Performance Evaluation of a Desiccant Dehumidifier using Different Desiccants

Prachi Rai
S.K. Shukla

ABSTRACT

This article reports an analysis of different empirical dehumidification effectiveness correlations for packed bed columns using two desiccant solutions viz. lithium chloride and calcium chloride. The analysis shows wide variations in effectiveness values ranging from 10% to 50% or more, with higher deviations occurring for lower ratios of liquid to gas flow rates. It is also reported that for same ratio of liquid to Gas flow rate, the dehumidification effectiveness is more for the desiccant system using LiCl. Design variables found to have the greatest impact on the performance of the dehumidifier, are desiccant concentration (slope = 2.7), desiccant temperature (slope = –1.4), air flow rate (slope = –0.9), and air humidity ratio (slope = 2.5).

Keywords: Dehumidifier, Hybrid desiccant cooling system, Lithium chloride and Calcium chloride, Cascade ring, Random packing

Nomenclature

Cp = Specific Heat (kJ/kg•K)
P = Pressure (bar)
T = Temperature (K)
t = Temperature (°C)
ε = Effectiveness
ρ = Density (kg/m^3)
ξ = Concentration
Φ = Relative Humidity (%)
ω = Specific Humidity (g/kg)
L = Latent Heat of Condensation of Water (kJ/kg)
m° = moisture removal rate (g/s)
G = mass flow rate (kg/s)

Subscripts

a	=	Air
s	=	Solution
i	=	Inlet
o	=	Outlet
HE	=	Heat Exchanger
$CaCl_2$	=	Calcium Chloride
LiCl	=	Lithium Chloride

INTRODUCTION

In a conventional cooling system, water of air is removed by condensation. However the overall Coefficient of Performance (COP) of a conventional cooling system decreases because reheating is often required after dehumidification. Hybrid desiccant cooling systems which integrate desiccant dehumidifier with conventional cooling system can handle both the latent and sensible load respectively. Additionally, hybrid desiccant cooling systems can also use lower-grade energy such as solar energy, waste heat etc. as heat source for regeneration (Jain S, et al. 1995; Mavroudaki P. et al. 2005; Halliday SP et al. 2002; Yadav YK 1995 and Dai Y.J. et al. 2001).

Dehumidification of the air is achieved with simple technological means through condensation of the contained water by under running the dew point temperature (Kinsara A.A. et al. 1997; Aristov YuI, et al. 2000; and Techajunta S. et al. 1999). The heart of this technology is the dehumidification wheel, which contains a matrix with a highly porous structure that captures the vapor molecules of an air stream on one side and transfers this humidity to a hot air stream on the other side (regeneration air) usually based on a combination of adsorptive dehumidification and evaporative cooling.

Jain S. et al. (2000) suggested that solar energy can be used as the driving heat, leading to interesting possibilities when the cooling loads of a building coincide with the availability of solar radiation. Mazzei et al. (2002) presented a theoretical and critical review of HVAC dehumidification systems for thermal comfort, including solid sorption cycles. Alizadeh et al. (2002) designed, optimized and constructed a prototype of a forced flow solar collector/regenerator. They employed an aqueous solution of calcium chloride as desiccant and studied the influence of parameters, such as air and desiccant solution flow rates as well as the climatic conditions on the regenerator's performance.

Dai Y.J. et al. (2002) revealed that solar desiccant systems presents some technical limitations in hot and humid climates, mainly due to the high latent loads handled by the wheel, and the reduced potential of evaporative cooling. Conde, M.R., (2004) explained specific sequences of air treatments which can be implemented to cope with these conditions and to improve the effectiveness of the cycle.

Shukla S.K., and Singh S.K., (2011) experimentally investigated a phase change material storage unit for cooling of room air using night coolness storage in harsh summer seasons. Qiu G.Q. & Riffat S.B. (2010) tested a novel air dehumidifier using HCOOK solution to be feasible and effective. It was found that the air flow rate and its relative humidity have big impacts on air dehumidification. Their study explains that the decrease of relative humidity could be over 25% using a strong HCOOK solution to desiccate highly humid air (>75% RH). However, it does not dehumidify efficiently the air with a relative humidity lower than 43% even if strong HCOOK desiccant solution is used. In highly humid spaces, such as bathroom, swimming pool, or kitchen, air dehumidification using a liquid desiccant is very efficient.

Xiong Z.Q. et al., (2010) first studied exergy performances of the dehumidifier and the regenerator to find out the proper ranges for the important operating parameters such as the desiccant concentration, air humidity ratio, flow rate ratio of desiccant to air, and regeneration temperature. They also carried out the exergy analysis of the basic liquid desiccant dehumidification system with appropriate parameters. It was revealed that the exergy efficiency of the basic system is only 6.3%. It was also found that desiccant–desiccant heat exchanger (HE), hot-water–desiccant HE, and cooling-water-desiccant HE contributes to 75.9% of the entire exergy loss because of the high temperature difference between the hot regenerated liquid desiccant and the warm diluted liquid desiccant, and the incomplete heat recovery.

Shukla S.K and Patnawar P., (2012) presented parametric study of a hybrid desiccant cooing system using liquid H2O/CaCl2 as desiccant. The effects of various parameters on performance of the system have also been observed. It is found that the influence of the parameters studied on the dehumidification rate is similar to those reported earlier. In the present work, the experiments have been performed to compare the performance of the desiccant cooling system using CaCl2 and LiCl as desiccant. It is because of that reason that we need an economic system which can handle air to a much lower temperature while maintaining good thermal performance and lower cost if compared with conventional desiccant cooling system.

