

Experimental & Theoretical Analysis of Solar Powered Desiccant Dehumidification System for Indoor Air

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ABSTRACT: The exponentially increasing need of human comfort in area of conditioned air has given giant thrust in energy consumption, specifically electrical energy, to run AC equipment's used in home or industry. Energy demand depletes the fossil fuel reservoir, affects the climate changes adversely and hence it is a matter of utmost concern to save the both. Calcium Chloride (CaCl_2) has been one of the most economical liquid desiccants and can be processed in many ways to reduce the latent heat content of air. The main objective of this research is to investigate the energy performance enhancement. Inconventional air cooling operation, the boiler providing regeneration heat to the weak desiccant solution consumes most of the energy. To reduce the regeneration energy consumption in the conventional air cooling system, a heat pump-integrated liquid desiccant is suggested in this research. This paper, gives us a detailed study of (CaCl_2) driven air dehumidifier. Consideration of basic working cycles of solar powered system, properties of liquid desiccant and practical methods used for heating, ventilation and air conditioning applications. Also explains, the performance review of the calcium chloride driven air dehumidifier for HVAC applications as its inception to the recent research going on (CaCl_2) incisively given with vital effects in its performance parameters to help summarize the progress and trends of application. It is noticeable from the given performance review that (CaCl_2) driven air dehumidifier has progressed well, working and experimented in different conditions with and without internal cooling of air dehumidifier in the field of HVAC. However, there is scope of improvement to satisfy the unmet demand of performance and to accommodate into its fully developed form to take its place in day to day need of air conditioning in place of conventional vapour compression system.

KEYWORDS: Solar collector, Liquid desiccant, Regeneration, Dehumidification.

I. INTRODUCTION

It is self-evident that electrical power consumption has been increasing significantly and has reached to the peak of all the other energy consumptions forms. The average temperature range over the year in building can be expected between temperature ranges of 12°C to 44°C in average in India [1]. Therefore, it is likely to have increased use of air conditioning specifically during hot and humid climate conditions. The contribution of power consumption for running the conventional vapour compression type air conditioning equipment's in domestic/commercial sector is as high as 30 to 40 % and highest among all remaining type of energy consuming equipment's used in domestic/commercial sector [1]. It directly correlates the consumption of fossil fuels, which in turn increases the greenhouse gases and hence global warming of the environment. In near future, if the price of fossil fuels shoots very high due to fast depletion of it, then it is almost impossible to use the dependent air conditioning equipment by the average and below average class of

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consumers. By the time ahead, if the reservoirs of the fossil fuel are exhausted too much then there must be an economic alternative option of having conditioned air to have comfort climate to inhabitant of hot and humid climate of tropical region of the planet earth. Desiccant in different forms have been used vastly in industry for dehumidification of air and proved economic. Desiccant dehumidification air-conditioning systems have excellent potential for cost-effective application in commercial buildings located in hot, humid climates. Hybrid liquid desiccant/vapour compression air-conditioning systems, in particular, are promising because they take advantage of the high efficiency for heat transfer inherent in vapour-compression systems and the high mass transfer potential of the liquid desiccants. Eventually, to be cost effective, hybrid systems should supply operational savings over challenging vapour compression air-conditioning systems and must be less costly to design and fabricate. In the hybrid air conditioning system, a desiccant is circulated between the evaporator and condenser of a vapour-compression air conditioner providing both operational and first-cost savings. Continuing analytical work at The University of Texas has shown potential first-cost savings resulting in a 34% reduction in the exchange area for both the evaporator and condenser and a 25% reduction in the required compressor capacity. Additionally, electric power drawn by the compressor is expected to drop by 25%, resulting in savings on electric utility bills. These savings are possible without sacrificing cooling efficiency or thermal comfort [2].

