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Performance Enhancement of The Liquid Desiccant Dehumidification System

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Abstract **The latent thermal load contributes plays a vital role in controlling the indoor parameters and consumes about half of the total energy in maintaining the thermal comfort for the occupants. Scientist and researcher are continuously looking for a sustainable solution which not only helps in reducing energy consumption but also environmentally friendly. One of the efficient methods adopted in this regard is the liquid desiccant dehumidification (LDD). In this paper, the thermodynamic model of containing two LDD systems; with and without regeneration system is first developed using waste heat as a source. Then the performance of each system is compared on the basis of their humidity ratio, relative humidity, temperature of supply air and condensation rate in the dehumidifier. According to results, it has found that the overall COP with and without regeneration system reaches to 1.165 and 0.566, respectively.**

***Keywords*: Dehumidification, Desiccant, Moisture Desorber, Lithium Bromide, Regeneration,**

I.INTRODUCTION

Nowadays the use of heating, ventilation and air conditioning (HVAC) is increasing due to the higher standard of living and climate change. However, 40% of total global energy is used in buildings[1]. According to researchers HVAC system frequently consumes most of the energy use in large buildings. The main function of the HVAC system to remove the air conditioning load to achieve required temperature. Thermal comfort is the state of mind that expresses contentment with the environment and is assessed by subjective evaluation. The air conditioning load is comprised of two loads sensible (air temperature) and latent load (relative humidity), energy consumed by both load is 20 to 40% of overall energy consumption in building respectively[2]. The ideal humidity ranges between 45-55% for thermal comfort and temperature range 20-25℃[3]. The focus of researchers is increasing day by day to decrease the energy consumption and greenhouse gas emissions (GHG) related to the use of mechanical (HVAC) systems. 40% of the HVAC energy consumed in the latent load of the building. By using a desiccant assisted air AC system about 40% of energy will be saved as compared to vapor compression cooling (VCC) system[4]. Therefore, liquid desiccant based air conditioning systems have been put forward as alternatives to the solid desiccant, vapor absorption, and convention vapor compression based air conditioning systems to control the latent load[5, 6], particularly in humid and warm climates. Many other researchers have investigated the physical properties dehumidification, and regeneration performance of some LDs including $CaCl_2$, LiBr, and LiCl aqueous solution, liquid desiccant $CaCl_2$ is the economical while has not good absorption ability[7, 8]. Low-quality waste heat is the driving source of the LD-based AC device, besides LD can engross some amount of noxious waste from processed air. The working principle of this system as follows: The LD solution after freezing is sprayed at the uppermost of the packed tower with the help of an anticorrosive pump and air is supplied with help of blower in the bottom of the dehumidifier. Then the air contacts with the liquid desiccant solution and exchange energy (heat and mass exchange) with the LD solution, the processed air supplied to the room. During the dehumidification process, the air be cooled and dehumidified, the liquid the desiccant solution is heated and water vapors are added. Then the diluted liquid desiccant solution will be heated by low-grade heat then be sprayed at the top of the tower and OFA is supplied at the bottom of the dehumidifier. The LD solution contacts with the OFA and exchange energy (heat and mass exchanger) with another stream of the air. The air is heated and humidified during regeneration, the liquid desiccant solution is cooled and water vapors are extracted.

Many researchers studied the simultaneous mass and heat transfer process that takes place along the LD desorber[9-12] and dehumidifier [12-14] by conducting experimental studies. By employing numerically schemes[15, 16] and dehumidifier [17]. Mass and heat transfer characteristics of the film plate type LD desorber calculated experimentally [12, 18]. Many researchers proposed the hybrid LD based VCC air conditioning system in which condenser waste heat of is used to regenerate the LD in the desorber, cooling effect is used in dehumidified and the cool the air[19-21]. Some proposed hybrid LD base vapor absorption cycle with the use of solar energy[13, 22, 23]. This paper presents to influences of using regenerators (R-I and R-II) on performance parameters of a hybrid vapor absorption cycle (VAC) based (LDD) system. The liquid desiccant in desorber of the LLD system and VAC are regenerated by the hot flue gases. The cooling of VAC is used in the dehumidifier of the LLD system. Compared with the results from literature experimentally and mathematically[15, 17, 24].

II.SYSTEM DESCRIPTION.