EXPERIMENTAL SETUP

In order to perform the experimental studies on hybrid desiccant cooling system, an experimental setup has been designed and fabricated (Figure 1). The Hybrid desiccant cooling system basically consists of two packed towers. One is for absorption and another one is for regeneration. Both towers are cylindrical in shape and of similar size. The towers are made of fiber reinforced plastic (FRP) of thickness 4 mm and it has a constant height of 100cm. Packing is done using polypropylene cascade ring of specific surface area 205 m2/m3 for a height of 30 cm. Beneath both the towers a collection tank made up of aluminium each for storage of the liquid desiccant has been provided. The arrangement for the tanks is such that heated desiccant is sprayed on to the regenerator and cool desiccant is sprayed on to the absorber.

Now in the absorber, moisture is absorbed from the incoming air stream, due to the vapor pressure difference between the air and solution. Thereby, the dehumidification takes place. From the absorber the desiccant falls into a collection tank where it is heated to an elevated temperature so as to increase its vapor pressure and this heated desiccant is sprayed on to the regenerator where the moisture is transferred into the air stream. For achieving this 1 kW immersion type heater along with a digital temperature controller for achieving the desired desiccant temperature is used. A PT100 sensor is also used to sense the temperature of the heated desiccant. The other tank is cooled using an ice which is covered with thermocol to avoid exchange any heat with surrounding.

In the towers, the desiccant and air flow in a counter current manner. Desiccant is sprayed on to the packing and air moves upward through the packing. For obtaining the airflow a centrifugal fan of capacity of 10 m^3/min are installed at the entry of the two towers. Two centrifugal pumps each which have a maximum discharge of 800 liter/hour are used for pumping the desiccant into the two towers. Desiccant is distributed into the towers by means of multi point distributor made of PVC pipes and it trickles down through the packing to the outlet which is situated at the bottom. Demister pads are placed at the top of the two towers to eliminate desiccant carry over through the air stream. Once steady state conditions have been achieved, the data for different parameters like inlet and outlet temperature of the desiccant and air, inlet and outlet air relative humidity and inlet and outlet desiccant concentration have been taken.

UNCERTAINTY ANALYSIS

An important argument is the accuracy of measured data as well as the results obtained by experimental studies. An uncertainty analysis was performed using the method described by Holman (1994). In the present study, the temperatures were measured with appropriate instruments

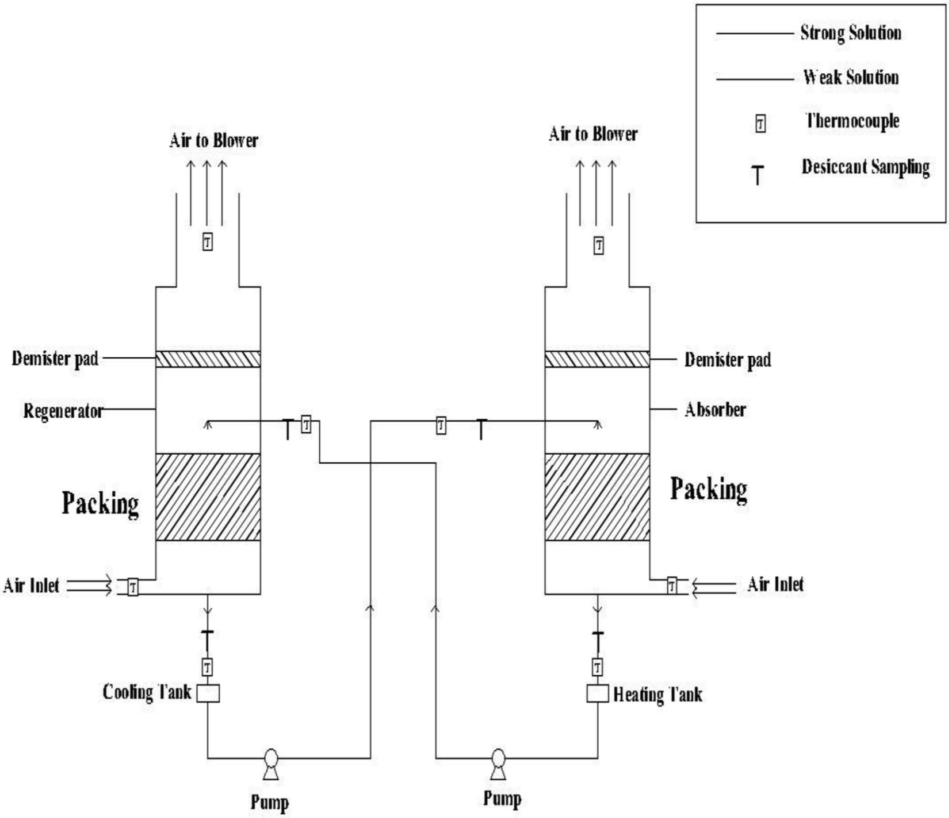

Figure 1. Schematic diagram of Experimental Setup

explained previously. Type B uncertainty, being dependent upon the measuring instrument, cannot be reduced by augmenting the number of measurements. Generally, sensor specifications furnish the accuracy, a, of an instrument, which is the maximum deviation from the true value. Using root sum square method the overall accuracy in COP_P was determined to be ±14.77%

Mathematical Modeling

Following assumption were made in order to incorporate the proposed model [28]:

- The dehumidification and enthalpy effectiveness are constant.
- Latent heat released from the condensate moisture was absorbed by the cold desiccant solution.
- The water absorption is negligible compared to the air and desiccant flow rates (i.e., the air and desiccant flow rates are constant through the conditioner).
- No desiccant particulates were carried out by the supplying air.
- Desiccant solution temperature and concentration sprayed on each cellulose fiber membrane were the same.
- The conditioner is adiabatic (i.e., no heat losses to the ambient surroundings occur).

