Int. J. of Thermal & Environmental Engineering
Volume 4, No. 1 (2012) 39-46

Modeling and Simulation of Desiccant Operated Humidity Pump (DOHP)

A. M. Hamed[a, b, *], A. S. Alosaimy[b]

[a] *Mechanical Power Department, Faculty of Engineering, Mansoura University*
[b] *Mechanical Engineering Department, Faculty of Engineering, Taif University, Saudi Arabia*

Abstract

In the present work, novel configuration of rotating desiccant wheel using calcium chloride as the working desiccant is presented. The proposed system is called desiccant operated humidity pump (DOHP). Mathematical model, which can be applied for analysis of the proposed system, is developed. The proposed system can be powered by low grade heat sources such as solar energy. Absorption-regeneration cycle for the DOHP is described and analyzed. An expression for the efficiency of the simple cycle is introduced. Theoretical analysis shows that strong and weak solution concentration limits play a decisive role in the value of cycle efficiency. System efficiency with consideration of heat and work added to the system is well defined. Dimensionless parameters defining the system design parameters are introduced. The limits of regeneration temperature and mass of strong solution per kg of produced vapor are found highly dependent on the operating concentration of desiccant.

Keywords: *Desiccant; humidification; absorption; air conditioning; Calcium chloride; humidity pump*

1. Introduction

Sorption air conditioning systems attain considerable attention as they can be driven either by waste heat sources or by renewable energy sources. Several desiccant dehumidifier configurations including solid packed bed, rotating horizontal bed, multiple vertical beds, rotating honeycomb, fluidized bed, and inclined bed have been investigated for dehumidification purpose [1-8].

The investigation of the combined heat and mass transfer characteristics accompanied with such a sorption process is, therefore, very important in designing and optimizing the operation of sorptive based dehumidifiers. Much research on the solid desiccant dehumidifiers has been accomplished, and many successful, effective mathematical models to predict the heat and mass transfer process in such dehumidifiers have emerged [9-11].

Desiccant materials may be in a liquid or solid form. The use of solid adsorbents usually requires a relatively high regeneration temperature, compared with liquid desiccant such as calcium chloride and lithium chloride. On the other hand, the use of liquid sorbent leads to the need to use an apparatus in which continuous transport of liquid solution is carried out. However, liquid desiccant can be used in a stationary bed when a solid porous material carrying the solution is used [12–14].

For improving the performance of desiccant dehumidification systems, new approaches are highly welcome. An attempt to improve the performance of rotary desiccant bed using liquid absorbent should include an investigation on the performance of new system configurations. In the present study a novel configuration of desiccant operated humidity pump is presented. The aim of the present work is to evaluate the effect of system design parameters and ambient conditions on the performance, when desiccant wheel with liquid desiccant is applied. Also, it is objected to determine the adsorption limits at which maximum system efficiency can be attained.

2. Isothermal Desiccant Operated Humidity Pump (DOHP)

Compared with the vapor compression heat pump, desiccant dehumidification system, in general, functions as humidity pump. The function of the humidity pump is to transfer the humidity of room air (indoor condition) to the outside air (outdoor conditions). In most cases, the indoor humidity is objected to be lower that of the outdoor air. The energy required to power such systems is mainly the regeneration heat required to heat the desiccant regenerator. The basic concept of humidity pump is demonstrated in Fig. 1. As shown in figure, air at the indoor conditions is dehumidified isothermally through the dehumidification process R-O, where the conditions R and O represent the room and outside conditions, respectively. The system which carries out this process is called a humidity pump (HP). However, isothermal absorption

[*] Corresponding author. Tel.: +966599121850
Fax: +96676543210; E-mail: amhamed@mans.edu.eg

DOI: 10.5383/ijtee.04.01.006

of water vapor from air can be carried out with continuous cooling of the desiccant during the process. At the end of absorption the desiccant must be regenerated to remove the absorbed water and re-concentrate the solution.

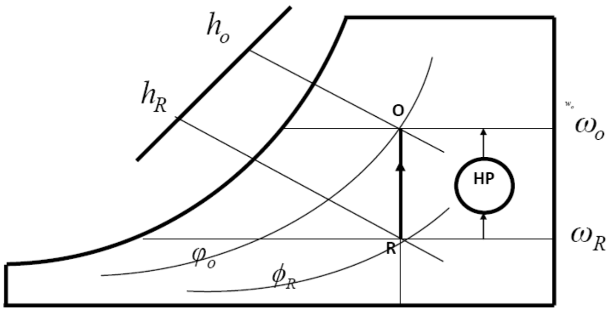

Fig. 1: Humidity pump concept.

