International Journal of the Physical Sciences Vol. 6(12), pp. 2852-2860, 18 June, 2011
Available online at http://www.academicjournals.org/IJPS
DOI: 10.5897/IJPS11.400
ISSN 1992 - 1950 ©2011 Academic Journals

Full Length Research Paper

An experimental and modeling study of a dehumidification tower

Hesamoddin Salarian, Hossein Ghadamian, Morteza Khalaji Assadi and Abtin Ataei*

Department of Energy Engineering, Graduate School of Environment and Energy, Science and Research Branch, Islamic Azad University, Tehran, Iran.

Accepted 16 May, 2011

In this paper, the size and performance of a dehumidification tower were studied by simulating varying operating conditions. Based on the experimental results, this study presents the performance of a packed tower absorber for a lithium chloride desiccant dehumidification system. The effects of the main variables - airflow rate, liquid desiccant flow rate, and inlet air temperature on the rate of dehumidification were reported. It was found that the influence of these variables could be assumed to be linear. A finite difference model was developed to determine the packing height of the dehumidification towers. This model was worked out in MATLAB code, which is a suitable model for measuring the optimum height of a tower. The validity of this model was compared with published experimental data and our data. Comparisons between the simulated packing height and the actual packing heights used by experimental studies illustrated that our finite difference model is acceptable. With this model, we predict the packing height for every condition, and then we constructed the dehumidifier based on the results of our finite difference model.

Key words: Dehumidification, finite difference, liquid desiccant, measurement device, packing height.

INTRODUCTION

Each year, massive amounts of fossil fuels are used for moisture removal in industrial and agricultural processes. The control of humidity is essential for maintaining healthy, productive and comfortable conditions (Aktacir, 2011; ASHRAE, 2008; Bozdemir, 2010, 2011; Roth et al., 2002; Yoo et al., 2010). The vapor compression cooling system (VCS) is the most common method of providing refrigeration and air conditioning (Ibrahim et al., 2011; Roth et al., 2002; Yoo et al., 2010). It is also one of the major causes of ozone depletion and uses a considerable amount of electricity. The application of desiccant air conditioning systems is proposed as an alternative solution that reduces energy consumption and greenhouse gas emissions in hot and humid locations (Ramzy et al., 2010). The main components of a desiccant cooling system are the absorber or dehumidifier, the regenerator, and the cooling unit (Figure 1).

An experimental liquid desiccant cooling system is represented in Figure 2. Moist air enters the bottom of the dehumidification tower and travels up through the packing material. The cool, strong liquid desiccant enters the top of the tower and travels down the packing materials' counter-current to the airflow; since the cool, strong desiccant vapor pressure is less than the moist air vapor pressure, the water vapor will be transferred from the air to the liquid desiccant, and then the dehumidified air leaves the absorber tower. The performance of liquid desiccant cooling systems depend on thermodynamic variables such as air and desiccant flow rate, air temperature and humidity, desiccant concentration, and temperature (Bozdemir, 2010, 2011).

As to cooling, the load in air-conditioning systems can be divided into the sensible and the latent load (Ibrahim et al., 2011). Energy can be saved through eliminating the latent load. In traditional air conditioning, the moisture of air is removed by lowering the temperature of the moist air below its dew point. The air is then reheated to the desired comfort temperature for the air-conditioned space. In the cooling unit in Figure 1, the evaporator of a traditional air conditioner, a cold coil or an evaporative cooler, can handle the sensible load. If we can delete or reduce the latent load, we can save energy from two standpoints: energy that reduces air temperature to below its dew-point and energy used to reheat air.