Moisture balance on the air side:

$$\frac{\partial}{\partial t}\left(\rho_v A_v\right) + \frac{\partial}{\partial z}\left(\rho_v V A_c\right) = \rho_{da} K_m P \left(Y_w - Y_a\right) \tag{1}$$

Moisture balance on the desiccant side:

$$\frac{\partial}{\partial t}\left(m_w\right)_d = \rho_{da} K_m A_s \left(Y_a - Y_d\right) \tag{2}$$

Energy balance on the air side:

$$\left(\rho_{da} C_{pda} + \rho_v C_{pv} Y_a\right) A_c \frac{\partial T_a}{\partial t} + \left(m_{da} C_{pda} + m_v C_{pv}\right) \frac{\partial T_a}{\partial z} K_h P \left(T_d - T_a\right) + \rho_{da} K_m P \left(Y_a - Y_d\right) C_{pv}\left(T_d - T_a\right) \tag{3}$$

Energy balance on the desiccant side:

$$m_d C_{pd}\frac{\partial T_d}{\partial t} + m_w C_{pw}\frac{\partial T_d}{\partial t} + m_m C_{pm} Y_a A_c \frac{\partial T_d}{\partial t} + \left(m_{da} C_{pda} + m_v C_{pv}\right)\frac{\partial T_a}{\partial z} = K_h A_c \left(T_a - T_d\right) +$$

$$\rho_{da} K_m A_s \left(Y_a - Y_d\right) Q + \rho_{da} K_m A_s \left(Y_a - Y_d\right) C_{pv} \left(T_a - T_d\right) \quad (4)$$

where Y is the humidity ratio, T is the temperature, t and z are the time and axial direction, K_m and K_h are the mass and heat transfer coefficients, A and P are the area and the perimeter, ρ is the density, V is the velocity, Cp is the isobaric specific heat, and Q is the adsorption heat. The subscripts d, d_a, l, m, v and w stand for desiccant, dry air, liquid water, matrix material, water vapor and duct wall respectively. Equations (1) through (4) will be solved. In the current work, the period of the wheel rotation is fixed at 500 s, the wheel thickness is 0.2 m and its diameter is 0.5 m. Also the dehumidification and regeneration areas are considered equal and the air velocity is set at 3 m/s.

The air moisture removal effectiveness is defined by

$$\varepsilon_m = \frac{P_{a,i} - P_{a,o}}{P_{a,i} - P_{s,i}} \quad (5)$$

where Pa,i, Pa,o, Ps,i, designate respectively the air inlet water vapor pressure, air outlet water vapor pressure, and the solution vapor pressure. The effectiveness of the heat exchanger is assumed to be equal to 0.8 and it can be written as follows:

$$\varepsilon_{HE} = \frac{T_{s,o} - T_{s,i}}{T_{s,o} - T_{c,i}} \quad (6)$$

Where Ts,o, Ts,i, Tc,i, designate, respectively, the desiccant solution outlet temperature, the desiccant solution inlet temperature, the cooling medium inlet temperature.

The outlet temperature of the desiccant solution is derived from the above expression and represented by the expression

$$T_{s,o} = \frac{T_{s,i} - \varepsilon_{HE} T_{c,i}}{1 - \varepsilon_{HE}} \quad (7)$$

The relation linking the concentrations of inlet and outlet desiccant solution is given by

$$\frac{1}{\varepsilon_o} = \frac{1}{\varepsilon_i}\left(1 + \frac{\dot{m}}{G_i}\right) \tag{8}$$

Finally the mass rate of moisture removal is obtained from above equation.

Now, if a heat exchanger is used between the absorber and regenerator, then the moisture removal rate is given by following equation:

$$\dot{m} = 1/L[\{(Cs * \varepsilon HE)(Ts,i - Tc,i)\}/(1 - \varepsilon HE) - \{Ca * (Ta,i - Ts,i)\}] \tag{9}$$

where Cs and Ca designate the heat capacities of the solution and the air, respectively, L designates the latent heat of condensation of water.

The moisture absorption capacity per liter solution per second is defined as the dehumidification capability of this sort of solution or the moisture removal rate by air is given by the following equation:

$$\varepsilon_{moisture} = m_{moisture} = m_a\,(\omega_a - \omega_s) \tag{10}$$

where,

$$\omega_a = 0.622\,\frac{\Phi\% * Pa}{P - \Phi\% * Pa} \tag{11}$$

$$P = 101.325 \text{ kPa} \tag{12}$$

$$\omega_s = (1.43248e^t/14.31737 - 0.14869) - (1.43248e^t/14.31737 - 0.14869 - 1.3372e^t/22.12633 - 0.81576)/5 * (35 - \xi) \tag{13}$$

Vapor pressure of the air is calculated by

$$P = 0.61078 \exp\left(\frac{17.269 * t}{237.3 + t}\right) \tag{14}$$

Vapor pressure of the LiCl is calculated by

$$P_{CaCl2} = A_{25}\, f(\xi,\theta) + f(\text{water}) \tag{15}$$

$$\theta = \frac{T}{228} - 1 \tag{16}$$

$$f(\xi,\theta) = A + B\theta \tag{17}$$

$$A = 2 - \left(1 + \left(\frac{\xi}{A0}\right)^{A1}\right)^{A2} \tag{18}$$

$$B = \left(1 + \left(\frac{\xi}{A3}\right)^{A4}\right)^{A5} - 1 \tag{19}$$

$$A_{25} = 1 - \left(1 + \left(\frac{\xi}{A6}\right)^{A7}\right)^{A8} - A9e^{\frac{-(\xi-1)^2}{0.005}} \tag{20}$$

f (*water*) is the vapor pressure above the normal water surface at different temperature.