The different types of desiccants in many forms are used to dehumidify air. The desiccants are natural or synthetic substances capable of absorbing or adsorbing water vapour due to difference of water vapour pressure between the surrounding air and the desiccant surface. They are available in both liquid and solid states. Each of liquid and solid desiccant systems has its own advantages and shortcomings. In addition of having lower regeneration temperature and flexibility in utilization, liquid desiccant have lower pressure drop on air side. Solid desiccant are compact, less subject to corrosion and carryover. Commonly used desiccant materials include lithium chloride (LiCl), lithium bromide (LiBr), triethyleneglycol (TEG), and calcium chloride (CaCl₂), silica gels, aluminium silicates (zeolites or molecular sieves). Calcium chloride is one of the most potent moisture absorbing compounds found in nature. As the second most commonly occurring substance in seawater next to salt, calcium chloride is also non-toxic and environmentally friendly. Calcium chloride dissolves into a liquid brine partly due to the actuality that it is able to attract many times its self-weight in water. At a relative humidity (RH) of only 50%, calcium chloride's moisture absorption is 150% its weight in water. Its absorption increases exponentially as RH rises, to 600% at 85% RH.

II. RELATED WORK

According to the recent literature [1], liquid desiccant and evaporative cooling-assisted air conditioning systems are attracting more interest for realizing energy conservative air-conditioning options. The liquid desiccant system, which dehumidifies the process air, plays an important role in enhancing the performance of evaporative coolers. In a liquid desiccant system, the desiccant solution should be heated for regeneration and cooled for enhancing the moisture removal efficiency, which accounts for over 40% of the total energy consumption in a liquid desiccant and evaporative cooling-assisted air conditioning system [2]. However, if combined with low-grade heat sources, the liquid desiccant-integrated system may provide high energy saving potential over conventional air conditioning systems [3]. There is research available that [4–8] addresses the potential of deploying waste heat recovery or solar energy as a heat source, for regenerating the desiccant solution, and the water-side free cooling approach can be applied for cooling the desiccant solution.

Katejanekarn and Kumar [4] conducted a simulation on a solar-regenerated liquid desiccant system for dehumidifying the outdoor air. They concluded that solar radiation, ventilation rate, and desiccant solution concentration were the most influential factors determining the performance of their proposed system. Gommed and Grossman [5] proposed a liquid desiccant system which is assisted by a solar thermal system and a cooling tower. Their experimentally validated simulation confirmed that their proposed system could provide a high degree of dehumidification. Jain et al. [6] conducted an

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experimental study on the performance of a liquid desiccant system with a water-side free cooling and an electric heater for solution cooling and heating. They used lithium chloride and calcium chloride solutions and found that lithium chloride showed a better dehumidification performance. Alizadeh [7] conducted a theoretical and experimental study on a liquid desiccant air conditioning system assisted by a solar thermal system composed of flat plate solar collectors and an evaporative cooling system. It was found that the suggested system was an effective way of air-conditioning in hot and humid climates. Buker and Riffat [8] studied the adsorption of a polyethylene heat exchanger installed under photovoltaic panels for liquid desiccant regeneration. They concluded that their proposed system could be beneficial in relation to energy and contributive to a sustainable environment.

On the other hand, the heat pump-driven liquid desiccant system has also been studied [9–12] by focusing on its dehumidification performance, energy consumption, and matching heat pump capacity to the required solution cooling and heating loads. Yamaguchi et al. [9] conducted heat pump-driven liquid desiccant system performance tests and discussed methods for improving the system efficiency. They concluded that the coefficient of performance (COP) of the heat pump-driven liquid desiccant system could be increased by enhancing the compressor isentropic efficiency and the solution heat exchanger efficiency. Bergero and Chiari [10] evaluated the system performance of an heat pump-driven liquid desiccant system with a hygroscopic solution and hydrophobic membrane using a SIMULINK program (The Math works, Inc., Natick, MA, USA) by varying a few significant operating parameters. The simulation results showed that compared to a conventional system, their system could achieve savings exceeding 50%, under certain operating conditions. Zhan et al. [11] focused on methods for effectively releasing the leftover heat from the heat pump condenser after regenerating the desiccant solution. They tested two different types of assistant condensers for exhausting the extra heat: air-cooled and water-cooled assistant condensers. They indicated that systems with air-cooled and water-cooled assistant condensers showed 18% and 35% higher COP, respectively, compared to the reference heat pump-driven liquid desiccant system with no assistant condensers. Niu et al. [12] investigated methods for matching the heat pump capacity to the desiccant solution heating and cooling loads. They showed that the solution flow rate in the condenser, the revolutions of the compressor, and the air flow rate in the air-cooled assistant condenser should be simultaneously controlled to match the load in the heat pump-driven liquid desiccant system.