*Corresponding author. Email: abtinataei@gmail.com

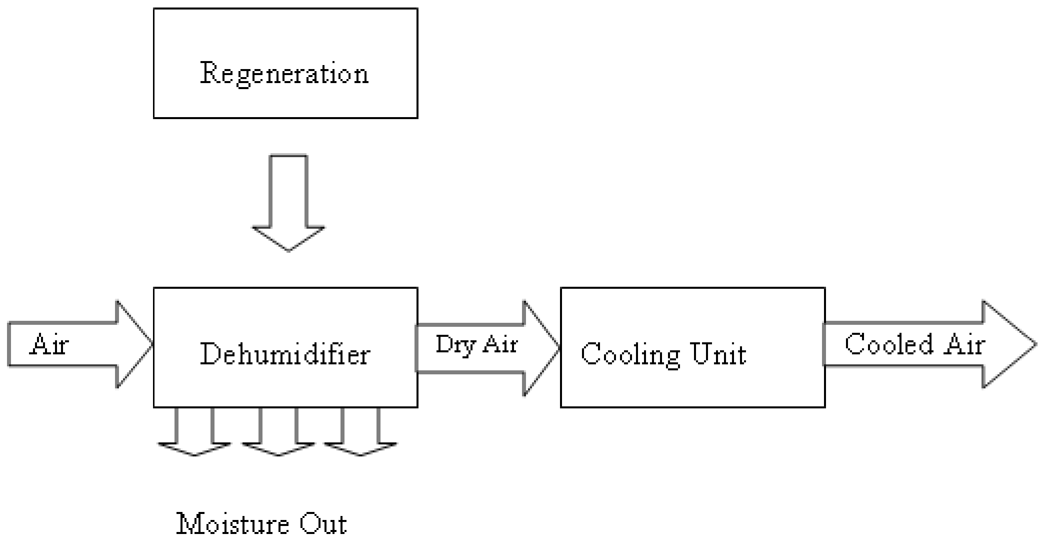

Figure 1. Main components of a desiccant cooling system

Figure 2. Main components of a packed bed liquid desiccant cooling system

If the ratio of sensible heat gain to total heat gain of the space (SHR) is low, these energies will be greatly reduced. A low SHR means that the total cooling load is predominantly the latent load. Researchers have shown that desiccant air conditioners have the potential to reduce energy consumption for cooling and dehumidification by handling the latent load independently from the sensible load and shifting the energy used away from electricity and toward renewable energy, for example, solar energy, cheaper fuels, and waste energy (Aktacir, 2011; Daou et al., 2006). Desiccant materials have a high affinity for water and can be used for absorbing water vapor; also, they can then be regenerated after becoming saturated with moisture. Desiccants are divided into liquids or solids. Examples of solid desiccants include silica gel, activated alumina, lithium chloride salt and molecular sieves. Liquid desiccants include lithium chloride, lithium bromide, calcium chloride, and triathlon glycol solutions.

Each of liquid or solid desiccant systems has its own advantages and disadvantages. Solid desiccant systems are compact and less subject to corrosion and carryover. Liquid desiccant systems have lower regeneration temperatures, lower pressure drops on the airside, and flexibility in utilization (Daou et al., 2006).

In liquid desiccant systems, the absorber and regenerator are in contact with the process air stream. It is possible that the configuration includes a finned-tube surface, a coil-type absorber, a spray tower, and a packed tower. The packing of packed towers can be regular or random (Daou et al., 2006; Mei and Dai, 2008).

The main aim of the current study is the prediction of the packing height of the absorber of a packed-bed-type liquid desiccant system using a mathematical model and then comparing the results with experimental studies.

METHODOLOGY

Finite difference model

A finite difference model was developed to determine the packing height of the dehumidification tower. This model is based on adiabatic gas absorption, which is a common assumption in other literature (Treybal, 1981; Fumo and Goswami, 2002; Oberg and Goswami, 1998; Mago and Goswami, 2003).

Figure 3 displays the control volume of a differential slice from the packed tower with significant material and heat effects entering and exiting the infinitesimal packing height. The direction of the mass and heat transfers is taken as positive from a gas to a liquid. The assumptions are: a) the packed tower is adiabatic, b) the heat of the solution is neglected, c) no resistance to heat transfer occurs in the liquid phase, d) the interfacial surface areas for heat and mass transfer are equal, and e) no axial dispersion exists. Therefore, a one-dimensional analysis is used.

The specific enthalpy of a desiccant solution and the specific enthalpy of moist air are written as:

$$h_l = c_{pl} T_l + h_{o\xi} \qquad (1)$$

$$h_a = c_{pa} T_a + \omega_a h_{vo} \qquad (2)$$

Airside heat and mass transfer equations are:

$$\dot{m}_a dh_a = h_c dz(T_l - T_a) + h_v \dot{m}_a d\omega \qquad (3)$$

The change of air humidity across the packing height is:

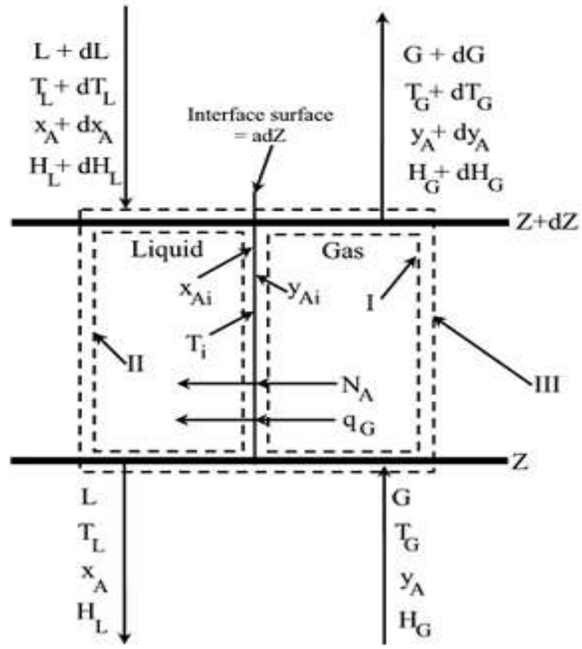

Figure 3. Differential segment from a packed tower

$$\frac{d\omega_a}{dZ} = -\frac{M_w F_G a_w}{\dot{m}_a} \ln\left(\frac{1-y_i}{1-y}\right) \qquad (4)$$

In this equation, the interfacial gas phase concentration is calculated by:

$$y_i = 1 - (1-y)\left(\frac{x}{x_i}\right)^{\frac{F_L}{F_G}} \qquad (5)$$

The change in air temperature across the differential segment can be expressed as:

$$\frac{dT_G}{dZ} = \frac{h'_G a_t (T_{G-}T_L)}{G.(C_{pa} + YC_{p,v})} \qquad (6)$$

$h'_G a_t$ is the Ackermann correction for simulated heat and mass transfer (Treybal, 1981).

