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Gurubalan A
Indian Institute of Technology Madras, Chennai, India, gurumariner60@gmail.com

Maiya M P
IIT Madras, India, mpmaiya@iitm.ac.in

Shaligram Tiwari
Department of Mechanical Engineering, Indian Institute of Technology Madras, India, shaligt@iitm.ac.in

A Novel Solar-assisted Membrane-based Liquid Desiccant Air Conditioning System for Hot and Humid Climatic Conditions

A. Gurubalan[1], M.P. Maiya[2*], Shaligram Tiwari[3]

[1,2,3] Department of Mechanical Engineering
Indian Institute of Technology Madras, Chennai, India - 600036

Email: [1]gurumariner60@gmail.com, [2]mpmaiya@iitm.ac.in, [3]shaligt@iitm.ac.in
Phone: [2]91-44-22574665, [3]91-44-22574729

* Corresponding Author

ABSTRACT

A novel solar assisted membrane based liquid desiccant air conditioning system is proposed and investigated for the hot and humid climatic conditions. It uses an air cooling heat exchanger after dehumidifier to maintain the desired temperature of the room air. The energy performance of the proposed system is evaluated in terms of room temperature and specific humidity, and coefficient of performance. In addition, the energy saving potential with respect to the conventional air conditioning system and environmental performance are studied. It is found that the proposed system is able to achieve the comfort room conditions (24°C, 0.0097 kg/kgda) and the yearly energy savings is around 62% when compared with the conventional air conditioning system. Moreover, it is estimated that the proposed system will avoid emissions of 1400, 16 and 6 tons/per year of Carbon dioxide, Nitrous oxide and Sulphur dioxide respectively to the ambient.

1. INTRODUCTION

According to the energy statistics of India, the building sector consumes 40% of electricity out of which nearly one third is consumed by heating, ventilation and air conditioning (HVAC) systems (CSE Energy, 2014). Currently, more than 90% of the HVAC systems are of vapor compression refrigeration type which are energy inefficient for humidity control (Goetzler *et al.*, 2014). Therefore, it is desirable to meet the latent load of such HVAC demands using an alternative system which is not only energy efficient, but can also utilize low grade energy sources such as solar or waste heat. The liquid desiccant system is one such prospective alternative which can utilize such sources for its desiccant regeneration. Such systems are classified as direct contact-packed bed system and indirect contact-membrane system. The latter is preferred, to avoid problems associated with the desiccant carryover with air stream.

The membrane based liquid desiccant system is often to be combined with a sensible cooling system for the desiccant and air cooling demands. Such combination is usually called as hybrid membrane based liquid desiccant air conditioning system (MLAC). The vapor compression refrigeration system is used in the present study to meet the cooling demands, since it promotes the use of MLAC for all climatic conditions. Abdel-Salam *et al.* (2014) analyzed the thermal, economic and environmental performances of the solar assisted MLAC. They concluded that the solar thermal system with backup of natural gas boiler has minimum payback period and operating cost than with heat pump as a backup system. Their MLAC performance is based on adiabatic dehumidifier and regenerator. The temperature of the desiccant to the dehumidifier is to be maintained in such a way as to simultaneously cool and dehumidify the air. However, it cannot cool and dehumidify the air to the desired level due to the resistance to heat and mass transfer in the intermediate membrane and the exothermic heat that is generated during absorption. To avoid such difficulty, the present study proposes a novel MLAC, as shown in Fig.1, with an additional air cooling heat exchanger between the dehumidifier and air conditioning room.

Performance analysis of such a novel solar assisted MLAC (SMLAC) is carried out for the hot and humid climatic condition prevailing in the city of Chennai, India. The year round performance of the SMLAC is evaluated in terms of the room temperature and specific humidity, and coefficient of performance. Moreover, the energy saving potential and the expected reduction in the environmental emissions with respect to the conventional air conditioning system is presented. The results of the present study will be useful for the optimum design of SMLAC for hot and humid climatic conditions.