The absorption-regeneration cycle, which can be applied, for operation as a humidity pump, is shown in Fig. 2. The theoretical cycle is plotted on the vapor pressure-concentration diagram for the operating absorbent and consists of four thermal processes which are [15]:

- Process 1-2: isothermal absorption of water vapor from room air;
- Process 2-3: constant concentration heating of the absorbent;
- Process 3-4: constant vapor pressure regeneration of absorbent and
- Process 4-1: constant concentration cooling of absorbent.

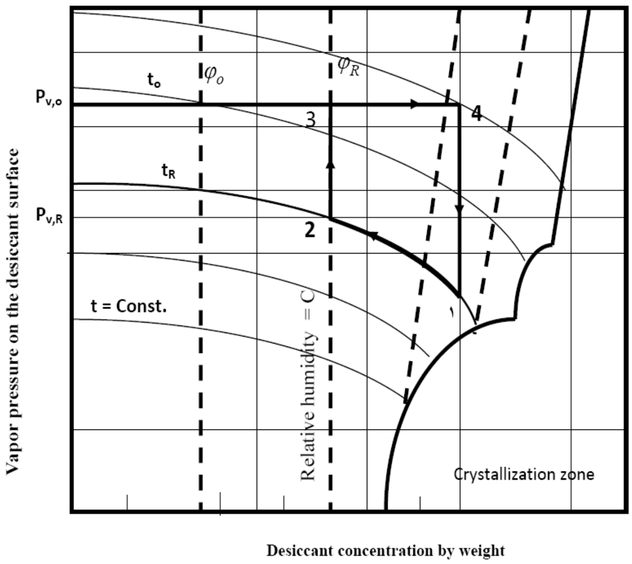

Fig. 2: Vapour pressure on the desiccant surface vs. desiccant concentration of the absorption-regeneration cycle

The thermal processes of this cycle are carried out between two concentration limits; x_1 and x_2 and the cycle has another operation limits which are its maximum regeneration temperature, t_4; condensation vapor pressure, $p_{v,O}$ and maximum absorption vapor pressure, $p_{v,R}$. Evaluation of these operation limits is important from the point of view of system design and construction. Therefore, description of the effect of weather conditions on the cycle operation is presented as follows: if the room temperature is equal to t_1, and strong solution concentration is x_1, absorption process starts only when the vapor pressure on the absorbent surface is lower than the vapor pressure in the room air $p_{v,R}$ (Fig. 2). Theoretically, absorption continues from 1 to 2, i.e. ends at equilibrium condition when the pressure of vapor on the absorbent surface is the same as that in room air. When the absorption process ends, absorbent is pumped to the regenerator and heated from an external source. Regeneration of weak absorbent can be carried out at constant pressure. The vapor pressure on the absorbent surface at point 2 is equal to $p_{v,R}$ which is determined in terms of room relative humidity and temperature. Constant pressure condensation at this pressure requires that the condensation temperature is the saturation temperature of water vapor corresponding to the vapor pressure of the outdoor air; $p_{v,O}$. When condensation is assumed to be at ambient temperature, weak solution must be heated from t_2 to t_3 where as concentration is constant and vapor pressure increases from $p_{v,R}$ to $p_{v,O}$, which is the saturation pressure of vapor corresponding to ambient (outdoor) conditions. The increase in temperature from t_2 to t_3 depends on the relative humidity of air or the weak solution concentration, x_2, which depends also on the relative humidity at the given ambient temperature.

During the constant pressure condensation, solution concentration increases from x_2 to x_4. The maximum regeneration temperature depends on the available heat source and the limits required of desiccant concentration. Strong (regenerated) solution at point 4 is not able to absorb vapor from room air due to its higher vapor pressure; therefore pressure is reduced again to p_1 by cooling from t_4 to the room temperature where the cycle ends at point 1.

3. Operation of the Proposed System

The configuration of the proposed system is demonstrated in Fig.3. As shown in figure it can be observed that the system is similar to the continuous absorption refrigeration system, and the absorber and regenerator in the present system deals with flowing air through rotating wheels. The lower section of the absorption rotating bed is immersed in the strong solution basin which functions as an absorber, where as the upper section of this bed, is subjected to the humid air at room condition to absorb water vapor from the room humid air. The absorption bed, which contains a porous material, rotates around its central axis and carries the solution from the lower section in the liquid side to the upper section in the air duct side. The weak solution resulting from the absorption process flows from the absorption basin to the regeneration basin, which is kept at the outdoor conditions. In the regenerator, the regeneration bed rotates and carries the solution from the regenerator to be heated and re-concentrated by the hot air through the air duct. Solution pump circulates the solution between the absorber and regenerator.