The system includes air loops, liquid loops, and hot flue gases loop. Two air loops are dehumidified air and Moisture Desorber

as shown in Fig. 1. The three liquid loops are liquid desiccant, chilled water, and heat transfer fluid (water) loop. The last loop is hot flue gases. As a result of high performance and stability, the lithium chloride was selected[8].
Air loop, the return/fresh air (inlet air) (0) is pumped into the bottom of the dehumidifier and openly constants the cold and strong LD solution (3) in counter flow mode. For the dehumidifier is randomly packed type bed is selected because of its density in terms of huge contact surface area per volume [25]. After processing the air (14) enters the room.
LD solution (4), after absorbs moisture and heat from the fresh air, turn into weaker and essentials to regenerate. The LD solution (4) and (5) is heated with/without heat Regenerator R-I and Regenerator R-II with warm and strong LD solution and outlet air of regenerated to use the waste of air (12) respectively. Then desiccant solution (6) enters the heat exchanger (HX-II) heated at the high temperature of hot flue gasses (8). The fresh air (0) enters in moisture desorber to absorb the water vapor from hot and weak the desiccant solution (7). For the desorber is randomly packed type bed is selected because of its density in terms of huge contact surface area per volume[25]. Processed air enters in heat exchanger R-II to heat the weak LD solution.
Beforehand incoming the dehumidifier (3) and finishing the cycle, the strong LD leaving the dehumidifier (1) is air-conditioned by chilled water (10) in a compact heat exchanger. Another compact regenerator (R-II) exchanges heat between the weak (1, 2) and strong (4, 5) solutions. The regenerator (R-II) is used to heat the weak LD (5, 6) with the proceed air of moisture desorber (12,13). When entering the dehumidifier, this will precool the strong desiccant and simultaneously preheat the weak desiccant that enters the regenerator. The friction of hot flue gases (15) enters the compact heat exchanger to heat the HTF (9) after exchange heat hot flue gasses (16). The remaining amount of hot flue gases enter in vapor absorption regenerator and Cooling water (11) enters the evaporator of the vapor absorption cycle to exchange the heat. Moreover, in entire system the vapor absorption cycle provides the cooling water when needed.

