Analysis and comparison between Maisotsenko cycle-based indirect contact M-Cooler dehumidifier and structured packing-based dehumidifier

A. Tryambke Pandey[1], B. P. K. S. Tejes[1], *C. B. Kiran Naik[1]

[1]Sustainable Thermal Energy Systems Laboratory (STESL), Department of Mechanical Engineering, National Institute of Technology Rourkela, Odisha–769008, INDIA,

*Corresponding author: k.bukke@gmail.com; naikkb@nitrkl.ac.in

Abstract

In this study, a novel liquid desiccant employing Maisotsenko cycle-based indirect contact M-cooler in the dehumidifier is proposed. The proposed system has multipurpose applications such as building space cooling and drying food products. Liquid desiccant chosen for the present investigation is lithium chloride (LiCl). The developed thermal model is used to predict the performance of the above-mentioned conventional and novel system. The proposed model has been validated with the data available in the literature and found to match well with a maximum allowable error of ±9.7%. Depending upon developed model, a performance comparison of conventional and novel systems is carried out. From the performance assessment, it is observed that cooling load is low and condensation rate is high for the novel system. Further the performance of the novel system is done using variable Prandtl and Eckert number. Also, for the given operating range, the maximum vapor absorption rate by the liquid desiccant from the atmospheric humid air is about 4.62 LPM.

Keywords: Thermodynamic model, hydrophobic membrane, Maisotsenko cycle, Liquid desiccant, indirect contact, Dehumidifier.

1. Introduction/Background

Development in the sectors of agriculture and human comfort is in high rise for high productivity and better air quality. In regions with high relative humidity such as in coastal zones, integrated Maisotsenko cycle (M-Cooler) based indirect (polyvinylidene fluoride hydrophobic membrane) contact dehumidifier (MCID) can be chosen as an alternative over the conventional structured based dehumidifier because of low energy consumption, ease of fabrication, low maintenance cost and capability to remove bacteria and virus [1], etc. The use of liquid desiccant (LD) dehumidification/desalination system has several promising advantages including effective utilization of industrial waste heat [2,3], low grade and renewable energy e.g., geothermal energy [4], solar energy [5,6]. The earlier developed liquid desiccant air conditioning systems (LACS) had certain drawbacks as due to direct interaction between liquid desiccant and air there is a chance of desiccant carryover with air which tends to corrode the air duct, room furniture, walls also may damage to the health of people in air conditioning space and may spoil perishable goods in the drying chamber, hence in this study a flat plate membrane-based type indirect evaporative cooler is used where liquid desiccant i.e., Lithium chloride (LiCl) and air are separated by a hydrophobic PVDF (polyvinylidene fluoride) membrane is used which facilitates exchange of mass and energy simultaneously with high efficacy and notable pressure drop. LiCl is preferred as it is the most reliable and chemically stable liquid desiccant due to its virtue of low vapour pressure during dehumidification and regeneration process. In addition to it is easier to compare conventional and novel systems, due to previously carried out research using LiCl. In dehumidifier, the liquid desiccant absorbs water vapour. The main driving force causing the energy and mass exchange to occur are temperature and vapour pressure gradient at the air-liquid desiccant boundary. Although air gets dehumidified in the conventional system yet the exit air temperature of the dehumidifier is low which makes it incapable of drying purposes; even air that is being released into the atmosphere is almost saturated which could have been used to extract pure water.

After an extensive literature study, it is evaluated very limited researchers proposed thermal models to investigate membrane-based LD dehumidifier performance [1-3,14]. Moreover, limited thermal models were developed for analyzing membrane based conventional LDAC system performance [4-9]. Further, very few researchers analyzed the performance of the conventional LDAC system for drying and fresh water extraction applications [10-13,15]. Furthermore, it is noticeable from the reported studies that there is a deficit in detailed research on incorporating Maisotsenko cycle-based M-cooler in the dehumidification process.

From the mentioned literature gap, in this analysis, a innovative system incorporating indirect contact-based M-cooler in the LD dehumidifier is proposed. The proposed novel system can be used for room air conditioning application (in between 18°C-25°C), drying the agricultural/food products at moderate temperature (in between 40°C-60°C), reducing energy requirement as well as enhancing conventional LDAC system performance. Further, a simplified thermal model is proposed to analyze the novel system performance for assessing the M-cycle based dehumidification performance. Moreover, the proposed thermal model also used to analyze the performance of structured packing-based dehumidifier. Further, employing the proposed model effect of dehumidifier inlet parameters on conventional and novel systems performance. In addition, impact of integrated Maisotsenko cycle (M-Cooler) based indirect (polyvinylidene fluoride hydrophobic membrane) contact dehumidifier (MCID) is assessed in detail.