A_0	A_1	A_2	A_3	A_4	A_5	A_6	A_7	A_8	A_9
0.28	4.3	0.6	0.21	5.1	0.49	0.362	-4.75	-0.4	0.03

Vapor pressure on the water surface is calculated by

$$\ln(p/p_c) = (a_1\tau + a_2\tau^{1.5} + a_3\tau^3 + a_4\tau^{3.5} + a_5\tau^4 + a_6\tau^{7.5})\, T_c/T \tag{21}$$

where p is the pressure, $T = T_{90}$, and subscript c indicates the values at the critical point; $\tau = 1 - T/Tc$. The values for substitution in the equation are:

$T_c = 647.096$ K, $p_c = 220.64$ kPa, $a_1 = -7.85951783$, $a_2 = 1.84408259$,

$a_3 = -11.7866497$, $a_4 = 22.6807411$, $a_5 = -15.9618719$, $a_6 = 1.80122502$

The Heat Capacity of Air, Ca is calculated by

$$Ca = 1.9327E - 10{*}T4 - 7.9999E - 07{*}T3 + 1.1407E - 03{*}T2 - 4.4890E - 01{*}T + 1.0575E + 03 \tag{22}$$

The Heat Capacity of LiCl is calculated by

$$Cs = C_{H2O}(T)\,(1 - f_1(\xi))\, f_2(T) \tag{23}$$

$$C_{H2O} = 88.7891 - 120.1958\,\theta^{0.02} - 16.9264\,\theta^{0.04} + 52.4654\,\theta 0.06 + 0.10826\,\theta^{1.8} + 0.4698\theta 8\theta \tag{24}$$

$$\theta = \frac{T}{228} - 1 \tag{25}$$

$$f_1(\xi) = A\xi + B\xi^2 + C\xi^3 \tag{26}$$

$$f_2(T) = F\theta^{0.02} + G\theta^{0.04} + H\theta^{0.06} \tag{27}$$

A	B	C	D	E	F	G	H
1.63799	-1.6900	1.05124	0.0	0.0	58.5225	-105.63	47.794

Density of LiCL

$$P_{LiCl} = \rho_{H2O}(\tau)\ \Sigma_{i=0}^{3} \rho i \left(\frac{\xi}{1-\xi}\right)^i \tag{28}$$

$$\rho_{H2O}(\tau)\ \rho_{cH2O}\left(1 + B_0\tau^{1/3} + B_1\tau^{2/3} + B_2\tau^{5/3} + B_3\tau^{16/3} + B_4\tau^{43/3} + B_5\tau^{110/3}\right)$$

ρ_0	ρ_1	ρ_2	ρ_3
1.0	0.836014	–0.436300	0.105642

where $\tau = 1 - \theta$ and ρ_{cH2O} is the density of water at the critical point (322 kg/m^3)

B_0	B_1	B_2	B_3	B_4	B_5
1.993771843	1.0985211604	-0.50944929	-1.76191242	-44.9005480	-723692.261

Coefficient of performance of the system is calculated by

$$COP = \frac{Q_{evap}}{p * t}$$

Where

Q_{evap} is the heat absorbed from the air.
P is power of pump
t is time
$Q_{evap} = ma * Cpa * (To - Ti)$

RESULTS AND DISCUSSIONS

The effect of the different variables namely air inlet temperature, desiccant inlet temperature and solution to air flow ratio on the system performance has been studied by changing the humid air temperature from 28°C to 42°C, and keeping the pre-set humid air relative humidity at 70% with LiCl as desiccant solution. It is found that as the temperature increases, moisture removal rate increases (Figure 2). But if the temperature of inlet air is considerably more than desiccant temperature, the desiccant temperature will increase, resulting in a reduction in the potential for mass transfer.

Figure 3 explains that the desiccant flow rate does not cause the significant variation in the water condensation rate; however, the liquid flow rate must be high enough to ensure wetting of the packing. For, the range of the variables studied, humidity effectiveness for the absorber remains mostly stable, and no variation higher than 6% was found.

The water condensation rate increases with the inlet air humidity ratio with a slope of 2.5 (Figure 4). It happens because a higher humidity implies a higher air vapor pressure and consequently higher potential for mass transfer.