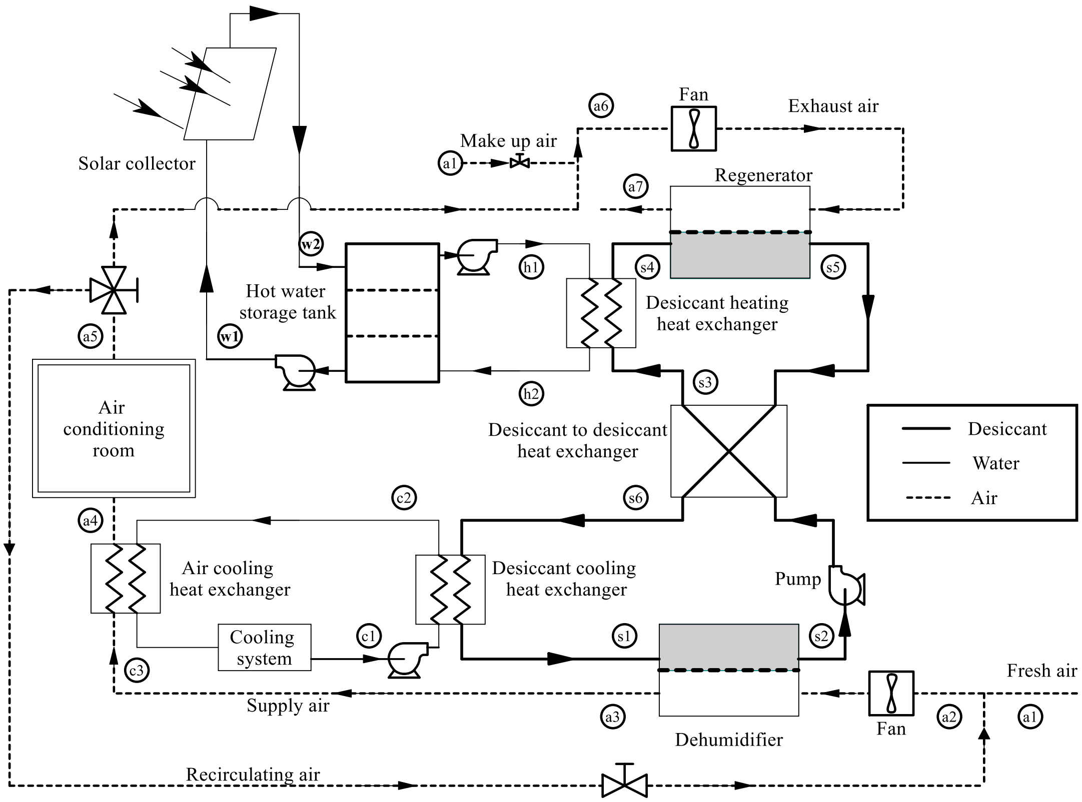

Figure 1: Schematic diagram of solar assisted membrane based liquid desiccant air conditioning system

2. MATHEMATICAL MODELLING

The proposed SMLAC has a number of components, namely dehumidifier, regenerator, heat exchangers, circulating pumps and fans. The default values of the design and operating parameters are listed in Table 1.

Table 1: Default values of the design and operating parameters

Sl. No.	Parameter	Unit	Value
1.	Desiccant type	-	LiCl
2.	Effectiveness of desiccant to desiccant heat exchanger	-	0.8
3.	Effectiveness of desiccant cooling heat exchanger	-	0.8
4.	Effectiveness of desiccant heating heat exchanger	-	0.8
5.	Effectiveness of air cooling heat exchanger	-	0.8
6.	Inlet temperature of cooling water	°C	15
7.	Mass flow rate of air	kg/s	0.735
8.	Mass flow rate of cooling water	kg/s	1.6
9.	Mass flow rate of desiccant	kg/s	0.8
10.	Room sensible load	kW	3
11.	Room latent load	kW	2
12.	Recirculation air mixing ratio (m_{ra}/m_{fa})	-	3:1

2.1 Dehumidifier/ Regenerator

In the proposed SMLAC, the dehumidifiers and regenerators are of flat-plate type with counter flow configuration. The basic working principles of the dehumidifier and regenerator are the same except the direction of water vapor transfer. Therefore, the governing equations and modelling technique are same for both. The design details of the dehumidifier and regenerator are listed in Table 2. The membrane properties are taken from Abdel-Salam *et al.* (2013) and the desiccant properties are calculated from the equations given by Conde (2014). The assumptions for the steady state heat and mass transfer model of the membrane-based dehumidifier have been listed in our previous study (Gurubalan *et al.*, 2017).

Table 2: Design details of the membrane dehumidifier and regenerator

Sl. No.	Parameter	Unit	Value
1.	Air channel thickness, mm	mm	4
2.	Breadth, m	m	0.3
3.	Desiccant channel thickness, mm	mm	3
4.	Length, m	m	1
5.	Number of channels	-	55

2.1.1 Governing equations: The governing equations for mass and energy balance of air and desiccant streams are obtained from Fig.2 and explained as follows.