2. Analytical model

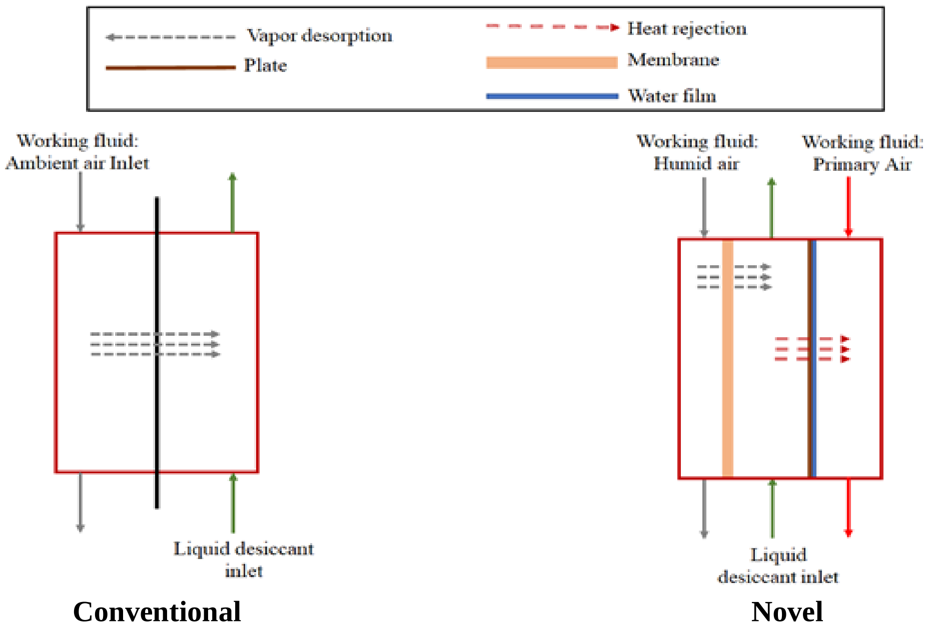


Figure 1. Comparison of dehumidifier in conventional and novel systems.

The MCID present in the novel system has three loops (Figure 2), they are humid air loop (i–ii), primary air loop (1–2) and LD loop (A–B). In humid air loop (i–ii), the humid air enters the dehumidifier in counter flow direction, and it is in indirect interaction with the strong and hot LD through the PVDF based hydrophobic membrane. When the humid air comes in indirect contact with the LD, moisture removal from the humid air occurs due to vapour pressure and temperature gradients as well as exothermic reaction. In MCID, by incorporating membrane in the dehumidifier eradicates the carryover and improves the vapour absorption rate due to hydrophobic nature of the membrane (i.e., only water vapour can penetrate through the membrane from humid air to the LD). In the primary air loop (1–2), the primary air and water with one another are interacted in counter flow. In this process, evaporation of water film occurs resulting in lowering the water temperature. This phenomenon leads to cool the LD which is in indirect contact with the water through a stainless-steel plate. In LD loop (A–B), the LD comes in indirect contact with the cold water and humid air which are separated by the plate and membrane, respectively. Thus, the hot and strong LD entering the MCID converts to cold and weak LD.