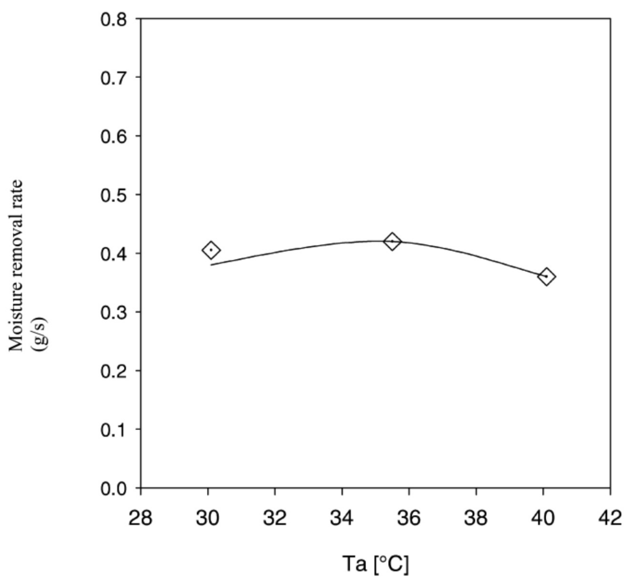

Figure 2. Variation of Moisture Removal Rate with Inlet Air Temperature

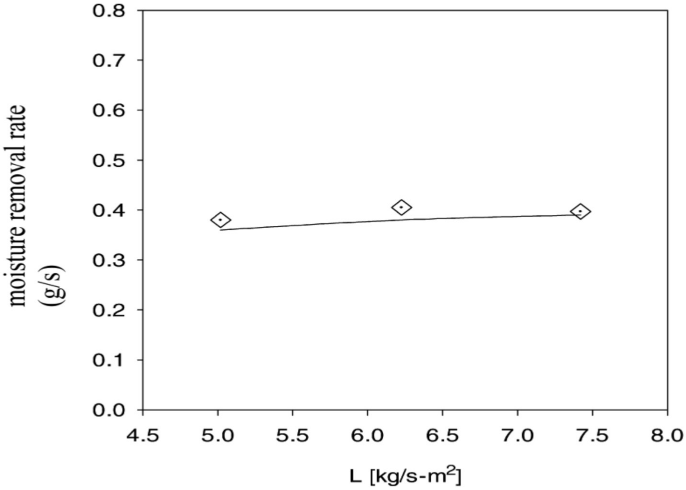

Figure 3. Variation of moisture Removal Rate with Desiccant flow rate

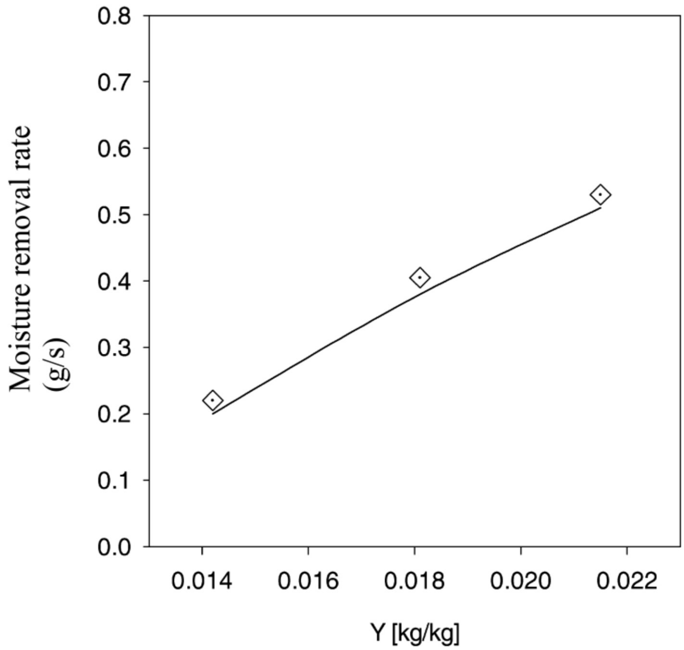

Figure 4. Variation of Moisture Removal Rate with Inlet Humidity

The water condensation rate decreases with the desiccant temperature with a slope of 21.4 (Figure 5). It is due to fact that a higher desiccant temperature gives a lower potential for mass transfer in the dehumidifier, resulting in a lower condensation rate.

The water condensation rate increases with the desiccant concentration with a slope of 2.7 (Figure 6). A higher desiccant concentration gives a higher potential for mass transfer in the dehumidifier resulting in a greater condensation rate.

COP is the ratio of heat absorbed by the system and the electrical energy. As the ambient temperature is increased the heat absorbed by the system increases and hence the COP. (Figure 7)

Comparison of the System with Licl and Cacl2 as the Desiccant

Figure 8 shows that the system using LiCl has better dehumidification effectiveness (moisture removal rate) than the system using CaCl2. For same ratio of liquid to Gas flow rate the dehumidification effectiveness is more for LiCl. So, LiCl proves to be a better desiccant.

