the applied conditions, that with increasing the condenser temperature, the COP of the hybrid system increases slightly whereas the COP of the vapor compression system decreases significantly. Based on the operating conditions, the saving energy by heating the outside air is relatively small to make a significant change on the hybrid system COP, whereas the increase in compressor work has a significant impact on COP of the vapor compression system. The energy saving to compressor work ratio is shown in Figure 9. It is clearly that ratio increases with increasing the condenser temperature up to 12 times. When the condenser temperature has increased by approximately  $17.5^\circ\text{C}$ , The  $COP_{vcs}$  as expected has dropped significantly by 47.24% whereas the  $COP_{hsys}$  increased by 49.84%.

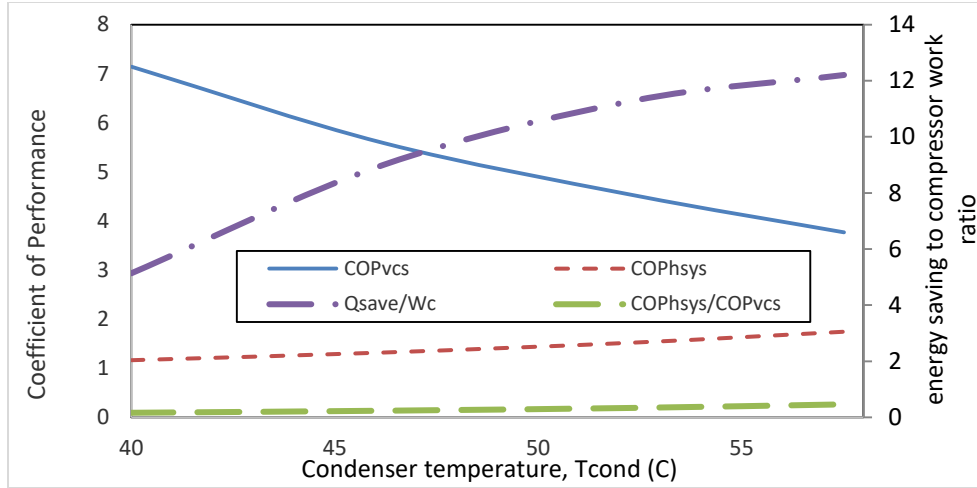


Fig. 9 the effect of the condenser temperature on the COP and the energy saving to compressor work ratio  
 $\dot{m}_a = 7.5 \text{ g/sm}$ ,  $T_{di} = 59.2 \text{ }^\circ\text{C}$ ,  $T_{a_{ri}} = 60 \text{ }^\circ\text{C}$ ,  $W_{a_i} = 20 \text{ g}_v/\text{kg}$ ,  $T_{con} = 46.2 \text{ }^\circ\text{C}$

## 5. Conclusions

A hybrid system arrangement is developed and studied. The dehumidifier is assumed internally cooled and the regenerator is adiabatic. Both the dehumidifier and regenerator are modeled as falling film parallel flow configuration. The study focused on the effect of air mass flow rates, the regenerator inlet liquid desiccant temperature on the dehumidifier's air temperature, humidity ratio, as well as on moisture and enthalpy effectiveness of both and the dehumidifier and the regenerators. In addition the effect of these two parameters on the hybrid system coefficient of performance. From this analysis, it can be concluded; (1) increasing the air mass flow rate the exit dehumidifier increases which reduces its performance. It also decrease both effectiveness in both the dehumidifier and the regenerator. 2) The hybrid system COP increases with increasing the air flow rate up  $\dot{m}_a = 0.0125 \text{ kg/ms}$  and then it begins to drop. This due to the fact that the regenerator air inlet condition must be adjusted to maintain the exit regenerator condition where the regenerator inlet air temperature increases at faster rate to balance the exit liquid desiccant concentration and so the energy required. (3) COP of the vapor compression system decreases with increasing the condenser temperature whereas the COP of the hybrid system increases. It should be noted that, For better extracting the usefulness of hybrid systems such as the current one, a comparison with the available hybrid system designs studied by various researchers it would highly beneficial. Therefore such a comparison should done in the near future.

## Nomenclatures

$C$	aqueous solution concentration ( $\text{kg}_w \text{ kg}_s^{-1}$ )
$COP$	the VCS coefficient of performance
$COP_{hsys}$	hybrid system coefficient of performance
$D$	mass diffusivity ( $\text{m}^2 \text{ s}^{-1}$ )
$g$	gravitational acceleration ( $\text{m s}^{-2}$ )
$H$	dehumidifier height (m)
$h_{fg}$	enthalpy of evaporation ( $\text{J kg}^{-1}$ )
$k$	thermal conductivity ( $\text{W m}^{-1} \text{ K}^{-1}$ )
$\dot{m}$	mass flow rate per unit width, ( $\text{kg s}^{-1} \text{ m}^{-1}$ )
$P_{ov}$	total power required to run system W/m
$P_s$	vapor pressure (mmHg)
$p$	partial pressure, (Pa)
$Q_{deh}$	the dehumidification energy W/m

$T$	temperature, ( $^{\circ}C$ )
$T_d$	liquid desiccant temperature in ( $^{\circ}C$ )
$u$	fluid velocity in the x-direction ( $m\ s^{-1}$ )
$W$	humidity ratio ( $kg_v\ kg_a^{-1}$ )
$W_c$	compressor power (W/m)
$W_{outsource}$	outsource power (W/m)
$X$	liquid desiccant concentration ( $kg_a\ kg_s^{-1}$ )
$x$	x-coordinate ( $m$ )

#### Greeks

$\alpha$	thermal diffusivity ( $m^2\ s^{-1}$ )
$\delta$	Thickness (m)
$\varepsilon$	effectiveness
$\mu$	Viscosity ( $kg\ m^{-1}\ s^{-1}$ )
$\rho$	density, ( $kg\ m^{-3}$ )
$\nu$	kinematic viscosity ( $m^2\ s^{-1}$ )

#### Subscripts

$a$	air
$d$	desiccant
$i$	inlet
$h$	enthalpy
$m$	moisture
$r$	regenerator

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