The change in desiccant temperature, flow rate, and concentration across the differential segment are given by the following equations:

$$\frac{\boldsymbol{dT_l}}{\boldsymbol{dZ}} = \frac{\boldsymbol{G}}{\boldsymbol{L.c_{pL}}} \{((C_{p,v} + Y.c_{pa})\frac{dT_G}{dZ} + [C_{pv}(T_G - T_0) - C_{PL}(T_L - T_0) + \lambda_0]\frac{\boldsymbol{dY}}{\boldsymbol{dZ}}\} \qquad (7)$$

$$\frac{dL}{dZ} = G.\frac{dY}{dZ} \qquad (8)$$

$$\frac{dX}{dZ} = -\frac{G}{L}.X.\frac{dY}{dZ} \qquad (9)$$

Table 1. Outdoor and indoor conditions

Condition	WB (^{o}C)	DB (^{o}C)	RH %	W (kg/kg)
Outdoor	29	35	85	0.025
Indoor	17	24	50	0.0094
Difference	12	11	25	0.0156

LAT = 37.5, ALT = -20m, DP = 27.25°C, and DR = 5.4-8.3°C.

Table 2. Specifications of different measurement devices.

Devices	Type	Accuracy	Range
Thermometers	K-type thermocouple 0412-yk90HT	0.1°C	0-50°C
Air flow meter	Hot Wire Anemometer YK-2004AH	0.001 m^3/min	
Solution flow meter	Rotameter	10 L/min	100-1000 L/min
Humidity meter	YK-90HT	3%	10-90%

The finite difference model was coded using MATLAB software. The parameters required as the input data include air humidity ratio at the inlet and outlet of tower, air mass velocity, liquid desiccant
liquid desiccant, and nominal size of the packing. The inlet conditions of the desiccant are known, and so the outlet desiccant solution conditions are solved for the boundary conditions across the entire tower. To determine the height of the tower, starting with the first segment, the air and liquid states for a segment are determined. If the comparison between the calculated humidity ratio at the top of the segment and the known humidity ratio at the top of the tower show that the humidity ratio at the top of the tower has not been reached, then another segment is added until required exit humidity ratio is reached. By this method, we can calculate the optimum packing height for the tower.

Experimental apparatus and case study area

An experimental apparatus was designed for studies on a packed bed liquid desiccant absorber and regenerator based on the results of the finite difference model. The apparatus was set up in the moderate and humid climate of Iran in City of Noshahr near the Caspian Sea. In this location, the average high temperature in summer is about 35°C and the relative humidity is 80-85%. The outdoor design conditions for that city are shown in Table 1.

In this area, the latent load plays a major role. Figure 2 is a photograph of the liquid desiccant system that was manufactured in this location. The system was designed for a maximum inlet air temperature of 35°C and relative humidity of 80%. For these conditions, for the finite difference model, the packing height of the dehumidifier was 85 cm (size might vary for other conditions). The packing material was polypropylene with a specific surface area of 125 m^2/m^3.

As shown in Figure 2, the absorber and regenerator towers have 0.09 m^2 of cross-sectional area (30 cm × 30 cm).

A circulation pump was used to circulate the lithium chloride. Unused desiccant was stored in a tank, and its temperature was adjusted by a heat exchange with the environment. Axial extract fans were fixed on the top of the towers to extract the outdoor air from the bottom of the towers and to provide a counter flow to the lithium chloride in the absorber and the regenerator. The desiccant was distributed by a sprayer at the top of the tower.

The following instruments were used to measure different variables:

a) A portable digital hot-wire anemometer was used to measure the air velocity and airflow rate.
b) A portable digital humidity meter, with a range of 10 to 95% R.H and a resolution of 0.01%, was used to measure the relative humidity (RH).
c) The temperature of the air and the liquid desiccant were measured by a K-type thermocouple. This instrument measures the air-dry bulb, dew point, and desiccant temperatures. By using the wet bulb and dry bulb temperatures of the air, the humidity of the air can be obtained.
d) Rotameters, with a range of 100 to 1000 (L/min), were used to measure the desiccant flow rate.
e) A refractometer was used to measure the desiccant concentration.

The main features of the different measurement devices are shown in Table 2.

RESULTS AND DISCUSSION

The packing height

The packing height was predicted by the finite difference method. A comparison between the simulated packing height and the actual packing height from Fumo and Goswami (2002) is presented in Table 3, while the comparison between the simulated packing height and actual packing height from Mago and Goswami (2003) are displayed in Table 4. In Tables 3 and 4, the packing height for all of the experimental data is constant and equals to 60 cm. The desiccant solution used in these studies was LiCl, and the packing used was 2.54 cm (1 in) polypropylene Rauschert Hiflow® rings with a specific surface area of 210 m^2/m^3.