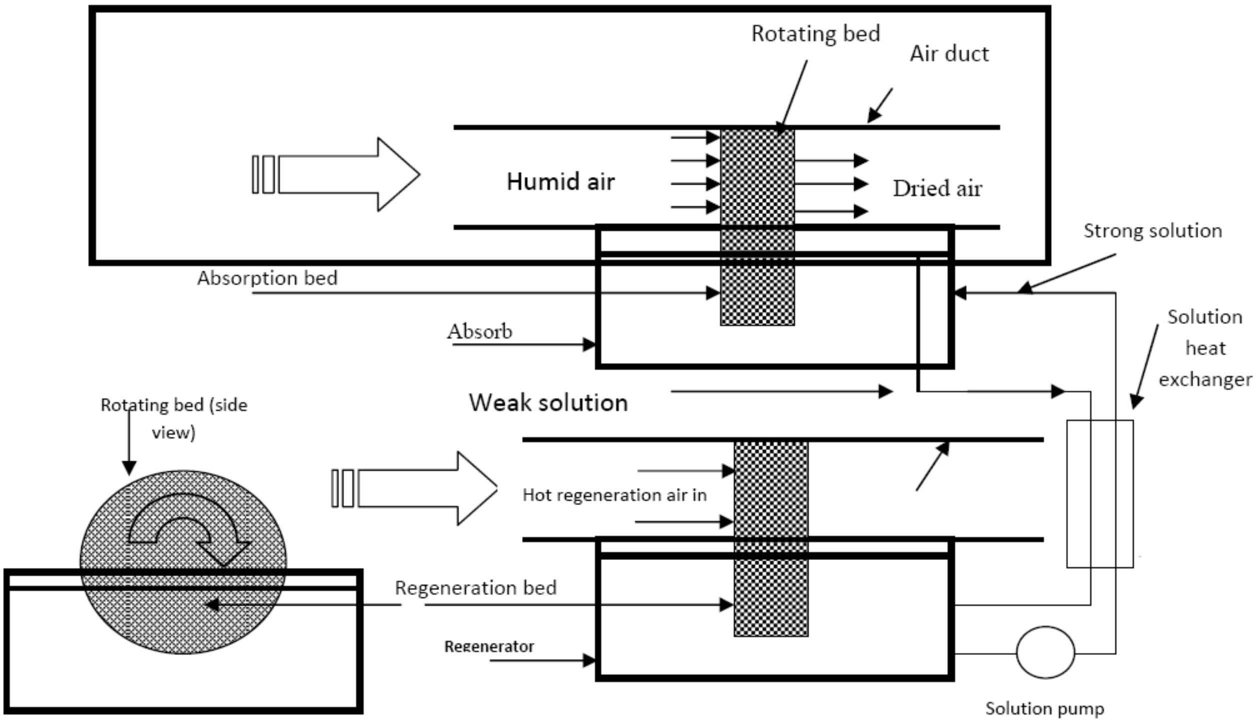

Fig. 3 The configuration of the proposed system

4. System parametric analysis

Because of the abundance of parameters involved in the system operation, some sort of systematic approach must be employed in order to evaluate the effect of various factors. Some parameters such as ambient air temperature and humidity are uncontrollable and determine an operation limits for the system. Other system parameters, such as initial concentration of desiccant, concentration at the end of absorption, and design characteristics of the desiccant bed can be selected for optimum system performance.

An important aspect of any system simulation is the determination of the conditions for optimized performance. When sorption / desorption cycle is applied with natural absorption of water vapor from air, system efficiency can be simply evaluated from the multiplication of the cycle efficiency by the efficiency of the heat transfer equipment (ex. collector efficiency, when solar energy is applied). In case of forced air absorption, when rotating bed is applied, energy added to forced air stream must be accounted in definition of system efficiency. Therefore, system efficiency is defined as the ratio of heat added to generated water vapor q_v to the total equivalent heat added to the system H, i.e

$$\eta_{sys} = q_v / H \tag{1}$$

The heat added to generated water vapor is assumed equal to the latent load removed by the system from the dehumidified space. Total heat added to the system can be expressed as:

$$Q_t = Q_b + Q_d \tag{2}$$

where Q_b is the heat added to bed material and Q_d is the heat added to desiccant carried by the wheel and heat required to generate vapor,

$$Q_b = M_b Cp_b (T_r - T_a) \tag{3}$$

where M_b and Cp_b are the mass and specific heat of bed material; respectively,

T_r and T_a are the bed temperatures at the end of regeneration and end of adsorption; respectively.