III. WORKING PRINCIPLE

This is an evaporative liquid desiccant based system that uses 100% outdoor air in compliance with the thermal demands of the rooms being served. The system handles sensible and latent load separately with the selective components required according to the outdoor air conditions. The (Figure 1) consists of a liquid desiccant unit for dehumidifying the process air, and an indirect evaporative cooler and direct evaporative cooler for sensible and adiabatic cooling of the process air. To enhance the cooling effect of the indirect evaporator cooler, the exhaust air can be supplied to the secondary channel of the indirect evaporative cooling experiencing direct evaporative cooling when its wet bulb temperature is lower than that of the outdoor air; otherwise, the outdoor air is used as scavenger air supplied to the secondary channels of the indirect evaporative cooler. The heating coil and sensible heat exchanger are located at the exhaust air side to maintain the neutral deck temperature set point when the dual duct approach is selected as the air distribution system.

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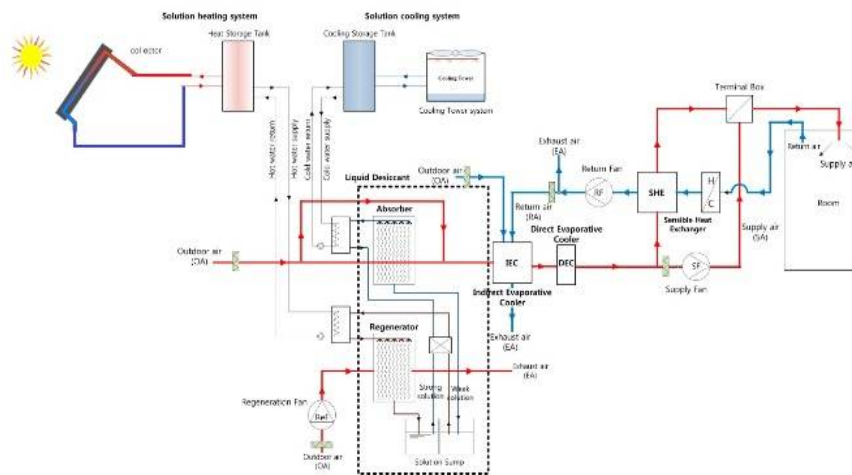


Figure 1 Schematic diagram of desiccant cooling system

A liquid desiccant unit is composed of an absorber, a regenerator, a sensible heat exchanger, a solution heating coil, and a solution cooling coil. In the absorber, the strong solution dehumidifies the entering process air. This dehumidification process is an exothermic process; thus, the strong solution should be cooled by the solution cooling coil before entering the absorber. The weak desiccant solution leaving the absorber should be delivered to the regenerator after it is heated by the solution heating coil up to the regeneration temperature. Both the dehumidification and regeneration processes are repeated in the LD unit during the operation, and the driving force of moisture transfer to and from the desiccant solution is the vapour pressure difference between the desiccant solution and the air passing through the solution in the absorber or in the regenerator [15]. In general, the weak desiccant solution that enters the regenerator should be heated to 45–80 °C for the regeneration process, whereas the strong solution entering the absorber should be cooled down to 15–30 °C for the dehumidification process [4–7, 16]. To reduce the energy consumed in solution cooling and heating, the solutions that leave the absorber and regenerator are preheated and precooled, respectively, by the sensible heat exchanger. The solutions leaving the sensible heat exchanger are heated and cooled in the heating and cooling coils, respectively, and a gas boiler and a chiller are generally used to treat the required solution heating and cooling loads in the coils. A packed-bed tower type absorber and a regenerator are commonly used [17]. A conventional liquid desiccant system with a gas boiler and a cooling tower to treat the required solution heating and cooling loads in the coil is used in this study as a reference system for comparing the system performance and energy consumption with a heat pump driven liquid desiccant system.