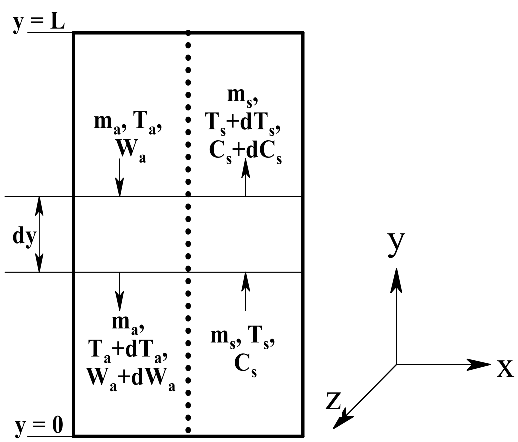

Figure 2: Schematic representation of heat and mass transfer processes in the membrane dehumidifier

Air Side Equations

The mass balance equation on the air side is given as

$$\frac{dW_{\mathrm{a}}^*}{dy^*} = NTU_{\mathrm{M}} \times \left(W_{\mathrm{s}}^* - W_{\mathrm{a}}^*\right) \tag{1}$$

where dimensionless specific humidity $W^* = \dfrac{W - W_{\mathrm{a,in}}}{W_{\mathrm{s,in}} - W_{\mathrm{a,in}}}$

and dimensionless length, $y^* = \dfrac{y}{y_{\mathrm{o}}}$

The number of mass transfer units (NTU_M) is the non-dimensionalized mass transfer area of the dehumidifier, which is given as

$$NTU_{\mathrm{M}} = \frac{U_{\mathrm{M}} \times A_{\mathrm{t}}}{m_{\mathrm{min}}} = \frac{U_{\mathrm{M}} \times A_{\mathrm{t}}}{m_a}$$

The energy balance equation on the air side is given as

$$\frac{dT_{\mathrm{a}}^*}{dy^*} = NTU_{\mathrm{H}} \times \left(T_{\mathrm{s}}^* - T_{\mathrm{a}}^*\right) \tag{2}$$

where dimensionless temperature $T^* = \frac{T - T_{a,in}}{T_{s,in} - T_{a,in}}$

The number of heat transfer units (NTU_H) is the non-dimensionalized heat transfer area of the dehumidifier, which is given as

$$NTU_H = \frac{U_H \times A_t}{\left(m \times C_p\right)_{min}} = \frac{U_H \times A_t}{\left(m \times C_p\right)_a}$$

Desiccant Side Equations

The mass balance equation on the desiccant side is given as

$$\frac{dC_s}{dy^*} = \left(NTU_M \times \left(W_a^* - W_s^*\right) \times m_r \times \left(1 + C_s\right) \times W_o\right) \tag{3}$$

where W_o is the mass transfer potential of the dehumidifier from which the local mass transfer potential at any cross stream location is non-dimensionalized. It is given as

$$W_o = W_{s,in} - W_{a,in}$$

The desiccant mass fraction (C_s) is defined as

$$C_s = \frac{\text{mass of water}}{\text{mass of salt}}$$

where X_s is the desiccant mass concentration which is defined as

$$X_s = \frac{\text{mass of salt}}{\text{mass of solution}} = \frac{1}{1 + C_s}$$

The mass flow ratio (m_r) of air and desiccant in Eqn. 3 is defined as

$$m_r = \frac{m_a}{m_s}$$

The energy balance equation on the desiccant side is given as

$$\frac{dT_s^*}{dy^*} = \left(R_1 \times NTU_H \times \left(T_a^* - T_s^*\right)\right) + \left(R_2 \times NTU_M \times \left(W_a^* - W_s^*\right)\right) \tag{4}$$

where dimensionless heat capacity ratio, $R_1 = \frac{\left(m \times C_p\right)_a}{\left(m \times C_p\right)_s}$ and dimensionless absorption heat ratio

$$R_2 = \frac{m_a \times h_{abs} \times \left(W_{s,in} - W_{a,in}\right)}{\left(m \times C_p\right)_s \times \left(T_{s,in} - T_{a,in}\right)}$$

2.1.2 Overall heat and mass transfer coefficients: The overall heat and mass transfer coefficients are calculated using the individual heat and mass transfer coefficients of air, desiccant and intermediate membrane, as follows.
Overall mass transfer coefficient (U_M) is given as

$$U_M = \left(\frac{1}{h_{D,a}} + \frac{\delta_m}{K_m} + \frac{1}{h_{D,s}}\right)^{-1} \tag{5}$$

Overall heat transfer coefficient (U_H) is given as

$$U_H = \left(\frac{1}{h_{c,a}} + \frac{\delta_m}{k_m} + \frac{1}{h_{c,s}}\right)^{-1} \tag{6}$$

2.1.3 Boundary conditions: The boundary conditions for solving the normalized governing equations, i.e., Eqns.1 to 4, are as follows.