During regeneration, heat must be applied to the desiccant to accomplish the following:

1. heat the solution to minimum regeneration temperature (process 2-3);
2. vaporize the liquid water;
3. heat the solution to its final temperature in the (process 3-4) and
4. heat the regenerated vapor to its final temperature.

The heat added to desiccant and water vapor during the desorption process can be expressed as given in [15] as

$$Q_d = [\ m h_v + M_d h_r - (M_d + m) h_a\ ] \tag{4}$$

where h is the enthalpy. Subscripts v, r and a denote water vapor, desiccant condition at the end of regeneration and desiccant condition at the end of adsorption, respectively.

Knowing desiccant concentration limits and mass of desiccant M_d one can evaluate the mass of adsorbed (or regenerated) water vapor as follows,

$$m = \left(\frac{X_r - X_a}{X_a} \right) M_d \tag{5}$$

where X_r and X_a are desiccant concentration at the end of regeneration and end of adsorption, respectively. As the mass of desiccant changes during the processes, in eq.(5) M_d is taken at start of adsorption. The heat equivalent to energy added to air stream, blown through the desiccant wheel, can be evaluated by dividing the mechanical work by the Carnot energy factor,

$$Q_e = \frac{1}{Ca}(v_o A)\Delta P(\tau_a) \tag{6}$$

where v_o is the air velocity at bed entrance, A is the area of wheel in contact with air, ΔP is the pressure drop through the wheel, τ_a is the adsorption time and Ca is the Carnot energy factor. This factor is dependent on the operating temperature limits of the cycle. As the temperature limits of the operating cycle are the room temperature, which is equal to the adsorption temperature T_a and maximum regeneration temperature, which is equal to the desiccant temperature at the end of regeneration process T_r. Consequently, Carnot energy factor can be expressed as

$$Ca = \left(\frac{T_r - T_a}{T_r} \right) \tag{7}$$

The total equivalent heat added to the system is expressed as

$$H = Q_b + Q_d + Q_e \tag{8}$$

Room conditions (temperature and relative humidity) are indoor controlling parameters, which determine the maximum possible mass of vapor adsorbed by the desiccant. In other words, for specific sorbent the lower value of desiccant concentration at the end of sorption process is dependent on indoor parameters. When Calcium Chloride is the working desiccant, the lower concentration can be expressed as given in [15] by

$$X_{min} = \left[\ln p_v - \left(a_o - \frac{b_o}{t + 111.9} \right) \right] \Big/ \left(a_1 - \frac{b_1}{t + 111.9} \right) \tag{9}$$

where, t is the sorbent temperature, ^oC, a_o, b_o, a_1, b_1 are regression constants given in [15]. The vapor pressure in room air is dependent on the relative humidity φ and saturation pressure of water, $p_s(t)$, i.e

$$p_v = \varphi p_{s(t)} \tag{10}$$

Cycle efficiency is defined as the ratio of heat added to generate vapor to the heat added to desiccant, during regeneration process, i.e.,

$$\eta_{cyc} = q_v / Q_d \tag{11}$$

where

$$q_v = m L \tag{12}$$

In terms of desiccant parameters, cycle efficiency can be expressed as given in [11],

$$\eta_{cyc} = L \Big/ \left[h_v + \frac{X_a h_r - X_r h_a}{X_r - X_a} \right] \tag{13}$$

where L is the latent heat of evaporation of water at the condensation pressure, h_v is the enthalpy of generated vapor, x and h are the concentration and enthalpy of desiccant, respectively. Subscripts a and r denote the conditions at the end of absorption and regeneration, respectively.