According to research [2], the operating modes of the liquid desiccant system are determined by considering the thermodynamic outdoor air conditions on the psychrometric chart. The air conditions on the psychrometric chart are separated into four regions, as shown in Figure 2. There are three lines that compare the chart into four regions. Lines 1 and 2 stand for the humidity ratio and the wet bulb temperature lines at the supply air set point, respectively. Line 3 represents the dry bulb temperature line at the supply air set point. Table 1 is the summary of operation modes of A liquid desiccant unit is composed of an absorber, a regenerator, a sensible heat exchanger, a solution heating coil, and a solution cooling coil. In Region A, the outdoor air is primarily dehumidified with the liquid desiccant unit, to make the humidity ratio of the air reach Line 1, as shown in Figure 2. The dehumidified air is sensibly and

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adiabatically cooled by the indoor evaporative cooler and direct evaporative cooler in series, to meet the target supply air condition (i.e., 15 °C and saturation). In Region B, the outdoor air condition on the psychrometric chart falls between Line 1 and Line 2. Since the air in this region has a lower humidity ratio than Line 1, as shown in Figure 2, the liquid desiccant unit is deactivated and bypassed, and the air is directly sent to the indirect evaporative cooler for sensible cooling, until the air condition meets Line 2. Subsequently, the air is adiabatically cooled by the direct evaporative cooler to meet the target supply air condition. In Region C, the outdoor air has lower wet bulb temperature than that at the supply air target condition. In this region, the liquid desiccant and indirect evaporative cooler are turned off and bypassed. Only the direct evaporative cooler is operated to meet the supply air dry bulb temperature setpoint (i.e., 15 °C), while the wet bulb temperature of the supply air may be somewhat lower than the target value. In Region D, the DBT and the WBT of the OA are lower than those at the SA target condition; thus, the LD and the DEC are deactivated and bypassed. Only the IEC is activated to work as a SHE to recover sensible heat from the EA stream for preheating the introduced OA. If the sensible heat reclaimed by the IEC is not sufficient enough to meet the cold deck SA temperature (i.e., 15 °C) or the neutral deck temperature (i.e., 20 °C), the auxiliary HC installed at the EA side is activated.

IV. MATHEMATICAL EXPLANATION AND UNIT MODEL

A. Liquid Desiccant

a. Absorber Model Dehumidification

The performance of a liquid desiccant system can be evaluated based on several standards [25], and the effectiveness of the absorber is the most frequently used performance indicator [26]. Several models for predicting the effectiveness of the absorber have been established with experimental data [25–33] or theoretical analyses [34–37]. In this simulation, the linear regression model suggested by Chung and Luo [25] was selected to determine the dehumidification effectiveness of the absorber. They expressed the dehumidification effectiveness of the absorber as a function of seven operational parameters: mass flow rate of the process air (G_{in}), solution mass flow rate (L_{in}), solution inlet temperature (T_{Lin}), air inlet temperature ($DBT_{G_{in}}$), surface area-to-volume ratio of packing (a), packing height (Z), and vapor pressure difference of the desiccant solution to the vapor pressure of water (X).

$$\epsilon_{abs} = \left(1 - 0.024 \frac{\left(\frac{G_{in}}{L_{in}} \right)^{0.6} \exp\left(1.057 \frac{DBT_{G_{in}}}{T_{Lin}}\right) / (aZ - 0.185 X^{0.638})}{1 - 0.192 \exp\left(\frac{0.615 \frac{DBT_{G_{in}}}{T_{Lin}}}{\pi - 21.498}\right)} \right) - 1$$

In addition to numerous established models, the effectiveness of dehumidification is generally defined as the ratio of the changes in temperature or in humidity ratio of the process air in the absorber, as shown in Equations (2) and (3). Once the dehumidification effectiveness of the absorber is predicted by the Chung and Luo's model (Equation (1)), the temperature and humidity ratio of the process air that leaves the absorber can be obtained from Equations (2) and (3) with the assumption that the ϵ_{abs} , $\epsilon_{abs, w}$, and $\epsilon_{abs, T}$ are identical [4]. The equilibrium humidity ratio ($w_{abs,eq}$) of the desiccant solution in the absorber can be determined by Equation (4) [37]. The vapor pressure at the saturation condition of the desiccant solution in the absorber is determined by Equation (5), suggested by Fumo and Goswami [26], with the model coefficients shown in Table 3. In this research, it was assumed that