(i) $W_a(y = y_0) = W_{a,in}$; (ii) $T_a(y = y_0) = T_{a,in}$; (iii) $C_s(y = 0) = C_{s,in}$; (iv) $T_s\left(y = 0\right) = T_{s,in}$

2.1.4 Solution methodology: The governing equations for mass and energy balance of air and desiccant streams, i.e., Eqns.1 to 4, are solved using fourth order Runge–Kutta (RK4) and Newton quadratic interpolation technique. A MATLAB computer program was written to carry out the analysis with the dehumidifier height divided into 10^5 segments. The governing equations are solved for each segment from the bottom to the top of the dehumidifier. At the bottom of the dehumidifier, outlet temperature and specific humidity of the air are initially guessed. Then, iterations are performed to guess the temperature and specific humidity of air at the top of the dehumidifier that can give the specified inlet conditions of air.

2.1.5 Validation of the numerical model: The numerical model has to be validated before it is used to evaluate the performance of the proposed SMLAC. The experimental results given by Ge *et al.* (2014) are used for the validation. Figure 3 shows the comparison between the experimental and numerical results and it is observed that the numerical results are in good agreement with the experimental results. Therefore, it is concluded that the developed model reliably evaluate the performances of the dehumidifier and regenerator used in the proposed SMLAC.

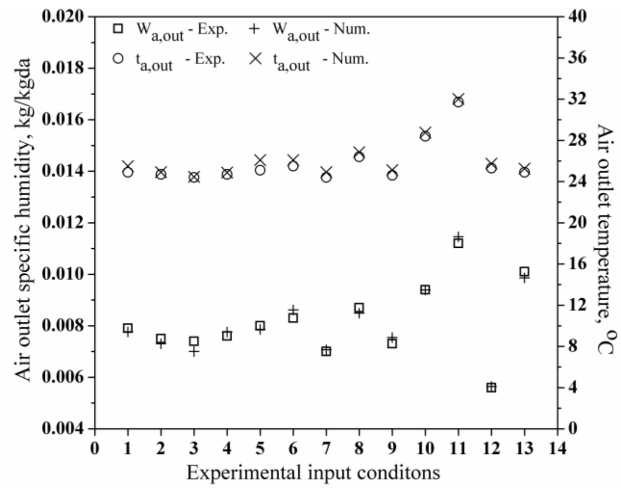

Figure 3: Comparison between the numerical and experimental results (Ge *et al.*, 2014) of membrane dehumidifier

2.2 Cooling System (CS)

Vapor compression refrigeration system (VCR) is used to cool the desiccant and air, and its energy input ($E_{CS,in}$) is calculated as follows.

$$E_{CS,in} = \frac{m_c \times C_{p,c} \times \left(T_{c3} - T_{c1}\right)}{COP_{VCR}} \tag{7}$$

where COP_{VCR} is the coefficient of performance of the VCR which is calculated using the equation given in literature (Abdel-Salam *et al.*,2013) as a function of evaporator temperature. It is assumed that the evaporator temperature is 3°C lower than the cooling water supply temperature (T_{c1}).

2.3 Solar Thermal System

The solar thermal system consists of an evacuated tube type solar collector and a storage tank with aspect ratio (L/D) of 4:1. The equations used for the modeling of the solar thermal system are taken from an earlier study of Saleh and Mosa (2014).

3. RESULTS AND DISCUSSION

3.1 Performance Parameter

The performance of the proposed SMLAC is presented using room temperature and specific humidity and also by the coefficient of performance.

The coefficient of performance (COP) is defined as the ratio of the cooling energy obtained by the supply air to the total cooling energy consumed by SMLAC. It is calculated as follows.

$$\text{COP} = \frac{m_\text{a} \times \left(h_\text{a2} - h_\text{a4}\right)}{E_\text{CS,in}} \tag{8}$$

3.2 Solar Collector Area

Abdel *et al.* (2013) found that MLAC requires hot water supply temperature in the range of 50 - 55°C for its optimum performance. Therefore, the present study calculates the required solar collector area to provide hot water at 55°C. Figure 4 shows the required monthly averaged solar collector area. From Fig.4, it is found that July and January months require minimum (24 m^2) and maximum (39 m^2) solar collector area to provide the hot water at 55°C. This is due to corresponding high and low solar radiation during those months respectively, as shown in Figure 4. However, it is very difficult to vary the solar collector area during the operation of the system. Therefore, an average solar collector area of 30 m^2 is considered for the further performance study. Figure 5 shows the monthly averaged hot water supply temperature when the collector area is taken as 30 m^2. As shown, the hot water supply temperature is found to be maximum (58°C) and minimum (51°C) in the months of July and January respectively due to their corresponding high and low solar radiation than the other months.