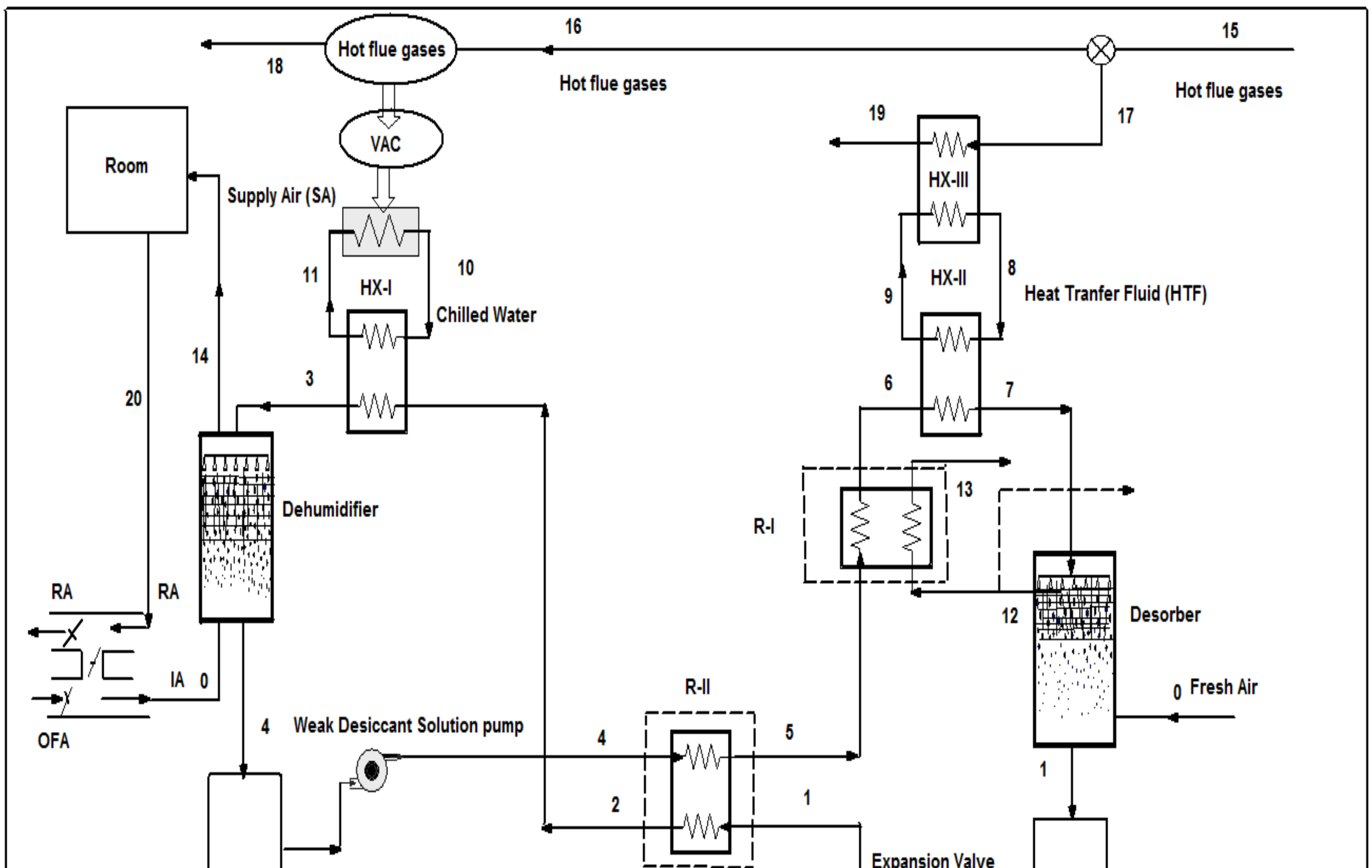

Fig. 1 The schematic diagram

III. THERMODYNAMIC MODELING.

The thermodynamic modeling of the system is depend on mass and energy balances with the assumptions as following.

A. Assumptions.

- Steady-state energy and mass balances-based analysis.
- The input of the work of pump and increment in temperature during pumping and blower is negligible.
- The air is considered is an ideal gas in this study so it ideal gas properties are constant.
- The desorber and dehumidifier are well insulated and work adiabatic conditions.
- The effectiveness of the regenerators, heat exchangers, dehumidifier, desorber and COP of the vapor absorption cycle are constant.
- The latent heat of vaporization of LD is constant.

B. Component wise model equations.

Dehumidifier.

The idea of performance review of the dehumidifier is taken from [17]. The energy equation of the dehumidifier is written below ,

$$\dot{m}_{a,0}\{c_{p,a}(T_0 - T_{14}) + \frac{0.622\times h_{fg}}{P_{atm}}(p_0 - p_{14})\} = \dot{m}_{d,3}c_{p,a}(T_4 - T_3) \quad (1)$$

At the inlet of the dehumidifier, the vapour pressure of the cold and strong LD solution is greater than the hot and weak LD solution at the outlet of the dehumidifier. The vapor pressure of the LD solution should be lesser than the vapor pressure in the air. The moisture removal effectiveness or dimensionless

Fig. 2 The schematic diagram

moisture difference of the dehumidifier is defined as[14, 17, 25].

$$\alpha_{DH} = \frac{\omega_{14}-\omega_0}{\omega_3-\omega_0} = \frac{p_{14}-p_0}{p_3-p_0} \quad (2)$$

The vapor pressure in the air p_0.

$$\omega_0 = 0.622\frac{p_0}{P_{atm}-p_0} \quad (3)$$

The partial pressure of desiccant solution inlet desiccant solution (lithium chloride, LiCl) in the dehumidifier is given by the second-order polynomial equation in terms of temperature and concentration [26]. To make the effective mass transfer in dehumidifier the value of should be lower as possible so that after regeneration hot and strong desiccant solution passing through heat exchangers.

$$p_3 = \left(k_0T_3 + k_1T_3 + k_2{T_3}^2\right) + \left(l_0T_3 + l_1T_3 + l_2{T_3}^2\right)\mathcal{E}_3 + \left(m_0T_3 + m_1T_3 + m_2{T_3}^2\right){\mathcal{E}^2}_3 \quad (4)$$

The values of constants for the dehumidification process are:
k_0=4.582080, k_1=-0.1591740, k_2=0.00725940,
l_0=-18.38160, l_1=0.56610, l_2 =-0.0193140
m_0=21.3120, m_1=0.6660, m_2=0.013320,
and for Regeneration:
k_0=16.2940, k_1=-0.88930, k_2=0.019270,
l_0=74.30, l_1=0.56610, l_2 =-0.01875
m_0=226.40, m_1=-7.490, m_2=-0.0390,
value of vapor pressure of inlet LD solution should be lesser than the vapor pressure in the inlet air because of vapor pressure difference between both fluids the mass transfer from the IA to the LD solution. So that the vapor pressure in the supply air is always lower than the vapor pressure of water inlet air [27] and the value moisture removal effectiveness is between one and zero.