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the equilibrium temperature of the desiccant solution in the absorber was identical to the temperature of the strong solution that enters the absorber (i.e., 25 °C)[28].

$$\varepsilon_{abs, w} = W_{air, abs, in} - W_{air, abs, out} / W_{air, abs, in} - W_{abs, eq} \quad -2$$

$$\varepsilon_{abs, T} = T_{air, abs, in} - T_{air, abs, out} / T_{air, abs, in} - T_{abs, eq} \quad -3$$

$$W_{eq} = 0.622(P_s/101.325 - P_s) \quad -4$$

$$P_s = (a_0 + a_1 \cdot T_L + a_2 \cdot T_L^2) + (b_0 + b_1 \cdot T_L + b_2 \cdot T_L^2) \cdot C_L^2 + (c_0 + c_1 \cdot T_L + c_2 \cdot T_L^2) \cdot T_L^2 \quad -5$$

<i>Dehumidification Process</i>	a_0	a_1	a_2
	4.58	-0.159	0.0072
	b_0	b_1	b_3
	-18.38	0.56	-0.019
	c_1	c_2	c_3
	21.31	-0.66	-0.013
<i>Regeneration Process</i>	a_0	a_1	a_2
	16.29	-0.88	0.019
	b_0	b_1	b_3
	74.3	-1.80	-0.018
	c_1	c_2	c_3
	-226.4	7.49	-0.039

Table -1 Coefficients of vapor pressure equations

The mass flow rate of the desiccant solution at each state point of the absorber side shown in Figure 5, can be determined by considering the mass balance expressed by Equations (6)–(8). The solution mass flow rates at the state points P1 and P2 are identical (Equation (6)). The amount of dehumidified moisture from the process air ($m_{bs, moi}$) is determined by Equation (7), and thus, the solution mass flow rate at the state point P3 can be expressed by Equation (8). Based on the mass balance, the concentration of the desiccant solution at each state point of the absorber side can also be expressed by Equations (9) and (10). By considering the energy balance at the absorber side, it is possible to estimate the enthalpy of the desiccant solution at the state point P3 by using Equation (11), under the assumption that the heat generated from the exothermic process of dehumidification is transferred to the desiccant solution leaving the absorber [13].

Mass balance:

$$\dot{m}_{p1, sol} = \dot{m}_{p2, sol} \quad -6$$

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$$\dot{m}_{bs,moi} = \dot{m}_{air,abs,in} (w_{air,abs,in} - w_{air,abs,out}) \quad -7$$

$$\dot{m}_{p3,sol} = \dot{m}_{p2,sol} + \dot{m}_{bs,moi} \quad -8$$

Solution concentration:

$$C_{p1,sol} = C_{p2,sol} \quad -9$$

$$C_{p3,sol} = \dot{m}_{p2,sol} \cdot C_{p2,sol} / \dot{m}_{p3,sol} \quad -10$$

Energy balance:

$$\dot{m}_{p3,sol} \cdot h_{p3,sol} = \dot{m}_{p2,sol} \cdot h_{p2,sol} + \dot{m}_{bs,moi} \cdot h_{fg,abs,moi} \quad -11$$

B. Regeneration Model

The effectiveness of regeneration is defined as the ratio of the changes in temperature or humidity ratio of the regeneration air in the regenerator, as expressed by Equations (12) and (13). The amount of moisture added to the regeneration air ($\dot{m}_{bs,moi}$) is expressed by Equation (14); thus, the humidity ratio variation of the regeneration air in the regenerator can be determined under the assumption that the regeneration rate in the regenerator is identical to the dehumidification rate in the absorber (Equation (15)). Once the humidity ratio of the regeneration air leaving the regenerator ($w_{air,reg,out}$) is determined by Equation (14), it is possible to obtain $\epsilon_{reg,w}$ by using Equation (12). In general, $\epsilon_{reg,w}$ and $\epsilon_{reg,T}$ are identical [4], thus the temperature of the regeneration air leaving the regenerator ($T_{air,reg,out}$) can be predicted from Equation (13). The equilibrium humidity ratio of the solution ($w_{reg,e}$) in Equation (12) is determined by Equations (4) and (5), while the equilibrium temperature of the solution ($T_{reg,e}$) in Equation (13) is the temperature of the desiccant solution that enters the regenerator [28].