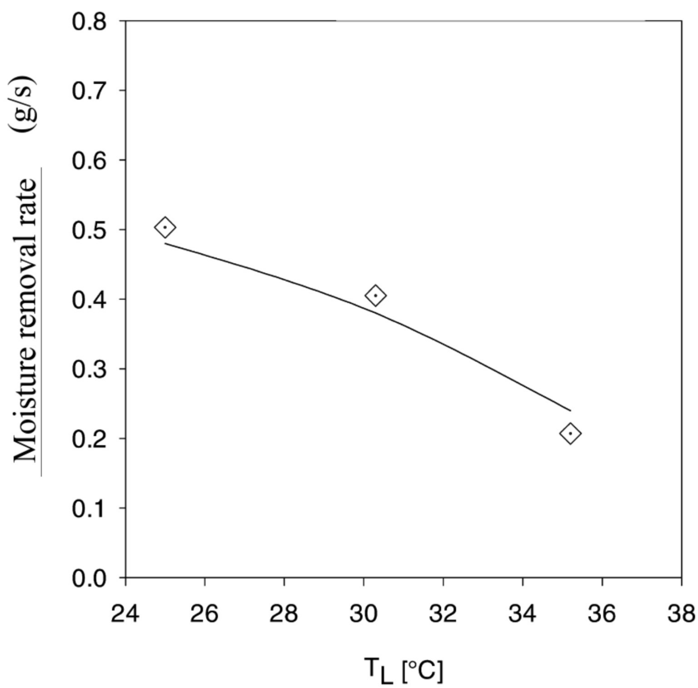

Figure 5. Variation Of Moisture Removal Rate with Desiccant temperature

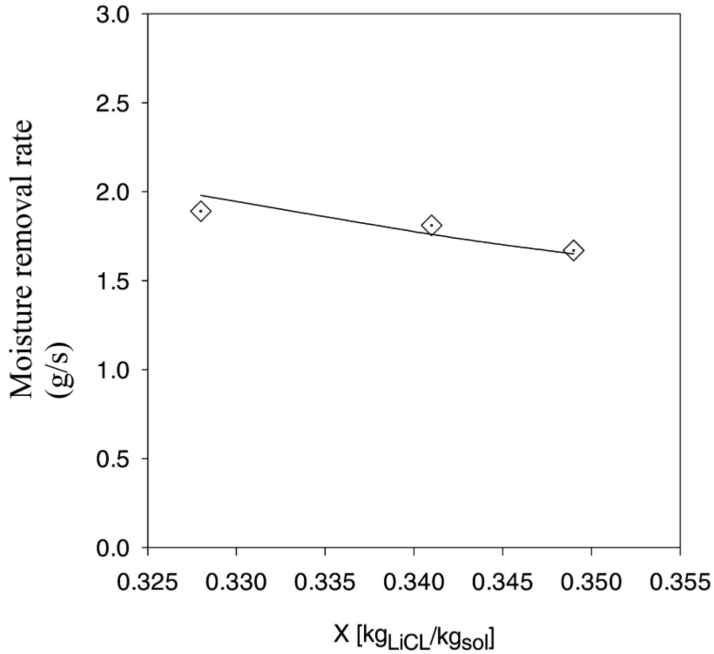

Figure 6. Variation of Moisture Removal Rate with Desiccant Concentration

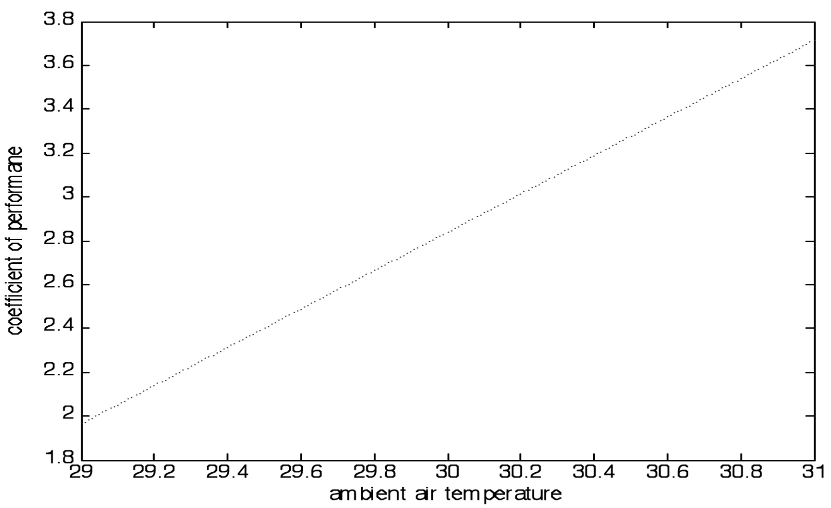

Figure 7. Variation of COP with Inlet Air Temperature

As we vary the inlet air temperature the dehumidification effectiveness increases but the effectiveness is more in case of LiCl than CaCl2 as depicted in Figure 9.

Figures 10, 11 and 12 show the comparative study of the LiCl desiccant system and CaCl2 desiccant system. From these figures, it is evident that LiCl has a better moisture removal rate than CaCl2 and in turn confirms to be a better desiccant than CaCl2.