The temperature difference ratio is defined per the vapor pressure difference ratio. Eq. (2) can be written as.

$$\beta_{DH} = \frac{T_0-T_{14}}{T_0-T_3} \quad (5)$$

The inlet temperature of the cool and strong LD solution is probable to be lesser than the value of the temperature of inlet air due to temperature difference between both fluids the heat transfer will happen due to contact of fluids. So that temperature of supply air is probable lower than the temperature of inlet air[27] and the value of is between one and zero.

The rate of Moisture removal or condensation rate of water in the dehumidifier from the inlet air is given as:

$$\dot{m}_{cond}/A = \dot{m}_{a,0}(\omega_0 - \omega_{14}) \quad (6)$$

And the flow rate of air after releasing the moisture is given by:

$$\dot{m}_{a,14} = \dot{m}_{a,0} - \dot{m}_{cond} \quad (7)$$

The supply air humidity ratio can be calculated as follow.

$$\omega_{14} = 0.622\frac{p_{14}}{P_{atm}-p_{14}} \quad (8)$$

The relationship between inlet and outlet concentration at the dehumidifier can be given as:

$$\dot{m}_{d,3}\mathcal{E}_3 = (\dot{m}_{d,3} - \dot{m}_{cond})\mathcal{E}_4 \quad (9)$$

After absorbing the moisture from the dehumidifier's inlet air, the outlet concentration of poor desiccant solution dehumidifier can be identified as:

$$\dot{m}_{d,4} = \dot{m}_{d,3} + \dot{m}_{cond} \quad (10)$$

C. Moisture desorber (MD).

The idea of regenerator performance has taken from[15]. According to numbers in the system in Fig. 1, the energy balance of the desorber.

$$\dot{m}_{a,0}\{c_{p,a}(T_{12} - T_0) + \frac{0.622\times h_{fg}}{P_{atm}}(p_{12} - p_0)\} = m_{d,6}c_{p,a}(T_7 - T_1) \quad (11)$$

At the outlet of the desorber, the vapor pressure of the strong and hot air desiccant solution is lesser than that of the weak and hot desiccant solution at the entrance. The vapor pressure of the desiccant solution must be greater than the partial pressure of water vapor in the air. The moisture removal effectiveness or dimensionless moisture difference of the dehumidifier is defined as.

$$\alpha_{MD} = \frac{\omega_0-\omega_{12}}{\omega_0-\omega_7} = \frac{p_0-p_{12}}{p_0-p_7} \quad (12)$$

The vapor pressure of inlet desiccant solution at the regenerator is the same as given in Eq. (5) where for regeneration the temperature, concentration, and constants are given below [32]. For the effective mass transfer in regenerator the value of should be higher as possible so that after dehumidification the cold and weak desiccant solution passing through heat exchangers.

The partial pressure of water vapor in the inlet air is must be lower than the partial pressure of inlet desiccant solution due to partial pressure between both fluids the mass transfer will occur due to contact of fluids. So that the partial pressure of water vapor in the supply air is always higher than the partial pressure of water inlet air [33] and the value of is between one and zero. According to the vapor pressure difference ratio, the temperature difference ratio is defined as:

$$\beta_{DH} = \frac{T_0 - T_{12}}{T_0 - T_7} \quad (13)$$

The inlet temperature of the weak and warm desiccant solution is likely to be higher than the temperature of inlet air due to temperature difference between both fluids the heat transfer will occur due to contact of fluids. So that the temperature of the supply outlet is expected higher than the temperature of inlet air [28]and the value of is between one and zero.