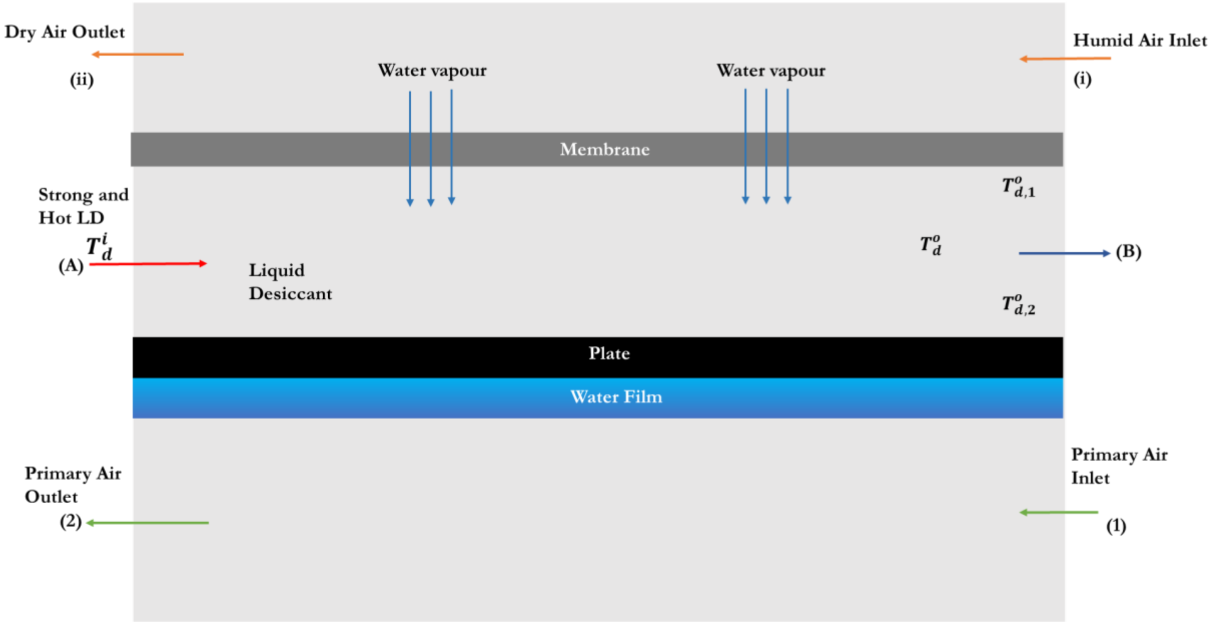


Figure 2. Internal structure of the MCID.

2.1. Thermal model

In order to evaluate the performance of the conventional and novel systems shown in Figure 1, certain assumptions are taken into consideration and they are presented as follows,

- Entire system is in steady state.
- Heat lost from the system to the surrounding is negligible.
- Latent heat which is dissipated in the M-Cooler based dehumidifier during the condensation process is entirely captivated by the LD.
- Energy consumed by pumps are considered insignificant.
- Mass flow rate of solution and fluids (both air and water) are assumed to be constant.

2.2. Governing Equations

Following are the equations obtained for the energy balance between the novel and the conventional dehumidifier,

Humid air enthalpy change $$\Delta h_{a,dh} = h_a^i - h_a^o \quad \text{(Eq. 1)}$$

Desiccant enthalpy change $$\Delta h_{d,dh} = c_{p,d}(T^o_{d,dh} - T^i_{d,dh}) \quad \text{(Eq. 2)}$$

Water enthalpy change $$\Delta h_{w,dh} = c_{p,w}(T^o_{w,dh} - T^i_{w,dh}) \quad \text{(Eq. 3)}$$

Heat transfer in dehumidifier: $$\Theta_{dh} = \dot{m}_d(h^o_{wf,dh} - h^i_{wf,dh}) = \dot{m}_a(h^i_{a,dh} - h^o_{a,dh}) \quad \text{(Eq. 4)}$$

where, $\Delta h_{a,dh}$, $\Delta h_{d,dh}$ and $\Delta h_{w,dh}$ are change in the specific enthalpy of air, LD, and water (kJ/kg) in dehumidifier, respectively. '$T^o_{a,dh}$' and '$T^i_{a,dh}$' are the outlet and inlet temperature of the working fluid flowing through dehumidifier and $\dot{m}_{wf}$ mass flow rate (kg/sec) of working fluid, '$h^o_{wf,dh}$' and '$h^i_{wf,dh}$' are the outlet and inlet specific enthalpy (kJ/kg) of the working fluid in the dehumidifier, '$c_{p,a}$', '$c_{p,d}$' and '$c_{p,w.}$' are heat capacities (kJ/kg-K) of air, LD, and water, 'Θ_{dh}' is rate of heat transfer (kW) in the dehumidifier. '$T^o_{a,dh}$' is the exit air temperature of dehumidifier.

To evaluate the performance of M-cooler, following relations are used,

Thermal effectiveness of dehumidifier: $$\varsigma_{T,d} = \frac{(T^i_{a,dh} - T^o_{a,dh})}{(T^i_{a,dh} - T^i_{d,dh})} = \frac{\gamma_d c_{p,d}(T^o_{d,dh} - T^i_{d,dh})}{\gamma_a c_{a,d}(T^i_{a,dh} - T^i_{d,dh})} \quad \text{(Eq. 5)}$$