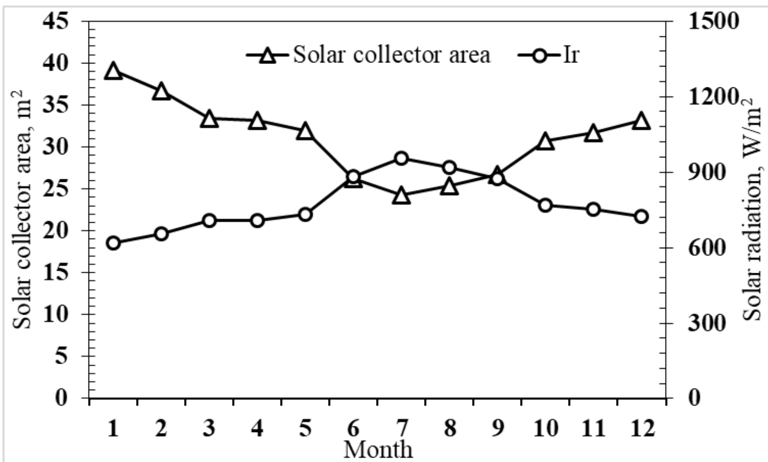

Figure 4: Monthly-averaged required solar collector area

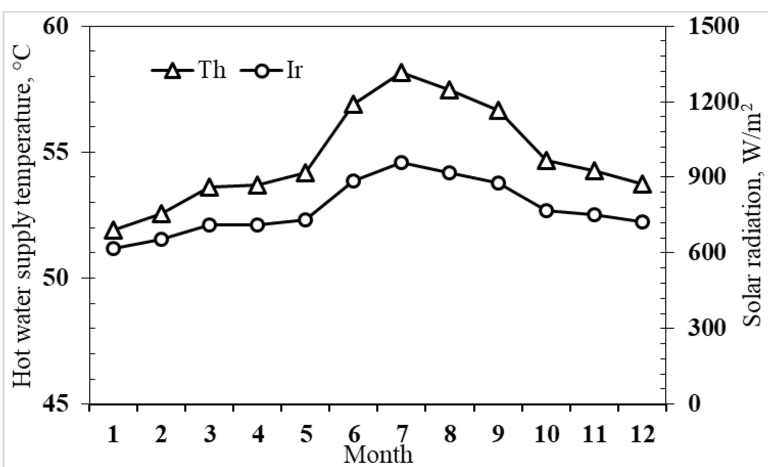

Figure 5: Monthly-averaged hot water supply temperature

3.3 Performance of the Proposed SMLAC

Figure 6 shows the monthly averaged specific humidity of the room and ambient air. As shown, the proposed SMLAC maintains the specific humidity of the room with a minimum fluctuation (0.0079 to 0.0097 kg/kgda) whereas the corresponding fluctuation in the ambient specific humidity is significant (0.018 to 0.026 kg/kgda). In

addition, the proposed SMLAC maintains the specific humidity of the room in the comfort range (0.0097 kg/kgda) specified by ASHRAE for almost all over the year. Moreover, as shown in Fig.6, the specific humidity of the room is found to be minimum during the month of December due to its comparatively lower ambient specific humidity. In addition, the specific humidity of the room during the months of June, July and August is also found to be lower. This is due to high solar radiation during these months which in turn increases the hot water supply temperature to SMLAC. Consequently, more regeneration takes place in the regenerator and thereby, the average concentration of the desiccant in the system increases. This in turn increases the mass transfer potential of the dehumidifier (i.e., the vapor pressure difference between the water vapor in the air and that in the air-desiccant interface) which reduces the specific humidity of the room air in such months.
Figure 7 shows the monthly averaged temperature of the room and ambient air. As shown, the proposed SMLAC maintains the temperature of the room air around 23°C throughout the year even though the ambient temperature varies between 28 to 37°C. This is due to the optimum selection of operating parameters which are listed in Table 2 and the additional cooling of air in the air cooling heat exchanger which further avoids the fluctuation in the temperature of supply air.