The moisture evaporation rate [kg/m^2s] of water from the inlet air in the Regenerator is given as:

$$\frac{\dot{m}_{evap}}{A} = \dot{m}_{a,0}(\omega_{12} - \omega_0) \quad (14)$$

And the flow rate of air after releasing the moisture is given by:

$$\dot{m}_{a,12} = \dot{m}_{a,0} + m_{evap} \quad (15)$$

The relationship between inlet and outlet concentration at the desorber can be given as:

$$\dot{m}_{d,6}\mathcal{E}_6 = (\dot{m}_{d,6} + \dot{m}_{evap})\mathcal{E}_1 \quad (16)$$

After absorbing the moisture from the entering air of dehumidifier, the outlet concentration of weak desiccant solution dehumidifier $\mathcal{E}_4$ can be found, strong desiccant becomes weaker and its flow rate is given by:

$$\dot{m}_{d,1} = \dot{m}_{d,6} - \dot{m}_{evap} \quad (17)$$

D. Regenerator (R-I).

The regenerator (R-I) is used to cool the strong desiccant solution which is coming from the regenerator by transferring heat to the weak desiccant solution which is entering from the dehumidifier. On one side, the weak desiccant solution helps to precool the strong desiccant solution before going to R-I and Dehumidifier. On the other side, the weak desiccant solution before going to R-I preheated by the help of the strong desiccant solution. The effectiveness of the R-1 given.

$$\text{Effectiveness} = \frac{Q_{act}}{Q_{max}} \quad (18)$$

Where

$$Q_{max} = C_{min}(T_{hot,in} - T_{cold,in}) \quad (19)$$

Using the above concept, the effectiveness in terms of temperature.

$$Q_{actual} = C_{cold,1}(T_5 - T_4) \quad (20)$$

$$Q_{actual} = C_{hot,1}(T_1 - T_2) \quad (21)$$

$$Q_{max} = C_{hot,1}(T_1 - T_4) \quad (22)$$

Substitute the Eq. 21. And Eq.22. In Eq. 18. We

$$Effectiveness_{reg1} = \frac{T_1 - T_2}{T_1 - T_4} \quad (23)$$

Mass balancing.

The total mass of hot fluid at inlet = Total mass of cold fluid

$$\dot{m}_{d,4} = \dot{m}_{d,5}, \text{ and } \dot{m}_{d,1} = \dot{m}_{d,2} \quad (24)$$

E. Regenerator (R-II).

The regenerator (R-II) preheats the weak desiccant solution which is coming from heat (R-I) before going to regenerator with the help of hot air coming from moisture desorber. The temperature of the waste heat exchanger which is coming from (R-I), greater than the temperature of the desiccant solution. R-II is used the waste of moisture desorber air. the same concept R-1 is used to find the outlet temperatures of fluids and actual heat transfer fluids and actual heat transfer.

F. The heat exchanger (HX-I).

The heat exchanger (HX-I) is used to cool the strong desiccant solution coming from (R-I) before it enters the dehumidifier. According to eq. 4, the vapor pressure of desiccant is the function of temperature, so for better performance of the dehumidifier, the temperature and vapor pressure of desiccant should be lower than the inlet air. So for the cooling purpose of strong desiccant the cooled water is used. To find the outlet temperatures and actual heat transfer the same concept (R-I) is used.

G. The heat exchanger (HX-II).

The heat exchanger (HX-II) is used to heat the weak desiccant solution coming from (R-II) before it enters the moisture desorber. According to eq. 5, the vapor pressure of desiccant is a function of temperature, so for better performance of the desorber, the vapor pressure and temperature of strong desiccant solution should be higher than the inlet air so that the heating purpose of strong desiccant i.e. the hot water is used. To find the outlet temperatures and actual heat transfer the same concept (R-I) is used.

H. The heat exchanger (HX-III).

The heat exchanger (HX-III) is used to heat HTF (water) which is used in HX-II to heat the weak desiccant solution coming from R-II before it enters the moisture desorber. The HTF is heated with the hot flue gases. To find the outlet temperatures and actual heat transfer the same procedure of R-I is used.

I. Vapor Absorption Cycle (VAC).

The vapor absorption cycle cools the cooling water which is used to in (HX-I) to cool the strong desiccant solution before enters in the dehumidifier. To find the heat input the concept of COP is used.

J. The COP overall system.

The concept of COP overall is taken from[8].

$$COP_{overall} = \frac{Q_a}{Q_{total}} \text{ where} \quad (25)$$

$$Q_a = m_{a,0}\{c_{p,a}(T_0 - T_{14}) + \frac{0.622 \times h_{fg}}{P_{atm}}(p_0 - p_{14})\} \quad (26)$$

$$Q\_total = Q\_REG + Q\_(act, III) \quad (27)$$

K. Model Validation

The simulated results of the thermodynamic model are related with a previously analytical model and experimental research work for the purpose of present model validation. The model is divided into parts of moisture desorber and dehumidifier. So, validated by two different published work.