Heat capacity: $$C_c = \dot{m}_d c_d,\ C_h = \dot{m}_{pa} c_{pa} \quad \text{(Eq. 6)}$$

$$C_{\min} = \min(C_c, C_h), C_{\max} = \max(C_c, C_h),\ R = \frac{C_{\min}}{C_{\max}}, NTU = \frac{UA}{C_{\min}} \quad \text{(Eq. 7)}$$

Overall heat transfer cofficient: $$\frac{1}{U} = \frac{1}{h_{pa}} + \frac{\delta_p}{k_p} + \frac{\delta_w}{k_w} \quad \text{(Eq. 8)}$$

Effectiveness of conunter flow M-cooler: $$\varepsilon = f(NTU, R, flow\,arrangement) \quad \text{(Eq. 9)}$$

$$\varepsilon = \frac{1 - \exp(-NTU(1-R))}{1 - R\exp(-NTU(1-R))}\ (counter\ flow) \quad \text{(Eq. 10)}$$

Maximum heat transfer rate: $$q_{\max} = C_{\min}(T^i_{pa} - T^i_d) \quad \text{(Eq. 11)}$$

Actual heat transfer rate: $$q = \varepsilon q_{\max} \quad \text{(Eq. 12)}$$

Primary air outlet temperature: $$T^o_{pa} = T^i_{pa} + \frac{q}{C_c} \quad \text{(Eq. 13)}$$

Plate temperature: $$T^o_p = T^i_p - \frac{q}{C_h} \quad \text{(Eq. 14)}$$

Temperature of LD on M-cooler side: $$T^o_{d,1} = T^o_p \quad \text{(Eq. 15)}$$

Exit temperature of LD from MCID: $$\frac{T^o_{d,dh} - T^o_{d,1}}{T^o_{d,2} - T^o_{d,1}} = \frac{1}{3} Ec\,\Pr(1 - \bar{y}^4) + \frac{1}{2}(1 + \bar{y}) \quad \text{(Eq. 16)}$$

Dimensionless numbers: $Ec = \dfrac{V_{\max}^2}{C_p(T_{d,2}^o - T_{d,1}^o)}$, $\Pr = \dfrac{\mu c_{p,d}}{k}$ (Eq. 17)

Vapour pressure effectiveness of dehumidifier: $\varsigma_{VP,dh} = \dfrac{(P_{a,dh}^i - P_{a,dh}^o)}{(P_{a,dh}^i - P_{d,dh}^i)}$ (Eq. 18)

where, '$\varsigma_{T,d}$' and '$\varsigma_{VP,d}$' are thermal and vapour effectiveness of dehumidifier, 'C_c' and 'C_h' are the cold and hot heat capacity (kJ/K) of primary air and LD respectively, 'C_{min}' and 'C_{max}' are the minimum and maximum heat capacity (kJ/K) among the interacting fluids (i.e., primary air and LD), R is the ratio of the least to extreme heat capacity, 'U' is the overall heat capacity, 'h_{pa}' is heat transfer coefficient (W/m^2-K) of the primary air, 'k_p' and 'k_w' are the thermal conductivity (W/m-K) of the plate and water film, ε is the effectiveness of the M-cooler based dehumidifier, 'δ_p' and 'δ_w' is the thickness of the plate and water film (mm), 'q_{max}' is the maximum amount heat transfer (kW) among the working fluids (i.e., primary air and LD), whereas, 'q' is actual heat transfer (kW) from LD, '$T_{p,a}^o$' and 'T_{pa}^i' are the outlet and inlet temperature of the primary air in the M-cooler, '$T_{d,1}^o$' and '$T_{d,2}^o$' are the temperature of the plate and membrane (Figure 2), '$\bar{y}$' is the centreline distance between the plate and the PVDF membrane, 'T_d^o' is the final temperature of the LD leaving the M-cooler based dehumidifier, 'Ec' and 'Pr' are two dimensionless number (Ec-Eckert, Pr-Prandtl), 'V_{max}' is the maximum velocity of the LD in the M-cooler based dehumidifier, 'μ' is the dynamic capacity of the LD.