Referring to Eq.(8) and substituting by Q_d, Q_b, and Q_a from equations (3), (4) and (6), respectively [16], yields

$$H = \left[m h_v + M_d h_r - (M_d + m) h_a + M_b Cp_b (T_r - T_a) + \frac{(v_o A \Delta P \tau_a)}{Ca} \right] \tag{14}$$

Substituting Eqs. (11), (12), (13) and (14) in Eq. (1), system efficiency can be expressed as

$$\eta_{sys} = 1 \Big/ \left[\frac{1}{\eta_{cyc}} + \frac{M_b Cp_b (T_r - T_a)}{mL} + \frac{v_o A \Delta P \tau_a}{mLCa} \right] \tag{15}$$

Substituting m from Eq. (5) in Eq. (15), yields

$$\eta_{sys} = 1 \Big/ \left[\frac{1}{\eta_{cyc}} + \frac{M_b Cp_b (T_r - T_a)}{M_d [(X_r - X_a)/X_a] L} + \frac{v_o A \Delta P \tau_a}{L M_d Ca [(X_r - X_a)/X_a]} \right] \tag{16}$$

To evaluate the effect of system parameters and ambient conditions on the system efficiency η_{sys}, In Eq.(16) the pressure drop ΔP and adsorption time τ_a must be expressed in terms of system operating parameters.

The pressure drop ΔP suffered by a fluid in flowing through a bed of packed solids such as spheres, cylinders, etc. is dependent on bed length Z, porosity ε and reasonably well correlated by the Ergun equation [17]

$$\frac{\Delta P}{Z} \frac{\varepsilon^3 d_p \rho_g}{(1-\varepsilon) G^2} = \frac{150(1-\varepsilon)}{Re} + 1.75 \tag{17}$$

where

$$Re = \frac{d_p G}{\mu} \tag{18}$$

and d_p is the effective diameter of the particles of the desiccant wheel and G is the mass velocity of air stream.

According to the analysis of the adsorption process through porous bed impregnated with liquid desiccant [12], the dimensionless adsorption time ψ is dependent on the mass transfer potential ratio and can be expressed in the following dimensionless form,

$$\Psi = \ln\left(\frac{C - C_o^*}{C - C^*} \right) \tag{19}$$

where C and C^* are the molar concentration of water vapor in room air and concentration at equilibrium with adsorbed phase, the subscript o denotes the initial condition of desiccant. The dimensionless time ψ is defined in terms of the mass transfer coefficient k, total interfacial area per unit volume a, adsorption time τ_a, bed height Z, bed porosity ε and air stream velocity v_o, which is given in [12] as:

$$\Psi = \frac{ka}{K(1-\varepsilon)}\left(\tau_a - \frac{Z\varepsilon}{v_o}\right) \tag{20}$$

where K is the an affinity constant for sorption equilibrium. From equations (19) and (20) it can be noticed that, for specific desiccant characteristics and ambient conditions, the adsorption time is dependent on the solute concentration in the absorber.

However, it is more convenient to express the system efficiency in terms of non-dimensional parameters. This expression can be obtained by substituting equations (19) and (20) in Eq. (16) and rearranging,

$$\eta_{sys} = 1 \Big/ \left(\frac{1}{\eta_{cyc}}\right) + \bar{X}\,\bar{M}\left[\bar{C}p + \frac{\bar{P}}{Ca}\left(1 + \bar{V}\Psi\right)\right] \tag{21}$$

The dimensionless parameters presented in Eq. (21) can be defined in two groups. The first group represents the indoor conditions and operating concentration limits of the applied desiccant. This group includes the cycle efficiency η_{cyc}, the dimensionless concentration ratio $\bar{X}$, the Carnot factor Ca and the dimensionless time ψ. The second group of parameters, which represents the bed and air stream conditions, includes the mass ratio $\bar{M}$, dimensionless specific heat of bed material, $\bar{C}p$, dimensionless pressure drop $\bar{P}$ and dimensionless velocity $\bar{V}$.

The dimensionless parameters are defined as follows:

$$\bar{X} = \frac{X_a}{X_r - X_a} \tag{22}$$

$$\bar{M} = \frac{M_b}{M_d} \tag{23}$$

$$\bar{C}p = \frac{Cp_b(T_r - T_a)}{L} \tag{24}$$

$$\bar{P} = \frac{\varepsilon \Delta P}{L \rho_b} \tag{25}$$

$$\bar{V} = \left(\frac{K v_o}{kaZ}\right)\left(\frac{1-\varepsilon}{\varepsilon}\right) \tag{26}$$

where ρ_b is the bed average density, defined as

$$\rho_b = \frac{M_b}{AZ} \tag{27}$$

5. Results and Discussion

An important factor in the operation of the humidity pump is the regeneration temperature required to re-concentrate the solution in the regenerator. This temperature determines the type of the heating source which can be applied in the regeneration of the system. The minimum limit of the regeneration temperature for the applied desiccant, at indoor temperature of 20 C, is plotted in Fig 4 at different values of room relative humidity, and outdoor temperatures versus the outdoor relative humidity. As shown in the figure, it can be observed that the minimum regeneration temperature increases with decrease in the room relative humidity, at constant values of ambient parameters (temperature and humidity). This can be explained by the need to a solution with higher concentration to absorb moisture from air with lower values of relative humidity at the same ambient temperature. In general, it can be stated that, the temperature potential required to pump humidity from the room condition to the ambient air is directly proportional with the humidity potential between the room and the ambient air. On the other hand, an increase in the outdoor temperature increases the required regeneration temperature for constant values of room and outdoor humidity.