Table 1 Model validation: Inlet parameters of MD for different cases

Cases No	α_{MD}	β_{MD}	$\dot{m}_a$	T_0	ω_0	T_6	$\dot{m}_{d,3}$	ε_4
1	0.87	0.82	0.83	30.40	0.0183	65.0	6.46	34.0
2	0.71	0.87	1.43	29.80	0.0177	65.10	6.47	34.5
3	0.78	0.75	1.10	40.0	0.0178	65.0	6.35	33.6

Table 2 Model validation: Compression of outlet parameters of MD different cases

Cases No		T_0	ω_{12}	T_1	ε_1	$\dot{m}_{evap}$
01	Experimental	58.90	0.0579	58.60	34.50	1.55
	Simulated	N/A	N/A	59.69	34.15	1.33
	Present model	58.91	0.0579	59.04	34.17	1.492
	Error (%)	0.017	0	0.748	0.9482	3.798
02	Experimental	57.50	0.0488	56.60	35.20	2.10
	Simulated	N/A	N/A	57.40	34.72	1.82
	Present model	57.51	0.04876	56.49	34.74	1.022
	Error (%)	0.018	0.04458	0.1945	1.326	3.797
03	Experimental	58.90	0.0548	57.60	34.20	1.91
	Simulated	N/A	N/A	58.79	33.79	1.71
	Present model	58.90	0.05477	58.03	33.82	1.843
	Error (%)	0.00	0.05196	0.7438	1.127	3.562

Table 3 Model validation: Inlet parameters of dehumidifier for different cases

Cases No	α_{DH}	β_{DH}	$\dot{m}_a$	T_0	ω_0	T_3	$\dot{m}_{d,3}$	ε_3
1	0.80	10.50	1.18	30.1	0.0181	30.30	6.22	34.7
2	0.79	0.52	1.18	35.5	0.0188	30.30	6.29	34.5
3	0.81	-4.00	1.21	30.3	0.0142	30.10	6.27	33.9

Table 4 Model validation: Compression of outlet parameters of dehumidifier for different cases

Case No		T_0	ω_{14}	T_4	ε_4	$\dot{m}_{cond}$
1	Experimental	32.20	0.0108	32.60	34.60	0.40
	Simulated	N/A	N/A	31.35	34.65	0.372
	Present model	32.20	0.0108	31.44	34.17	0.3899
	Error (%)	0.00	0.00	3.623	1.251	2.557
2	Experimental	32.80	0.0112	32.60	33.70	0.42
	Simulated	N/A	N/A	31.73	34.45	0.389
	Present model	32.80	0.01117	31.86	34.74	0.4103
	Error (%)	0.00	0.13410	2.296	3.039	2.337
3	Experimental	31.10	0.0103	31.50	34.80	0.23
	Simulated	N/A	N/A	30.70	33.88	0.206
	Present model	31.10	0.0103	30.75	33.82	0.2143
	Error (%)	0.00	0.00	2.410	0.059	7.067

IV. Results and Discussion.

A. *Base value parameters.*

Table 5 Input variables

Desiccant concentration	23% by mass
Airflow rate	1.27kg/m^2s
Desiccant flow rate	6.227 kg/m^2s
α	0.72
β	0.72
IAT	33°C
IARH	80%
Diameter	24cm
Effectiveness R-I	0.6
Effectiveness R-II	0.89
Effectiveness HX-I, II&III	0.6
HTF flow rate	5.00 kg/s
Hot flue gases flow rate in HX-II	0.17 kg/s
Hot flue gases temperature	210°C
Chilled water mass flow rate	1.0 kg/s
Hot flue gases flow rate in HX-I	0.87 kg/s
Chilled water temperature	20 °C
COP	0.7

Fig. 3 shows the outcomes of the supply air temperature, humidity ratio, and relative humidity with and without regenerator (R-I & II) with the input of base value in Tab. The supply air humidity ratio with R-I & R-II is lower than without R-I & R-II. This is because of the moisture removal effectiveness and depends upon the partial pressure difference between inlet air (IA) and desiccant solution so that the partial pressure of the LD solution is decreased with the R-I&R-II. Supply air temperature with R-I& R-II is lower than the without R-I & R-II. This is due to the temperature difference ratio dependent upon the inlet temperature of AI and desiccant solution. So that the inlet temperature of the solution is decreased with R-I & R-II.