Tables 3 and 4 illustrate that the finite difference model underestimates the packing height for the majority of the experimental runs. For example, the packing height for the finite difference model in Table 4 is 57 and 58 cm, but in the experimental study the packing height selected was 60 cm for every experiment. Evidently, values for height are very similar in these two works. Therefore, from an engineering point of view, this model and

Table 3. Comparison of the finite difference model with the experimental data, according to Fumo and Goswami (2002).

Inputs	**1**	**2**	**3**	**4**	**5**
Humidity ratio(kg_w/kg_a)	0.0180	0.0181	0.0215	0.0181	0.0181
Gas mass velocity [kg/(s-m^2)]	0.0890	1.513	1.187	1.180	1.176
Inlet gas temperature (°C)	30.1	30.2	29.9	30.1	30.0
Desiccant concentration(kg_{licl}/kg_{sol})	0.346	0.343	0.339	0.347	0.348
Desiccant mass velocity[kg/(s-m^2)]	6.124	6.113	6.272	6.227	6.206
Moisture to remove (%)	42.22	40.33	44.19	40.33	40.88
Inlet desiccant temperature(°C)	30.1	30.0	30.3	30.3	30.2
Packing height from finite difference model (cm)	59	61	58	57	57
Difference of packing height from finite difference model and experimental (cm)	-1	+1	-2	-3	-3

Table 4. Comparison of the finite difference model with experimental data, according to Mago and Goswami (2003).

Inputs	**1**	**2**	**3**
Humidity ratio(kg_w/kg_a)	0.0111	0.0111	0.0111
Gas mass velocity [kg/(s-m^2)]	2.436	2.639	2.842
Inlet gas temperature(°C)	26	26	26
Desiccant concentration(kg_{licl}/kg_{sol})	0.35	0.35	0.35
Desiccant mass velocity[kg/(s-m^2)]	2.084	2.084	2.084
Moisture to remove (%)	18.02	19.82	18.92
Inlet desiccant temperature(°C)	27	27	27
Packing height from finite difference model (cm)	57	57	58
Difference of packing height from finite difference model and experimental(cm)	-3	-3	-2

computer programming results are within the acceptable range. The main objective of the dehumidification tower is to remove moisture from the air. In Figures 4 and 5, the percentage of moisture removed is defined as following:

$$\frac{\text{inlet air humidity ratio} - \text{outlet air humidityratio}}{\text{inlet air humidity ratio}} \times 100$$

Figures 4 and 5 show that by increasing the desired amount of moisture removed, the tower height will increase. The reason is that by increasing the required moisture-removal percentage, the mass transfer area increases. In other words, the height of the tower is related to the amount of moisture that is removed from the inlet air. In Table 3, for L/G=6 and moisture removed of 40.88%, the packing height is 57 cm, but in Figure 4 for L/G=3 and with the same percent moisture removed, the packing height is 50 cm. These results show that the L/G ratio considered (6/1.5) is quite large and, according to some references, a ratio from 3 to 6 is ineffective for moisture removal and could cause an air-side pressure drop.

Since some of the references (Gommed and Grossman, 2007; Tu et al, 2009) noted the optimum range for L/G as 0.15-3, we selected a value of 3.0 for this parameter.

In this study, the packing material we selected was polypropylene with a specific surface area of 125 m^2/m^3. The finite difference model used to predict the packing height of dehumidification in a moderate climate, the condition for this location is shown in Table 1.

In Figure 5, the liquid solution mass flow rate to airflow rate ratio was selected as 0.96-1; since Chen et al. (2006) reported that the optimum air mass flow rate to solution mass flow rate ratio was between 0-1, this ratio is optimum, and the selected packing height is the optimum height. Based on these results, we determined that the packing height for the experimental system was 85 cm.

Effect of variables on dehumidification rate

The dehumidification rate can be calculated using the following equation:

$$m_e = m_a(\omega_{ai} - \omega_{ao}) \quad (10)$$

Figure 6 shows the effects of inlet air temperature on the dehumidification rate. In these experimental tests, the inlet air humidity varied from 50 to 60%, the inlet air temperature varied from 22 to 27°C, the desiccant

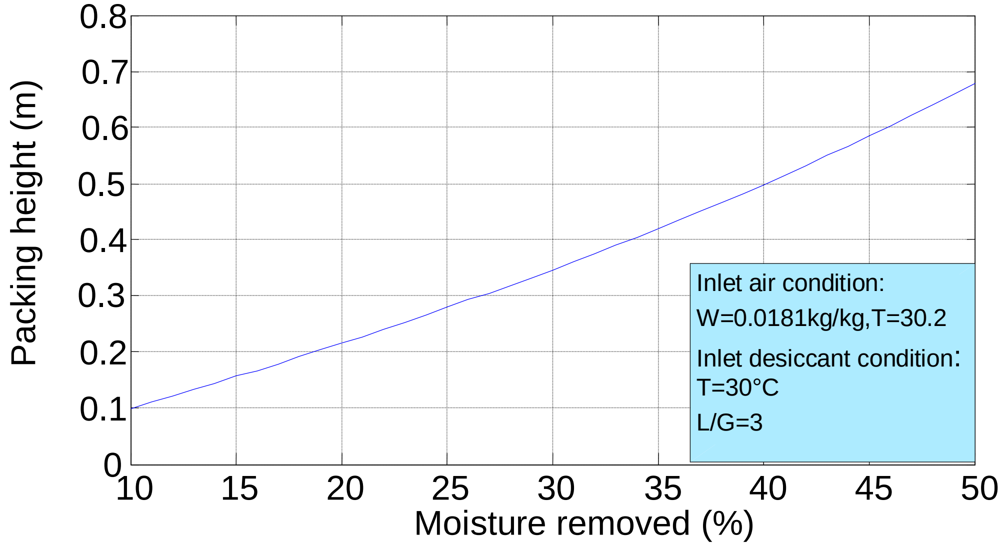

Figure 4. Packing height as a function of percentage of moisture removed.