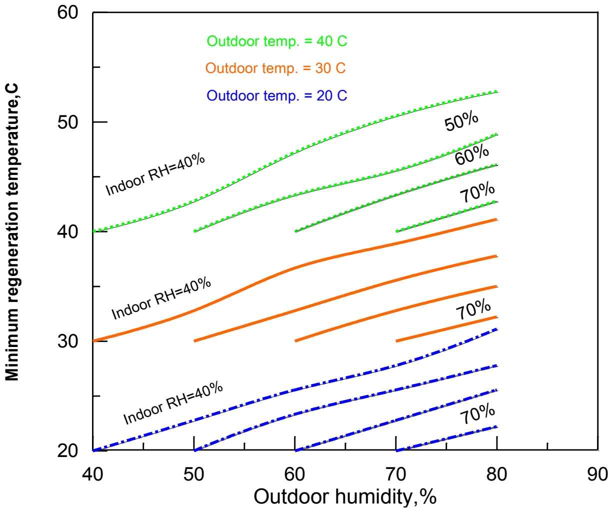

Fig. 4: Minimum regeneration temperature versus outdoor relative humidity at different values of indoor humidity and outdoor temperatures

From parametric analysis, Fig. 5 illustrates the effect of desiccant concentration at the end of sorption on the thermal efficiency of the adsorption/desorption cycle and the dimensionless adsorption time. It can be observed that the cycle efficiency has its higher value at lower adsorption concentration. On the other hand, adsorption time exponentially increases with decrease in sorbent concentration, which means an exponential increase in the energy, added to air stream flowing through the bed.

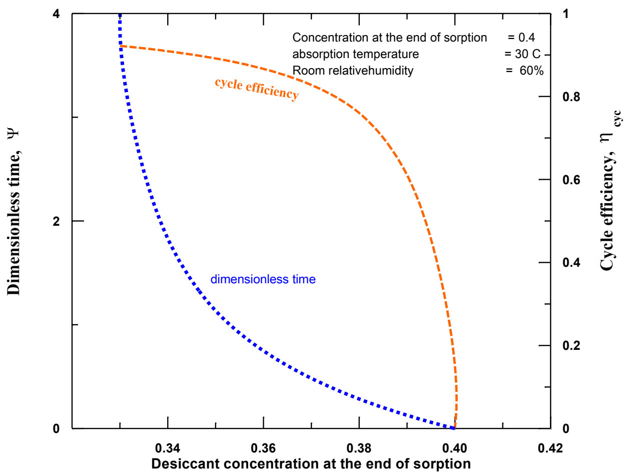

Fig. 5 Cycle efficiency and time versus desiccant concentration at the end of sorption

System efficiency is plotted versus desiccant concentration during sorption (Fig. 6). It is shown that a maximum value of system efficiency exists at specific value of concentration, for a given value of room relative humidity. In addition, system efficiency generally increases with increase in humidity of indoor air. Therefore, desiccant minimum concentration must be selected to maximize the system performance.