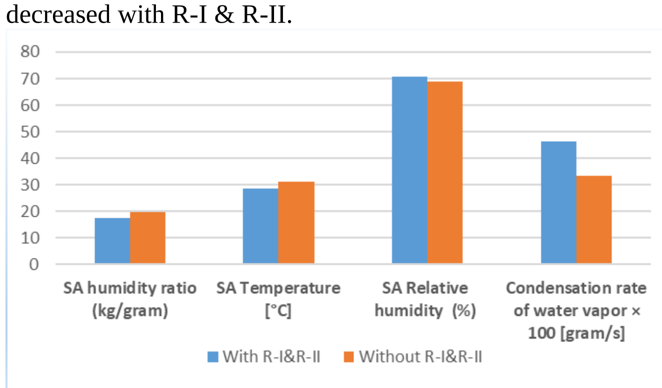

Fig. 3 Comparison of a performance parameter of both systems.

The supply air humidity ratio with R-I & R-II is slightly higher than without R-I & R-II. This due to the relative humidity is a function of supply air humidity ratio and temperature. The supply air temperature with R-I& R-II is lower than the without

R-I & R-II so that the supply air relative humidity will be increased with R-I& R-II.

The condensation rate of water vapor with R-I & R-II is higher than without R-I & R-II. This is due to the condensation of water vapor is depending upon the difference between the humidity ratio of IA and the supply air. So that the difference is increased by using R-I& R-II.

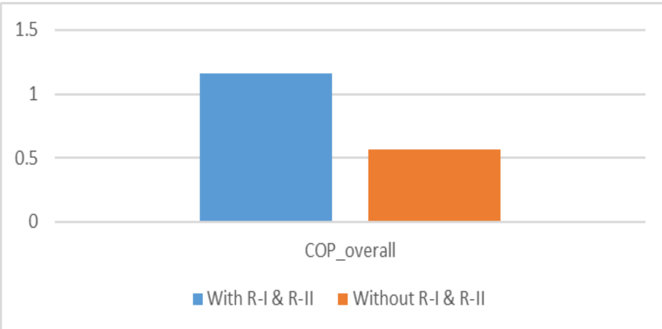

Fig. 4 Comparison of COP overall of both systems

Fig. 3 illustrates the results of the COP of the overall system with R-I & R-II and without R-I & R-II. The COP of the overall system with R-I & R-II is higher than without R-I & R-II. This due to the use of R-I and R-II heat is recovered.

V.CONCLUSION

The latent thermal load contributes a major role in controlling indoor parameters to obtain thermal comfort for the occupants. It also directly impacts the energy consumption for air-conditioning equipment. One of the efficient methods adopted in this regard is LDD. The aim of this study is to develop the thermodynamic model of two LDD systems; with and without regeneration using waste heat as a source. The performance of each system is compared on the basis relative humidity ratio, relative humidity and temperature of supply air, condensation rate in the dehumidifier and overall coefficient of performance (COP_overall). The relative humidity is 70.86% and 69.03% with and without regeneration with temperature 28.78°C and 31.24°C of supply air respectively and the condensation rate in a dehumidifier is 0.4618 and 0.339 gram/sec with and without regeneration respectively. COP_overall of the system with and without regeneration are 1.165 and 0.566 respectively. Results shows significant impact of regeneration on performance parameters.