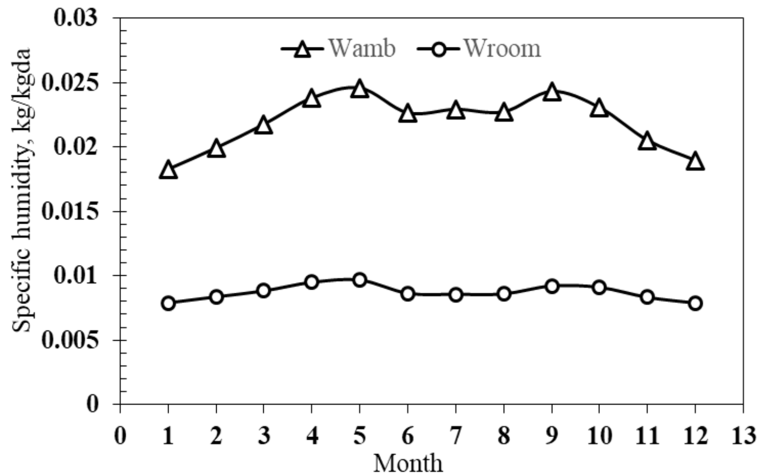

Figure 6: Monthly-averaged specific humidity of the room and ambient air

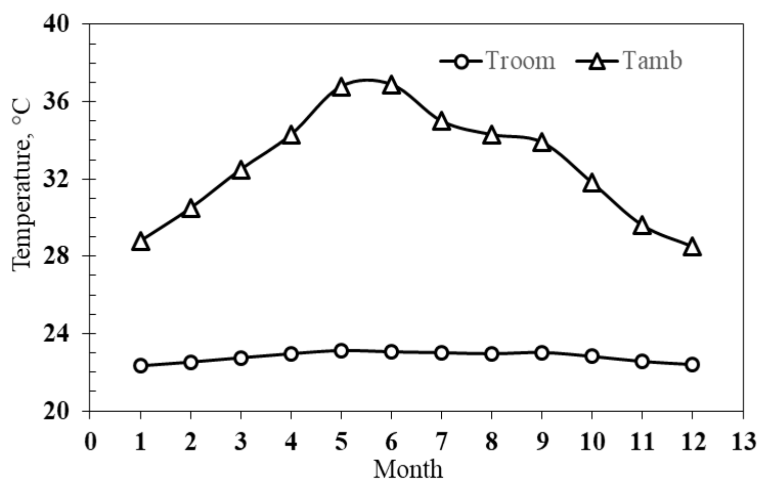

Figure 7: Monthly-averaged temperature of the room and ambient air

3.4 Coefficient of Performance

Figure 8 shows the monthly averaged COP of the proposed SMLAC. As shown, COP is found to be minimum (2.23) during the month of January due to low solar radiation and ambient specific humidity. The low solar radiation reduces the hot water supply temperature and thereby decreases the average concentration of desiccant in the system. This in turn decreases the mass transfer potential of the dehumidifier. In addition, low ambient specific humidity during that month also reduces the mass transfer potential of the dehumidifier. The decrease in the mass transfer potential of the dehumidifier reduces the cooling energy obtained by the supply air and therefore, COP is found to be lowest during the month of January, as shown in Fig.8. As illustrated, COP is found to be maximum during June which is due to comparatively higher solar radiation and ambient specific humidity than the other months.