$$\epsilon_{reg,w} = \frac{w_{air,reg,out} - w_{air,reg,in}}{w_{reg,e} - w_{air,reg,in}} \quad -12$$

$$\epsilon_{reg,T} = \frac{T_{air,reg,out} - T_{air,reg,in}}{T_{reg,e} - T_{air,reg,in}} \quad -13$$

$$\dot{m}_{reg,moi} = \dot{m}_{air,reg,in} (w_{air,reg,out} - w_{air,reg,in}) \quad -14$$

$$\dot{m}_{bs,moi} = \dot{m}_{reg,moi} \quad -15$$

In terms of the mass balance at the regenerator side, the mass flow rates of the desiccant solution at the state points P4 and P5 (shown in Figure 5) are identical to the solution mass flow rate at P3 (Equation (16)). The mass flow rate of the solution at P6 after regeneration can be expressed by Equation (17).

The concentration of the weak solution at the state points P4 and P5 is identical to the concentration at P3 (Equation (18)), while the concentration of the strong solution at P6 after regeneration is increased, and can be expressed by Equation (19). By considering the energy balance at the regenerator side, the enthalpy of the desiccant solution at the state point P6 can be estimated by Equation (20), under the assumption that the heat required for the endothermic process of regeneration is transferred from the desiccant solution [13].

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Mass balance:

$$\dot{m}_{p4,sol} = \dot{m}_{p5,sol} = \dot{m}_{p3,sol} \quad -16$$

$$\dot{m}_{p6,sol} = \dot{m}_{p5,sol} - \dot{m}_{reg,moi} \quad -17$$

Solution concentration:

$$C_{p3,sol} = C_{p4,sol} = C_{p5,sol} \quad -18$$

$$C_{p6,sol} = \dot{m}_{p5,sol} \cdot C_{p5,sol} / \dot{m}_{p6,sol} \quad -19$$

Energy balance:

$$\dot{m}_{p6,sol} \cdot h_{p6,sol} = \dot{m}_{p5,sol} \cdot h_{p5,sol} - \dot{m}_{reg,moi} \cdot h_{fg,reg,moi} \quad -20$$

V. RESULTS AND VALIDATION

The solar heater without the inverter control can be simulated by using a curve-fit model developed by Jin [38] and implemented in Energy Plus [39]. While this model is for a water-water solar water heater, it was considered as a solution-solution heat pump model in this research because the impact of the heat capacity of the desiccant solution, which is different from that of the water on the heat exchange effectiveness at the evaporator and condenser, was not significant as one may observe in graph 1. In the research conducted so far, there is no available model for predicting the performance of a water-water solar water heater. The most similar model is Jin's water-water curve fit model and Zang's Sensible & Latent model. After examining the model with theoretical analysis, we concluded that the model can be used as a solution-solution heat pump; mechanical relations of fundamental thermodynamic laws and heat transfer correlations are employed in the modeling procedure. To analyze the possibility of the model usage as a solution based heat pump, the heat exchange effectiveness in the evaporator and condenser was examined. Using the theoretical equations depicting the effectiveness of the evaporator and condenser, two cases of evaporator effectiveness when using water and solution have been compared by adjusting number of transfer units (NTU) value from 0.1 to 5, and the typical NTU values used in the study (0.13 to 2.05) are within the range. Results are shown in Figure 6 and it was found that there was not much difference between the two. Similar results were also found at the condenser side.