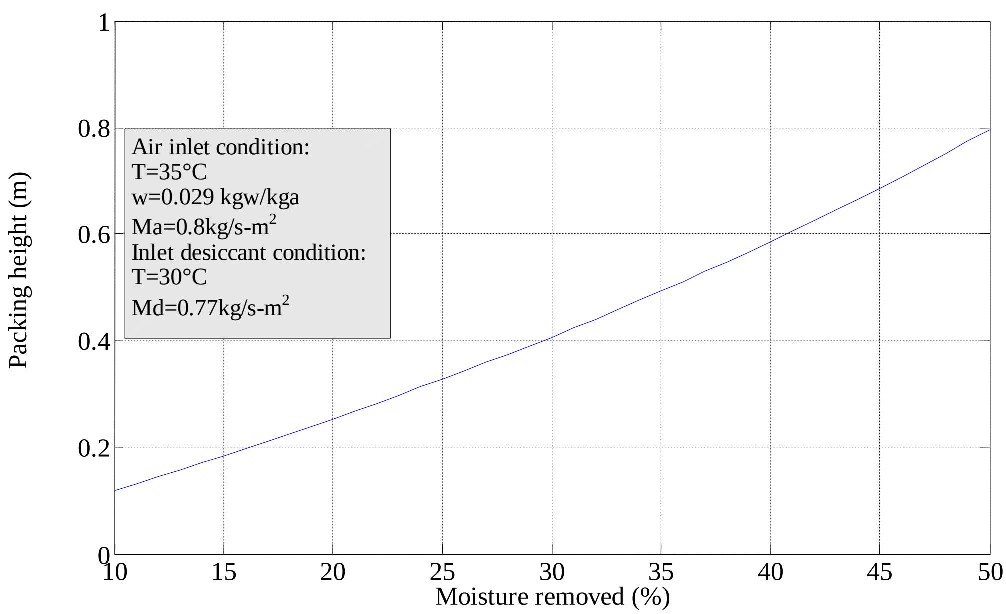

Figure 5. Packing height as a function of percentage of moisture removed

temperature was in the range from 20 to 25°C, and the airflow rate was 30 m^3/min.

Figure 6 shows that the dehumidification rate was decreased by increasing the inlet air temperature. This happens because, by increasing the air temperature, the desiccant temperature increases and thus reduces the dehumidification potential of the desiccant solution. This result is in agreement with the other experimental studies' results (Yin et al., 2007; Fumo and Goswami, 2002; Mago and Goswami, 2003; Moon et al., 2009).

As shown in Figure 7, the dehumidification rate increased with the increase of the airflow rate and was directly proportional to it. The higher airflow rate increases not only the mass transfer coefficient between the desiccant solution and the air, but it reduces the contact time and effectiveness. These findings of the present study agree with previous studies reported by Fumo and Goswami (2002), Gandhidsan (2004) and Moon et al. (2009).

The effect of liquid desiccant mass flow rate on the moisture removal is shown in Figure 8. The moisture