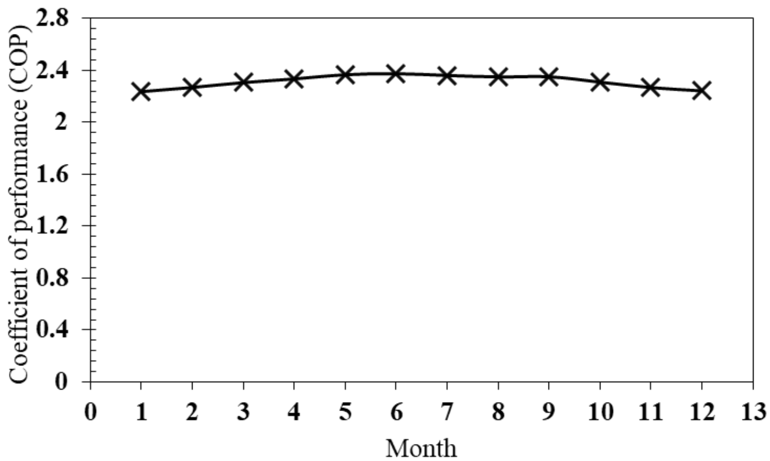

Figure 8: Monthly-averaged coefficient of performance of SMLAC

3.5 Energy Saving Potential of SMLAC

Figure 9 shows the primary energy consumption rate of the conventional air conditioning system (CAC) and the proposed SMLAC. Equations for the modelling of CAC are taken from an earlier study of Abdel-Salam *et al.* (2013). As shown in Fig.9, the energy consumption rate of the proposed SMLAC is substantially lower than that of CAC. This is due to high humid climatic conditions of Chennai throughout the year. The monthly averaged energy saving potential of the proposed SMLAC is shown in Fig.10. As shown, it varies between 59 to 67%. Moreover, the energy saving potential is found to be maximum during the months of January and December, even though the available solar radiation is comparatively minimum during these months. This is due to comparatively lower solution and air cooling loads due to lower ambient temperature and specific humidity than the other months. On the other hand, the energy saving potential is minimum during the month of May. This is due to comparatively higher ambient temperature and specific humidity than the other months which in turn increases the solution and air cooling loads. Therefore, the required energy input to SMLAC increases and consequently decreases the energy saving potential during the month of May, as shown in Fig.10.

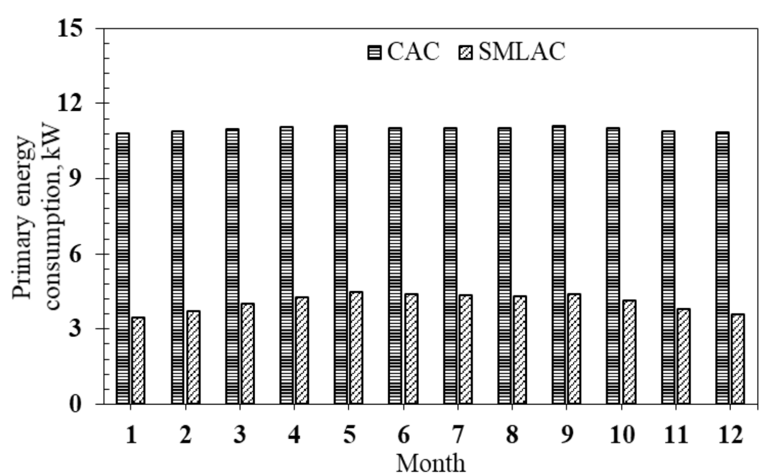

Figure 9: Monthly-averaged primary energy consumption of SMLAC and CAC

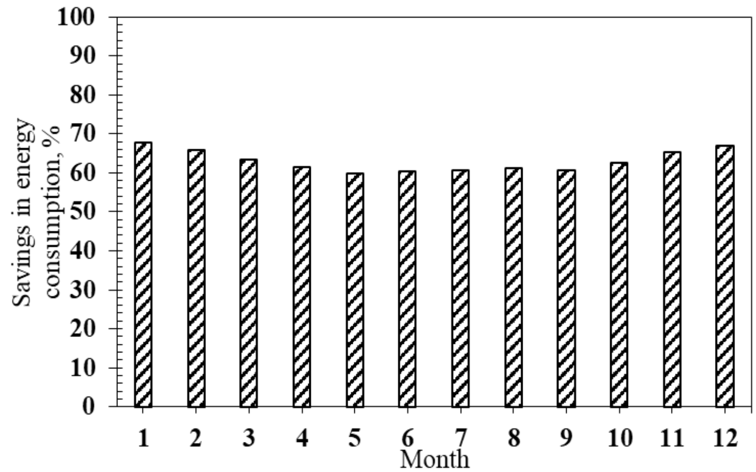

Figure 10: Monthly-averaged energy saving potential of SMLAC

3.6 Environmental Performance of SMLAC

The environmental performance of the proposed SMLAC is evaluated in terms of the reduction in the emissions of Carbon dioxide (CO_2), Nitrous oxide (NO_x) and Sulphur dioxide (SO_x) (Abdel-Salam *et al.*, 2013). As shown in Fig.10, the annual energy saving of the proposed SMLAC is around 62%. Therefore, the emissions resulting from the energy consumption of CAC from various sectors is also expected to reduce by 62%. The energy consumption share of India among the various sources and the various emission rates are listed in Table 3 (Akella *et al.*, 2009). As a result of the energy savings by the proposed SMLAC, the reduction in the emissions of CO_2, NO_x and SO_x are estimated according to the values listed in Table 3 and the results are listed in Table 4. As listed, the proposed SMLAC will avoid emissions of 1400, 16 and 6 tons/per year of CO_2, NO_x and SO_x respectively to the ambient.