VI.ACKNOWLEDGEMENTS

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REFERNCES

1. Yang, L., H. Yan, and J.C. Lam, *Thermal comfort and building energy consumption implications–a review.* Applied energy, 2014. **115**: p. 164-173.
2. Li, Z., et al., *New type of fresh air processor with liquid desiccant total heat recovery*. Energy and Buildings, 2005. **37**(6): p. 587-593.
3. American Society of Heating, R. and A.-C. Engineers, *ASHRAE Standard: Thermal Environmental Conditions for Human Occupancy*. 2010: ASHRAE.
4. Potnis, S.V. and T.G. Lenz, *Dimensionless mass-transfer correlations for packed-bed liquid-desiccant contactors.* Industrial & engineering chemistry research, 1996. **35**(11): p. 4185-4193.
5. Killion, J.D. and S. Garimella, *A critical review of models of coupled heat and mass transfer in falling-film absorption.* International Journal of refrigeration, 2001. **24**(8): p. 755-797.
6. Naik, B.K., et al., *Coupled heat and mass transfer analysis of an adiabatic dehumidifier–unique approach.* Energy Procedia, 2016. **90**: p. 305-315.
7. Conde, M.R., *Properties of aqueous solutions of lithium and calcium chlorides: formulations for use in air conditioning equipment design.* International Journal of Thermal Sciences, 2004. **43**(4): p. 367-382.
8. Liu, X., X. Yi, and Y. Jiang, *Mass transfer performance comparison of two commonly used liquid desiccants: LiBr and LiCl aqueous solutions.* Energy Conversion and management, 2011. **52**(1): p. 180-190.
9. Elsarrag, E., *Performance study on a structured packed liquid desiccant regenerator.* Solar Energy, 2006. **80**(12): p. 1624-1631.
10. Yin, Y., et al., *Experimental study on a new internally cooled/heated dehumidifier/regenerator of liquid desiccant systems.* International Journal of Refrigeration, 2008. **31**(5): p. 857-866.
11. Martin, V. and D. Goswami, *Heat and mass transfer in packed bed liquid desiccant regenerators—an experimental investigation.* 1999.
12. Katejanekarn, T., S. Chirarattananon, and S. Kumar, *An experimental study of a solar-regenerated liquid desiccant ventilation pre-conditioning system.* Solar Energy, 2009. **83**(6): p. 920-933.
13. Öberg, V. and D. Yogi Goswami, *Performance simulation of solar hybrid liquid desiccant cooling for ventilation air preconditioning.* Solar Engineering, 1998: p. 176-182.
14. Abdul-Wahab, S., Y. Zurigat, and M. Abu-Arabi, *Predictions of moisture removal rate and dehumidification effectiveness for structured liquid desiccant air dehumidifier*. Energy, 2004. **29**(1): p. 19-34.
15. Gandhidasan, P., *Quick performance prediction of liquid desiccant regeneration in a packed bed.* Solar Energy, 2005. **79**(1): p. 47-55.
16. Babakhani, D. and M. Soleymani, *Simplified analysis of heat and mass transfer model in liquid desiccant regeneration process.* Journal of the Taiwan Institute of Chemical Engineers, 2010. **41**(3): p. 259-267.
17. Gandhidasan, P., *A simplified model for air dehumidification with liquid desiccant.* Solar energy, 2004. **76**(4): p. 409-416.
18. Jain, S., P. Dhar, and S. Kaushik, *Experimental studies on the dehumidifier and regenerator of a liquid desiccant cooling system.* Applied thermal engineering, 2000. **20**(3): p. 253-267.

19. Al-Farayedhi, A., et al. *Experimental study of hybrid liquid desiccant based vapor compression cooling system*. in *The 6th Saudi Engineering Conference, KFUPM, Dhahran, December 2002*. 2002.
20. Khalil, A., *An experimental study on multi-purpose desiccant integrated vapor-compression air-conditioning system.* International journal of energy research, 2012. **36**(4): p. 535-544.
21. Yamaguchi, S., et al., *Hybrid liquid desiccant air-conditioning system: Experiments and simulations.* Applied Thermal Engineering, 2011. **31**(17-18): p. 3741-3747.
22. Zeidan, E.-S.B., A.A. Aly, and A.M. Hamed, *Modeling and simulation of solar-powered liquid desiccant regenerator for open absorption cooling cycle.* Solar Energy, 2011. **85**(11): p. 2977-2986.
23. Ahmed, C.K., P. Gandhidasan, and A. Al-Farayedhi, *Simulation of a hybrid liquid desiccant based air-conditioning system.* Applied Thermal Engineering, 1997. **17**(2): p. 125-134.
24. Fumo, N. and D. Goswami, *Study of an aqueous lithium chloride desiccant system: air dehumidification and desiccant regeneration.* Solar energy, 2002. **72**(4): p. 351-361.
25. Martin, V. and D.Y. Goswami, *Effectiveness of heat and mass transfer processes in a packed bed liquid desiccant dehumidifier/regenerator.* Hvac&R Research, 2000. **6**(1): p. 21-39.
26. Uemura, T., *Studies on the lithium chloride-water absorption refrigeration machine.* Technology reports of Kansai university, 1967. **9**: p. 71-84.
27. Chung, T.-W., T.K. Ghosh, and A.L. Hines, *Dehumidification of air by aqueous lithium chloride in a packed column.* Separation Science and Technology, 1993. **28**(1-3): p. 533-550.