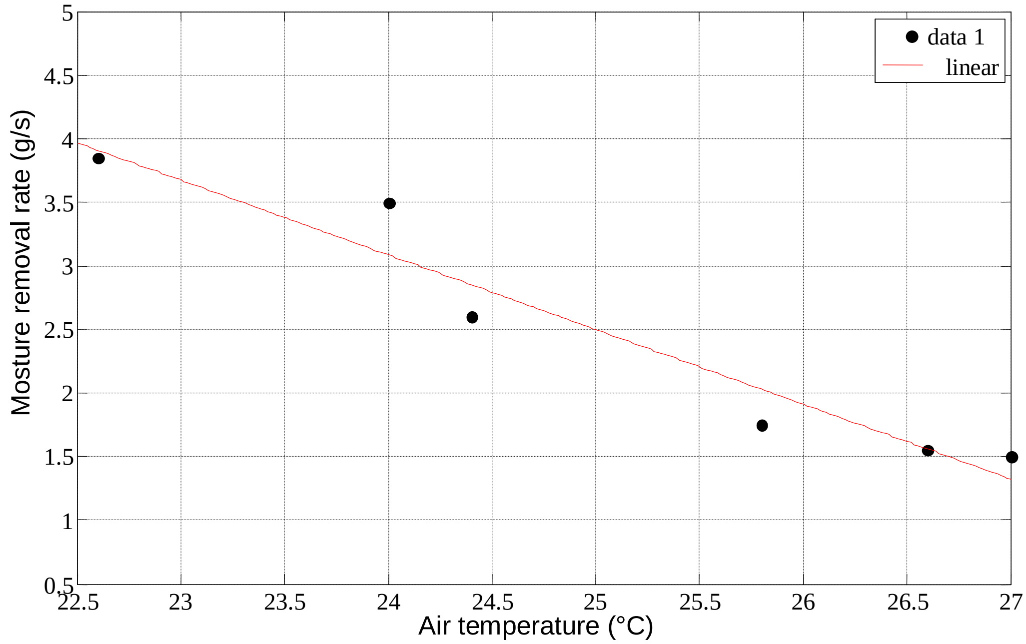

Figure 6. Effect of input air temperature on dehumidification

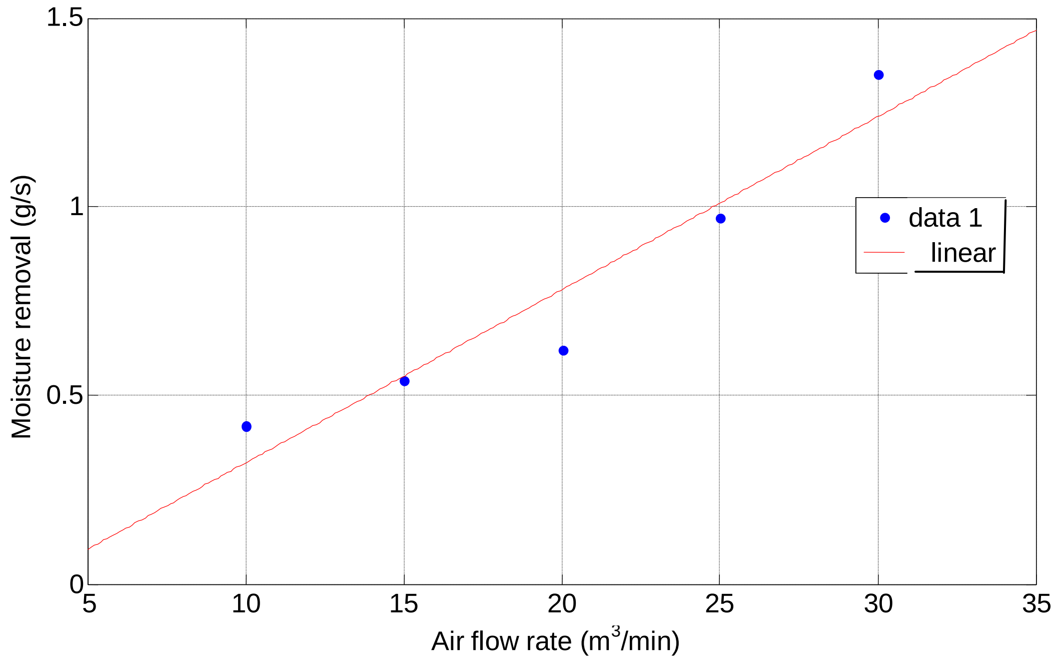

Figure 7. Effect of airflow rate on dehumidification

removed increases rapidly with the desiccant mass flow rate because increasing the solution flow rate ensures good contact between the desiccant and the air, which also increases the mass transfer coefficient.

Gandhidsan (2004) and Moon et al. (2009) reported that the condensation rate increased with the increase of solution flow rate, However, Fumo and Goswami (2002) reported that the desiccant flow rate did not cause significant variations in the water condensation rate. Although the dehumidification rate increased with an increase of the liquid desiccant flow rate, it stagnated at a high desiccant flow rate. Fumo and Goswami (2002) study does not contradict the findings obtained by these studies.