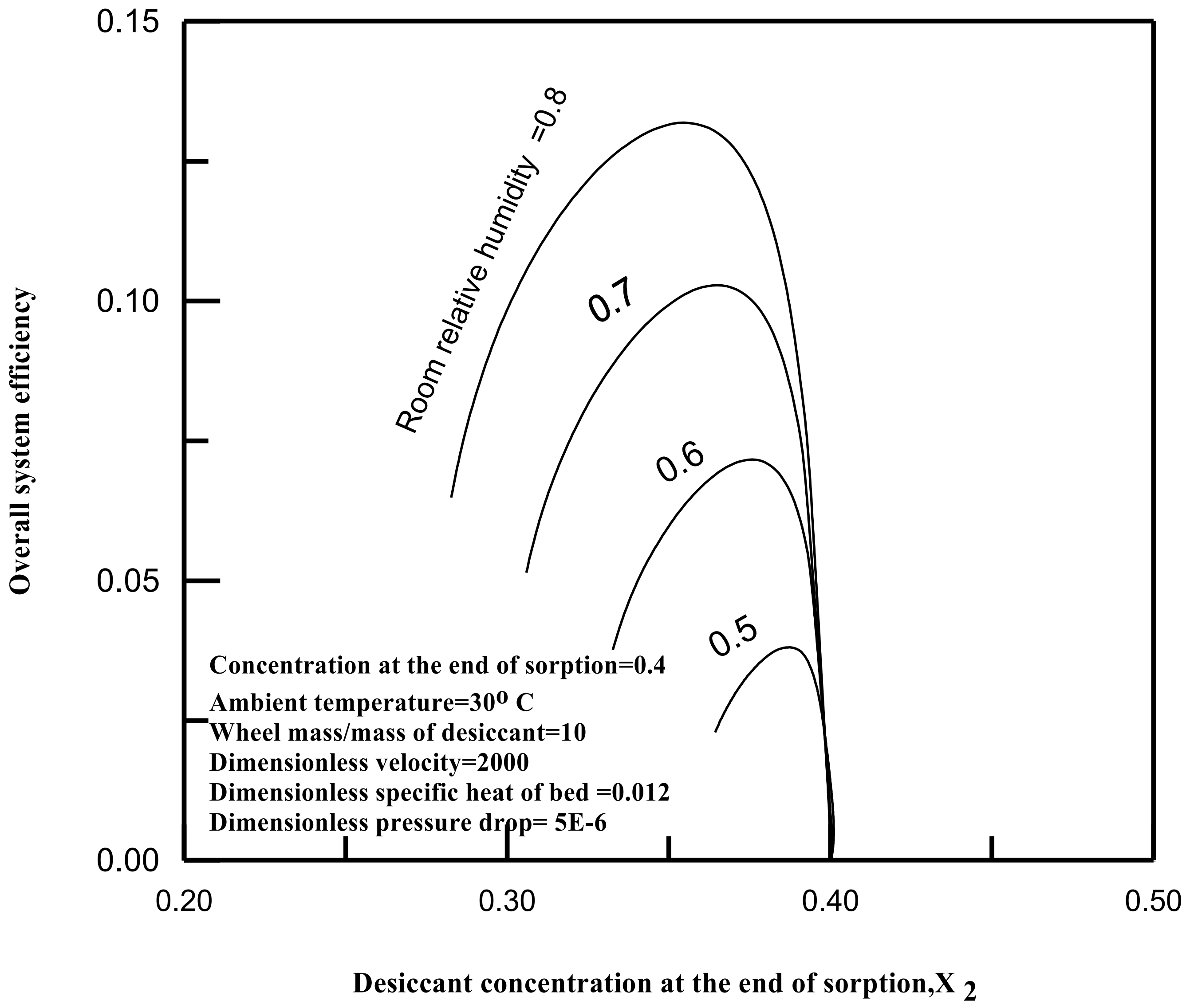

Fig. 6: Overall system efficiency versus desiccant concentration at the end of sorption

6. Conclusion

Novel design configuration of desiccant operated humidity pump has presented and analyzed. The effects of meteorological conditions and system design parameters are well defined. Also, system efficiency is defined in terms of operating cycle efficiency and system design parameters. Effect of indoor and outdoor parameters on the required regeneration temperature has been highlighted. The appropriate selection of desiccant concentration at the end of sorption has been discussed. Based on the obtained simulation results, the following conclusions can be drawn:

- Desiccant minimum regeneration temperature is proportional to the humidity potential between the indoor and outdoor conditions (temperature and humidity).
- For a given ambient parameters, system overall efficiency is highly dependent on relative humidity of the indoor air
- Cycle efficiency and system efficiency don't reach maxima at the same value of desiccant concentration at the end of adsorption.

Nomenclature

a	interfacial area per unit volume, m^2/m^3
A	cross sectional area of packed column, m^2
C	molar concentration of water vapor in air, mole/m^3
C*	water vapor molar concentration in air, at equilibrium with desiccant, mole/m^3
Ca	Carnot factor, dimensionless
Cp	specific heat, J/kg.°C
$\overline{C}p$	dimensionless specific heat
d	diameter, m
G	air mass velocity, kg/sm^2
h	enthalpy, J/kg
H	total equivalent heat, J
k	mass transfer coefficient, $m^{-2}s^{-1}$
K	affinity constant for adsorption equilibrium