Table 3: Energy share and environmental emissions from various power sources

Sl. No.	Types	Share (%)	CO_2 (g/kWh)	NO_x (g/kWh)	SO_x (g/kWh)	Efficiency
1	Coal	61	955	11.8	4.3	32
2	Gas	8	430	0	0.5	32
3	Hydro	14	9	0.03	0.07	85

Table 4: Savings in the environmental emissions

Sl. No.	Emission type	Actual (kg/year)	Savings (kg/year)
1	CO_2	2227186	1401668
2	NO_x	25970	16344
3	SO_x	9619	6053

4. CONCLUSIONS

The present study proposes a novel SMLAC for the hot and humid climatic conditions prevailing in Chennai, India. The system is designed to provide air conditioning for a room having low sensible heat factor (SHP) with loads of 3 kW sensible and 2 kW latent. Evacuated tube solar collector of 30 m^2 is used to harvest the solar energy. From the results obtained in the present study, it is concluded that the proposed SMLAC is able to achieve comfort indoor conditions (24°C and 0.0097 kg/kgda) throughout the year. Therefore, the selected values of the design and operating parameters are found to be suitable for hot and humid climatic conditions. It is also found that COP of the proposed SMLAC is found to be maximum during June which is due to the comparatively higher solar radiation and ambient specific humidity when compared to the other months. Moreover, the annual energy saving of the proposed SMLAC is around 65% when compared to CAC. Moreover, it is estimated that the proposed SMLAC will avoid emissions of 1400, 16 and 6 tons/ per year of CO_2, NO_x and SO_x respectively to the ambient.

NOMENCLATURE

A_t	Area	(m^2)
C_p	Specific heat capacity	(kJ/kg.K)
D	Diameter	(m)
h	Enthalpy	(kJ/kg)
h_{abs}	Enthalpy of absorption	(kJ/kg)
h_c	Convective heat transfer coefficient	(kW/m^2.K)
h_D	Convective mass transfer coefficient	(kg/m^2.s)
I_r	Solar radiation	(W/m^2)
k	Thermal conductivity	(kW/m.K)
K	Membrane permeability	(kg/m.s)
L	Length	(m)
m	Mass flow rate	(kg/s)
T	Temperature	(°C)
U_h	Overall heat transfer coefficient	(kW/m^2.K)
U_m	Overall mass transfer coefficient	(kg/m^2.s)
W	Specific humidity	(kg/kgda)
y	Length	(m)
Greek symbols		
δ	Thickness	(m)
Subscripts		
a	Air	
amb	Ambient	
c	Cooling water	
fa	Fresh air	
H	Heat transfer	
h	Hot water	
in	Inlet	
m	Membrane	
min	Minimum	
M	Mass transfer	
ra	Recirculation air	
s	Solution	

REFERENCES

1. Abdel-salam, A.H., Ge, G. & Simonson C.J. (2013). Performance analysis of a membrane liquid desiccant air-conditioning system. *Energy Build, 62,*559-69.
2. Abdel-salam, A.H., Ge, G. & Simonson C.J. (2014). Thermo-economic performance of a solar membrane liquid desiccant air conditioning system. *Sol Energy, 102,* 56-73.
3. Akella, A.K., Saini, R.P. & Sharma, M.P. (2009). Social, economical and environmental impacts of renewable energy systems. *Renew Energy, 34,* 390-96.
4. Conde, M.R. (2004). Properties of aqueous solutions of lithium and calcium chlorides: Formulations for use in air conditioning equipment design. *Int J Therm Sci., 43*(4), 367-82.
5. CSE Energy - A technical report of energy and buildings by Centre for Science and Environment. (2014). Available at <http:// www.cseindia.org/userfiles/Energy- and -% 20 buildings. pdf > [Accessed 5th June 2017].
6. Ge, G., Ghadiri Moghaddam, D., Abdel-Salam, A.H., Besant, R.W. & Simonson, C.J. (2014). Comparison of experimental data and a model for heat and mass transfer performance of a liquid-to-air membrane energy exchanger (LAMEE) when used for air dehumidification and salt solution regeneration. *Int J Heat Mass Transf, 68,*119-31.
7. Goetzler, W., Zogg, R., Young, J. & Johnson, C. (2014). Alternatives to Vapor-Compression HVAC Technology. *ASHRAE Journal*, *56*(10), 12-23.
8. Gurubalan, A., Maiya, M.P. & Tiwari, S. (2017). Performance characterization of membrane dehumidifier with desiccants in flat-plate arrangement. *Energy Build*, *156*,151-62.
9. Saleh, A.,Mosa, M. (2014). Optimization study of a single-effect water–lithium bromide absorption refrigeration system powered by flat-plate collector in hot regions. *Energy Convers Manage, 87,* 29-36.