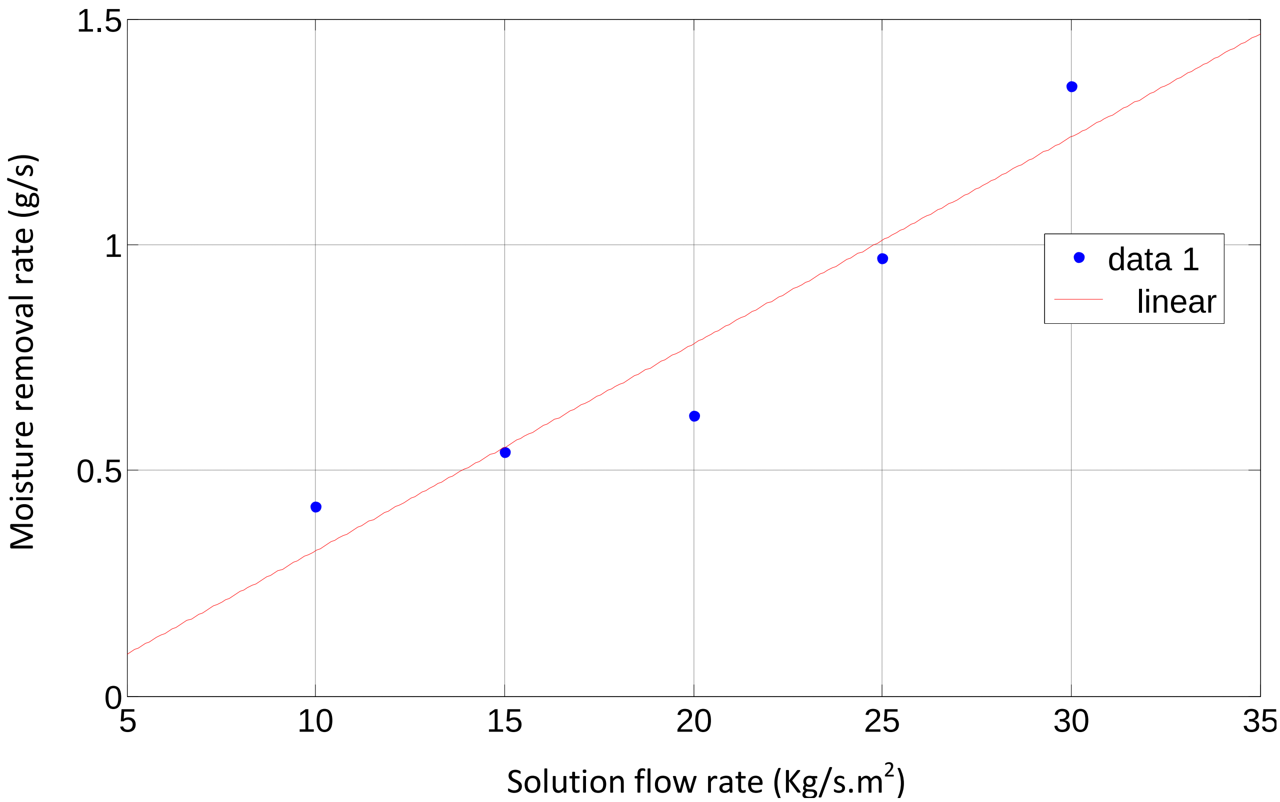

Figure 8. Effect of liquid desiccant flow rate on dehumidification.

Figure 8 revealed that the influence of the solution flow rate on the moisture removal can be assumed to be linear.

As shown in Table 5, the effects of the variables on the dehumidification rate in this study are compared with other studies reported in the literature. All of these studies show the trends of the effects of various design parameters, such as the mass flux of air and solution, inlet temperature and humidity of air, and inlet temperature and concentration of solution on the moisture removal rate.

Conclusions

The liquid desiccant air-conditioning system utilizes low-grade heat energy and is environmentally friendly. The dehumidifier and regenerator are the key components of the liquid desiccant air-conditioning system.

The packing height of the absorber tower was predicted using the finite difference model. This model gives a fairly accurate packing height based on fundamental equations, empirical correlations, and some assumptions. This experimental study was carried out to evaluate the dehumidification rate of air in a dehumidifier packed tower structure using the liquid desiccant H_2O/LiCl and by varying the liquid desiccant and airflow rates and inlet air and desiccant conditions.

Therefore, the study revealed that rate of condensation increases with airflow rate and desiccant mass flow rate and decreases when the inlet air temperature is increased. The influence of the design variables studied on the dehumidi-fication rate can be assumed to be linear.

Nomenclature: **A,** Area of packing (m^2); **ALT**, Altitude; **a_t**, total surface area density of packing (m^2/m^3); **a_w,** wetted surface area of packing (m^2/m^3); **Cp,** constant pressure heat capacity (kJ/kmol.K); **D,** diffusivity (m^2/s); **D_p**, nominal size of packing (m); **DP,** dew point; **DR,** daily range; **F_G**, gas mass transfer coefficient (Mol/s.m^2); **F_L**, liquid desiccant mass transfer coefficient (Mol/s.m^2); **G**, gas mass velocity (Kg/s.m^2); **g**, gravitational (m/s^2); **h**, enthalpy (kJ/kg); **h_c,** heat transfer coefficient (W/m^2.K); L, liquid mass velocity (kg/s.m^2); **LAT,** latitude; $\dot{m}$, mass flow rate (kg/s); **M**, molecular weight (kg/kmol); **RH,** relative humidity; **T,** temperature (°C); **x,** liquid mol Fraction ($mol_v/mol_{solution}$); **y,** gas mol fraction ($mol_v/mol_{solution}$); Z, packing height (m).

Greek: **λ**, Latent heat of condensation; **μ**, viscosity; **ρ,** density; ω_a, air humidity ratio (kg_{vapor}/kg_{air}); **ξ,** solution concentration.

Table 5. Summary results of dehumidifier performance.