System	OA			LD	IEC			DEC			
	T (°C)	ω (kg/kg)	Eff. (-)		T (°C)	ω (kg/kg)	Eff. (-)	T (°C)	ω (kg/kg)	Eff. (-)	T (°C)
Conventional System	33.7	0.0166	0.87	30.63	0.0085	0.75	21.69	0.0085	0.95	15.87	0.0109
Solar Powered	33.7	0.0166	0.84	26.75	0.0074	0.75	18.72	0.0074	0.95	13.49	0.0099

Table -4 Results comparison of both conventional and solar powered system

The solar powered liquid desiccant system is more effective than the conventional liquid desiccant system either powered by heat pump or gasifier.

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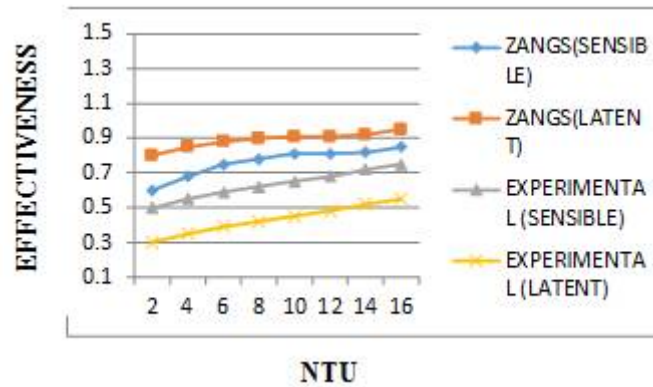


Figure.2 Graph Effectiveness Vs NTU

VII. NOMENCLATURE

ϵ efficiency

ρ density (kg/m³)

Subscripts

p1 – p6 designated points in liquid desiccant unit

e equilibrium

moi moisture

abs absorber

reg regenerator

sol desiccant solution

Greek Symbols

Δ difference

ϵ efficiency

ρ density (kg/m³)

Subscripts

p1 – p6 designated points in liquid desiccant unit

e equilibrium

moi moisture

abs absorber

reg regenerator

sol desiccant solution

in inlet

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out outlet
load load side
source source side
c cooling mode
re f reference

Abbreviations

ASHRAE American society of heating, refrigerating, and air conditioning engineers COP coefficient of performance

DBT dry bulb temperature

WBT wet bulb temperature

HP heat pump

HPLD heat pump driven liquid desiccant

LD liquid desiccant

LiCl lithium chloride

LD-IDECOAS liquid desiccant and evaporative cooling-assisted 100% outdoor air system

HPLD-IDECOAS heat pump driven liquid desiccant and evaporative cooling-assisted 100% outdoor air system

NTU number of transfer units

SHE sensible heat exchanger

PLR part load ratio

Freq frequency

OA outdoor air

SA supply air

RA return air

DEC direct evaporative cooler

IEC indirect evaporative cooler

HC heating coil

CFD Computational Fluid Dynamics

VIII. FUTURE SCOPE

A CFD (Computational Fluid Dynamics) analysis can be used to solve the engineering problems. The following conclusions can be drawn from analysis. The validation of the CFD method developed for successful system. Comparison of the larger the air mass flow rate, the smaller the decrease in the humidity ratio. Study of reduction in the concentration of desiccant film strengthening the dehumidification process of the air.

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XI. CONCLUSION

In this research, it was found that the solar powered liquid desiccant dehumidification system was able to provide a 33% reduction in annual primary energy consumption, compared with the conventional gasifier liquid desiccant dehumidification system previously recommended in the literature. The significant energy saving potential expected in the proposed system is mainly obtained through the reduction in the gas consumption (i.e., 83%) required for regeneration of the weak desiccant solution, which is accomplished by means of the heat pump in the liquid desiccant section of the system. In fact, the solar powered system was intended to realize an on-vapor-compression based air handling system. However, for successful deployment of the system in the relevant markets, the energy consumption for regeneration of the desiccant solution should be reduced as much as possible. As shown in this research, a solar system applied in the liquid desiccant system is an adequate solution for the relatively high regeneration energy consumption observed in the gasifier or heat pump desiccant system. By reclaiming sensible heat from the desiccant solution at the absorber side, and providing the recovered heat to the regenerator side using the solar system, the solar power desiccant system could enhance energy saving potential, and would be a successful renovation of the Conventional liquid desiccant system.

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