L latent heat of evaporation of water, J/kg
m mass of evaporated water during regeneration, kg
M mass of desiccant or bed
$\overline{M}$ mass ratio
p vapor pressure, mmHg
$\overline{P}$ dimensionless pressure drop
q heat added to vapor, J
Q heat added, J
t ambient temperature, ^{o}C
T desiccant temperature, K
v air stream velocity, m/s
$\overline{V}$ dimensionless velocity
X desiccant mass concentration
$\overline{X}$ concentration ratio
Z height of desiccant bed, m

Greek symbols

Δ pressure drop
ε porosity
η efficiency
φ relative humidity
μ dynamic viscosity
ρ density
τ time
Ψ dimensionless time

Subscripts

a condition at the end of adsorption
b bed
d desiccant
cyc cycle
e equivalent
o initial, inlet
r condition at the end of regeneration
s saturation condition
sys system
t total
v vapour

References

[1] Paradip M (1998). "Heat and mass transfer in composite desiccant pore structures for dehumidification", Solar Energy Eng., 62: 1-10.

[2] Elsayed MM, Chamkha AJ (1997), "Analysis and performance of radial flow rotary desiccant dehumidifiers", Solar Energy Eng., 119: 35-43. doi:10.1115/1.2871825

[3] Hamed A M (2005), "Experimental investigation on the adsorption/desorption process using solid desiccant in an inclined-fluidized bed", Renewable Energy, pp. 1-9. doi:10.1016/j.renene.2005.01.001

[4] Zhang HF, Dai YJ, Wang RZ (2001). "Parameter analysis to improve rotary desiccant dehumidification using a mathematical model", Int. J. Thermal Sci., 40: 400–408. doi:10.1016/S1290-0729(01)01224-8

[5] K. Daou, R.Z. Wang, Z.Z. Xia, Desiccant cooling air conditioning: a review, Renewable and sustainable reviews 10 (2006) 55–77.

[6] W. Ramadan, Theoretical and experimental study on the performance of a fluidized air dryer, M.Sc. thesis, Mechanical Engineering Department, Mansoura University, 2005.

[7] Awad, M.M., Ramzy, A.K., Hamed, A.M. and Bekheit, M.M. (2008). "Theoretical and Experimental Investigation on the Radial Flow Desiccant Dehumidification Bed," Applied Thermal Engineering, 28, 75-85. doi:10.1016/j.applthermaleng.2006.12.018

[8] Hamed. A.M., Abd-Elrahman, W.R. and El-Emam, S.H. (2010). "Experimental study of the transient adsorption/desorption characteristics of silica gel particles in fluidized bed," Energy, 35, 6, 2468-2483. doi:10.1016/j.energy.2010.02.042

[9] H.F. Zhang, Y.J. Dai, R.Z. Wang, Parameter analysis to improve rotary desiccant dehumidification using a mathematical model, International Journal of Thermal Sciences 40 (2001) 400–408. doi:10.1016/S1290-0729(01)01224-8

[10] J.L. Niu, L.Z. Zhang, Effects of wall thickness on the heat and moisture transfer in desiccant wheels for air dehumidification and enthalpy recovery, International Communications in Heat and Mass Transfer 29 (2) (2002) 255–268. doi:10.1016/S0735-1933(02)00316-0

[11] M.M. Elsayed, A.J. Chamkha, Analysis and performance of radial flow rotary desiccant dehumidifiers, Solar Energy Engineering 119 (1997) 35–43. doi:10.1115/1.2871825

[12] Hamed AM. Theoretical and experimental study on the transient adsorption characteristics of a vertical packed porous bed. Renewable Energy 2002;27:525–41. doi:10.1016/S0960-1481(01)00112-4

[13] Hamed AM. Desorption characteristics of desiccant bed for solar dehumidification/humidification air conditioning systems. Renewable Energy 2003;28:2099–111.

[14] Hamed AM. Experimental investigation on the natural absorption on the surface of sandy layer impregnated with liquid desiccant. Renewable Energy 2003;28:1569–87. doi:10.1016/S0960-1481(03)00005-3

[15] Hamed A.M., "Absorption-regeneration Cycle for Production of Water from Air-Theoretical Approach", Renewable Energy, No. 19, pp. 625-635, 2000. doi:10.1016/S0960-1481(99)00068-3

[16] Hamed AM. Parametric study of the adsorption–desorption system producing water from ambient air. International Journal of Renewable Energy Engineering 2000; 2(3): 244–52.

[17] Treybal R.E., “Mass Transfer Operations”, McGraw-Hill, Third Edition, 1981