X (kg/kg)	T_{si} (°C)	m_s (kg/m^2.s)	ω_{ai} (g/kg)	T_{ai} (°C)	m_a (kg/m^2.s)	Performance parameter	Desiccant	Reference
0.33-0.35 ↔	25-35 ↓	5.0-7.5 ↔	14-22 ↑	30-40 ↔	0.89-1.513 ↑	Range m_{cond}	LiCl	Fumo and Goswami (2002)
0.94-0.96 ↑	25-35 ↓	4.5-6.5 ↔	11-22 ↑	25-35 ↔	0.5-2.0 ↑	Range m_{cond}	TEG	Oberg and Goswami (1998)
↑	↔	↑	-	m_{cond}	3.5-6	Range	$CaCl_2$	Gandhidasan (2004)
28.5-34.5 ↑	-	104.2 (g/s)	0.011-0.018 ↑	38-40 ↔	74.9 (g/s) ↑	Range m_{cond}	LiCl	Yin and Zhang (2007)
0.33-0.43 ↑	26.2-38.2 ↓	1.26-2.57 ↑	16-24	26.8-39.0 ↓	0.91-1.99 ↑	Range m_{cond}	$CaCl_2$	Moon and Bansal (2009)
0.93-0.98 ↑	28-45 ↓	0.13-1.0 ↑	-	25.4-44 ↓	1.5-2.6 ↑	Range m_{cond}	TEG	Abdul-Wahab et al. (2004)
0.33-0.35 ↔	20-25 ↓	0.35-1.5 ↑	12-20 ↑	22-27 ↓	0.5-1.5 ↑	Range m_{cond}	LiCl	Present study

↑: Increasing trend, ↓: Decreasing trend, ↔: No significant effect.

Subscripts: a, Air; **I,** interface; **L,** liquid; **s,** desiccant solution; **0,** reference state; **v,** vapor; **W,** water; **G,** gas.

ACKNOWLEDGMENT

The support provided for this work under Iranian Fuel Conservation Company (IFCO) is gratefully acknowledged.

REFERENCES

Abdul-Wahab S A, Abu-Arabi M K, Zurigat Y H (2004). Effect of structured packing density on performance of air dehumidifier. Energ. Convers. Manage., 45: 2539-2552.

Aktacir M A (2011). Experimental study of a multi-purpose PV refrigerator system. Int. J. Phys. Sci., 6: 746-757.

ASHRAE (2008). Handbook–HVAC Syst., Equipment, American Society of Heating, Refrigerating and Air–Conditioning Engineers, Inc.

Bozdemir M (2010). The analysis of humidity factor in cestamide materials on surface roughness with the help of artificial neural network. Sci. Res. Essays, 5: 1370-1375

Bozdemir M (2011). The effect of water absorbtion in cast PA6G material on processing parameters. Sci. Res. Essays, 6: 714-719

Chen X Y, Li Z, Jiang Y, Qu K Y (2006). Analytical solution of adiabatic heat and mass transfer process in packed-type liquid desiccant equipment and its application. Sol. Energy, 80: 1509-1516.

Daou K, Wang R Z, Xia Z Z (2006). Desiccant cooling air conditioning: a review. Renew. Sustain. Energy Rev., 10:55-77.

Fumo N, Goswami D Y (2002). Study of an aqueous lithium chloride desiccant syst., air dehumidification and desiccant regeneration. Sol. Energy, 72: 351-361.

Gommed K, Grossman G (2007). Experimental investigation of a liquid desiccant system for solar cooling and dehumidification. Sol. Energy, 81:31-138.

Gandhidasan P (2004). A Simplified model for air dehumidification with liquid desiccant. Sol. Energy, 76:409-416.

Ibrahim TK, Rahman MM, Abdalla AN (2011). Improvement of gas turbine performance based on inlet air cooling systems: A technical review. Int. J. Phys. Sci., 6:620-627

Tu M, Ren CQ, Zhang LA, Shao JW (2009). Simulation and analysis of a novel liquid desiccant air-conditioning system. Appl. Therm. Eng., 29: 2417-2425.

Mago P, Goswami DY (2003). A study of the performance of a hybrid liquid desiccant cooling system using lithium chloride. J. Sol. Energy Eng., 125: 129-131.

Mei L, Dai YJ (2008). A technical review on use of liquid-desiccant dehumidification for air-conditioning application. Renew. Sust. Energ Rev., 12:662-689.

Moon CG, Bansal PK, Jain S (2009). New mass transfer performance data of a cross- flow liquid desiccant dehumidification system. Int. J. Refrig. 32: 524-533.

Oberg V, Goswami DY (1998). Experimental study of the heat and mass transfer in a packed bed liquid desiccant air dehumidifier. J. Sol. Energy Eng., 120: 289-297.

Ramzy AK, Hamed AM, Awad MM, Bekheit MM (2010). Theoretical investigation on the cyclic operation of radial flow desiccant bed dehumidifier. J. Eng. Technol. Res., 2: 96-110

Roth KW, Westphalen D, Dieckmann J, Hamilton SD, Goetzler W (2002). Energy consumption characteristics of commercial building HVAC systems. Volume III: Energy Savings Potential. TIAX LLC, Reference No. 68370-00

Treybal RE (1981). Mass Transfer Operations. McGraw –Hill, USA, New York.

Yin Y, Zhang X, Chen Z (2007). Experimental study on dehumidifier and regenerator of liquid desiccant cooling air conditioning system. Build. Environ., 42: 2505-2511.

Yoo C K, Ataei A, Kim YS, Kim MJ, Liu HB, Lim JJ (2010). Environ. Syst. Engineering: A state of the art review. Sci. Res. Essays, 5: 2341-2357.