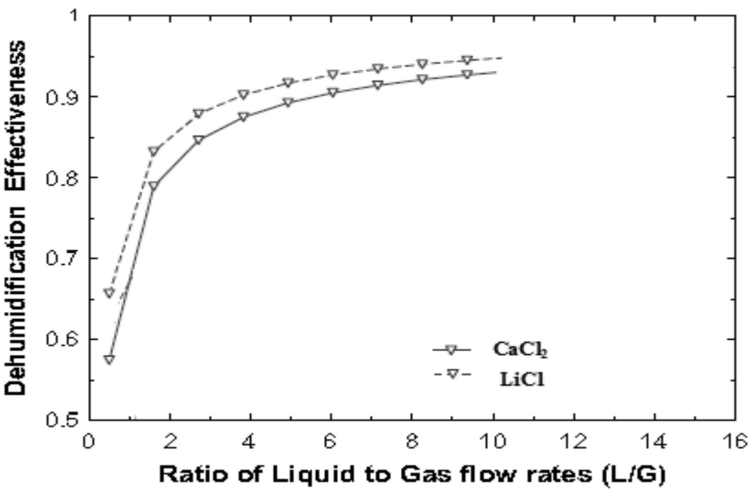

Figure 8. Comparison of CaCl2 and LiCl for Dehumidification Effectiveness

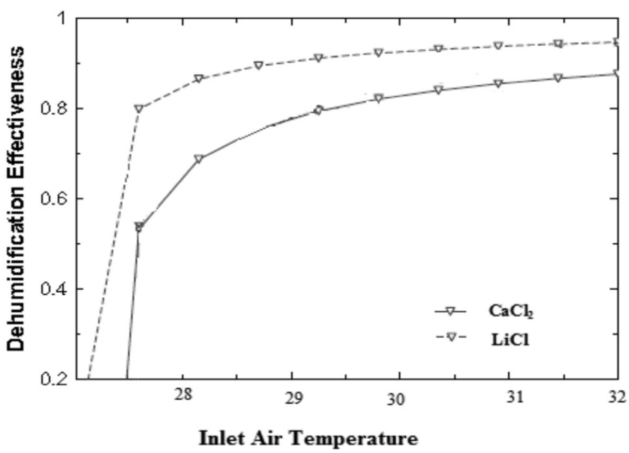

Figure 9. Comparison of Effectiveness of CaCl2 and LiCl system by varying inlet air temperature

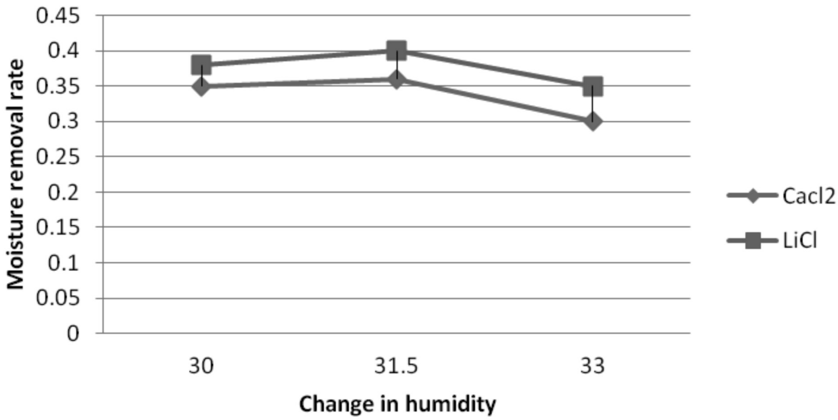

Figure 10. Comparison of Moisture Removal Rate of CaCl2 and LiCl system by Change in the humidity

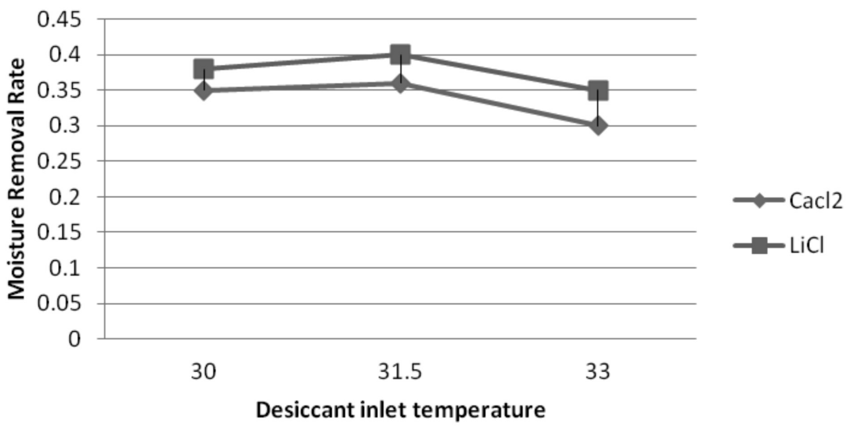

Figure 11. Comparison of Moisture Removal Rate of CaCl2 and LiCl system by varying desiccant inlet temperature

CONCLUSIONS

Reliable sets of data for air dehumidification and desiccant regeneration using lithium chloride were obtained. The influence of the design variables studied on the water condensation rate from the air and evaporation rate from the desiccant can be assumed linear.

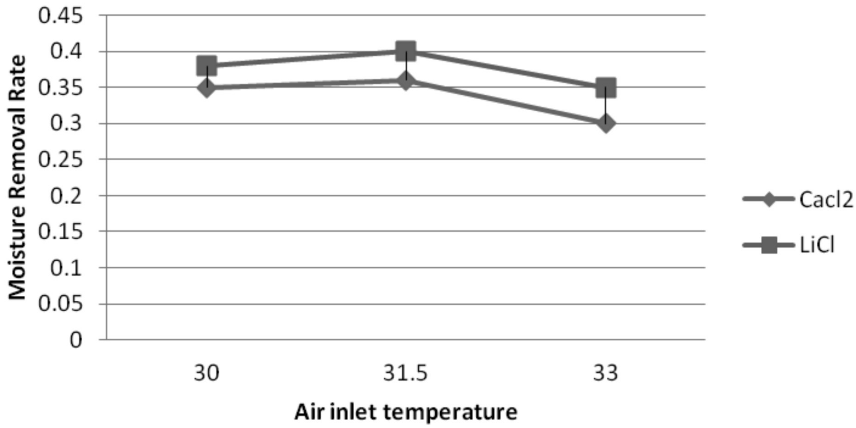

Figure 12. Comparison of Moisture Removal Rate of CaCl2 and LiCl system by varying air inlet temperature

Design variables found to have the greatest impact on the performance of the dehumidifier are: desiccant concentration (slope = 2.7), desiccant temperature (slope = –1.4), air flow rate (slope = –0.9), and air humidity ratio (slope = 2.5). Desiccant dehumidification systems are widely used in many industries including HVAC, for air-conditioning needs. These are commonly operated in a hybrid mode with vapor compression systems, to offer better indoor air quality (IAQ) and higher energy efficiencies especially for low sensible heat ratio and high humidity applications. The comparison of experimental performance of dehumidifiers reveals that proper selection of type of column with its operating parameters including solution to air flow rates, inlet concentration of desiccant and packing size results in high dehumidification effectiveness of about 0.9 or more. Internal cooling could help effectiveness values to exceed 1. A critical analysis of different empirical dehumidification effectiveness correlations for packed bed columns using two desiccant solutions viz. lithium chloride and calcium chloride has been reported in this article. The analysis shows wide variations in effectiveness values ranging from 10% to 50% or more, with higher deviations occurring for lower ratios of liquid to gas flow rates. It emerges from the study that there is a need to develop more comprehensive empirical models for performance prediction of desiccant dehumidifiers.

References

1. Jain S, Dhar PL, (1995). Evaluation of solid desiccant-based evaporative cooling cycles for typical hot and humid climates. International Journal of Refrigeration 18(5): 287–96.
2. Mavroudaki P, Beggs CB, Sleigh PA, Haliiday SP, (2005). The potential for Solar powered single-stage desiccant cooling in southern Euro. Applied Thermal Engineering, 22: 1129–40.
3. Halliday SP, Beggs CB, Sleigh PA, (2002). The use of solar desiccant cooling in the UK: a feasibility study. Applied Thermal Engineering, 22:1327–38.
4. Yadav YK, (1995). Compression-compression and liquid-desiccant hybrid solar space-conditioning system for energy conservation. Renewable Energy 7:719–23.
5. Dai Y.J., Wang R.Z., Zhang HF, Yu JD, (2001). Use of desiccant cooling to improve the performance of vapor compression air conditioning. Applied Thermal Engineering, 21:1185–205.
6. Kinsara AA, Omar M, Rabghi A, Alsayes MM, (1997). Parametric study of an energy efficient air conditioning system using liquid desiccant. Applied Thermal Engineering, 18(5):327–35.
7. Aristov YuI, Restucia G, Tokarew MM, Buerger H-D, Freni, (2000). A. Selective water sorbents for multiple applications. 11. CaCl2 confined to expanded vermiculite. React Kinet Catal Lett 71(2):377–84.
8. Henning H-M, Erpenbeck T, Hindenburg C, Santamaria IS, (2001). The potential of solar energy use in desiccant cycles. International Journal of Refrigeration, 24: 220–9.
9. Techajunta S, Chirarattananon S, Exell RHB, (1999). Experiments in a solar simulator on solid desiccant regeneration and air dehumidification for air conditioning in tropical humid climate. Renewable Energy 17:549–68.
10. Jain S, Dhar PL, Kaushik SC, (2000). Experiments studies on the dehumidifier and regenerator on a liquid desiccant cooling system. Applied Thermal Engineering, 20: 253–67.
11. Xiong Z.Q., Dai Y.J.& Wang R.Z. (2010). Exergy Analysis of Liquid Desiccant Dehumidification System,International Journal of Green Energy, 7(3): 241-262
12. Mazzei P, Minichiello F, Palma D, (2002). Desiccant HVAC systems for commercial buildings. Applied Thermal Engineering, 22: 545–60.
13. Alizadeh S, Saman WY, (2002). An experimental study of a forced flow solar collector/regenerator using liquid desiccant. Solar Energy 73(5):345–62.
14. Dai Y.J., Wang RZ, Xu YX, (2002). Study of a solar powered solid adsorption desiccant cooling system used for grain storage. Renewable Energy 25:417–30.
15. Conde, M.R., (2004). Properties of aqueous solutions of lithium and calcium chlorides: formulations for use in air conditioning equipment design. International Journal of Thermal Science 43:367–382.
16. Shukla S.K., and Singh, S.K. (2011), Analysis of PCM storage unit for night coolness storage in summer season, International Journal of Sustainable Energy 30(2):S142-S154.
17. Qiu G.Q. & Riffat S. B.(2010) Experimental Investigation on a Novel Air dehumidifier Using Liquid Desiccant, International Journal of Green Energy,7(2) : 174-180
18. Shukla S.K.and Patnawar P. (2012). Parametric studies of a hybrid desiccant cooling system. International Journal of Energy Engineering, 2(5): 253-258.
19. Holman, J.P., (1981), Heat Transfer, 5th Ed., McGraw-Hill Publishing Company.

ABOUT THE AUTHORS

Ms. Prachi Rai is a dual degree Master student, currently working in the Department of Mechanical Engineering IIT (BHU) Varanasi, India. Her areas of interest are Modeling of HVAC system, Life Cycle Cost Analysis etc.

Dr. S.K. Shukla is working as Associate Professor in Mechanical Engineering Department, Indian Institute of Technology (BHU), Varanasi, India. He has been post graduated and completed his Ph.D. from IIT Delhi. His areas of interest are Thermal Engineering, Heat and Mass Transfer Analysis in Solar Thermal Systems and Design of Renewable Energy Systems, Modeling of HVAC systems. E-mail: shuskla@gmail.com. Tele fax; +91-542-6702825.