2.2.1. Vapour pressure calculation

Through Antonie equation, the working fluid (water/air) and LD solution's vapour pressure is evaluated as below,

$$\rho_o = \exp\left(23.1964 - \left(\frac{3816.44}{T_o - 46.13}\right)\right) \quad \text{(Eq. 19)}$$

'ρ_o' is LD/fluid vapour pressure (Pa) and 'T_o' is LD/fluid temperature (K).

$$\rho_w = \exp\left(23.1964 - \left(\frac{3816.44}{T_w - 46.13}\right)\right) \ (\rho_o = \rho_w) \quad \text{(Eq. 20)}$$

$$\rho_a = \varphi_{rh} \exp\left(23.1964 - \left(\frac{3816.44}{T_a - 46.13}\right)\right) \quad \text{(Eq. 21)}$$

$$\rho_d = \chi_w \exp\left(23.1964 - \left(\frac{3816.44}{T_d - 46.13}\right)\right) \quad \text{(Eq. 22)}$$

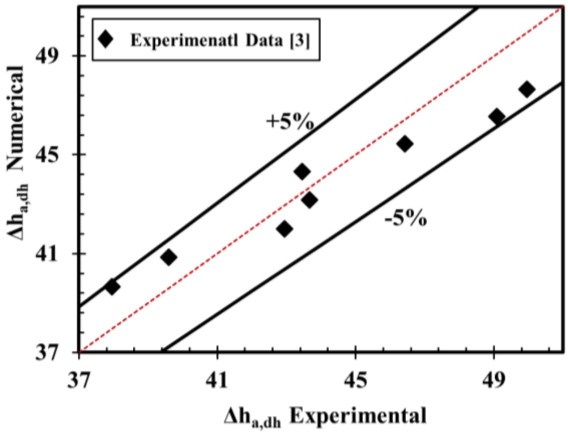
where, 'φ_{rh}' is the relative humidity, 'ρ_a' is the ambient vapour pressure (kPa), 'χ_w' is the water concertation (kg_{H_2o}/kg_d), and 'ρ_w' is the vapour pressure (kPa) of the LD. 'T_w,' 'T_a,' and 'T_d' are the temperatures of water, air, and LD, respectively.

Table 1. Specifications, and operating conditions of M-cooler/membrane-based dehumidifier.

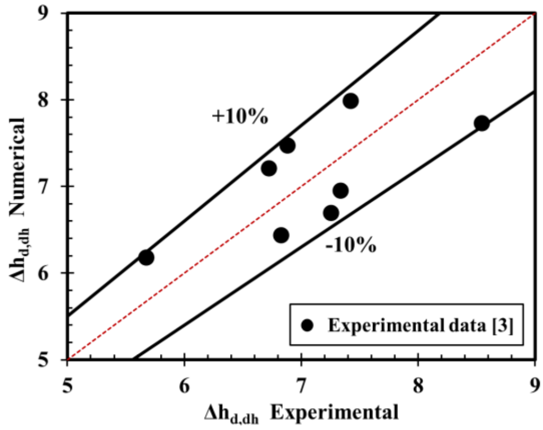
Specifications/operating parameters	Unit	Membrane based dehumidifier [3]	M-cooler [4]
Length	cm	20	20
Height	cm	20	20
Width of air channel	cm	0.4	0.4
Solution channel width	cm	0.4	0.4
Plate thermal conductivity	W/m-K	–	15
Membrane thickness	cm	0.018	–
Membrane thermal conductivity	W/m-K	0.3	–
Moisture conductivity of membrane	g/m-s	5.34 x 10^{-4}	–
Humid air temperature	°C	35-38	35-38
Liquid desiccant temperature	°C	24.8-28	–
Inlet air specific humidity	g_{wv}/kg_{da}	20-31	–
Liquid desiccant concentration	kg_{LiCl}/kg_{sol}	0.3-0.4	–
Eckert number	–	–	0.02-0.18
Prandtl number	–	–	1-12
Membrane	–	Polyvinylidene fluoride (PVDF)	
Liquid desiccant	–	Lithium Chloride	

2.3. Model validation

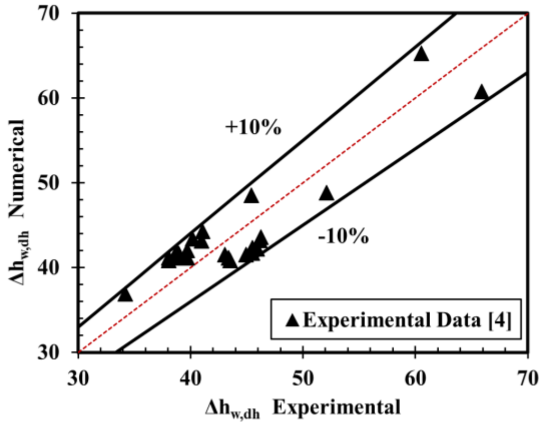
To predict the performance of developed model for both conventional and novel systems validation is required. Henceforth, the proposed model is validated with data available experimentally in reported studies [3,4]. Developed model equations have been validated in two segments, they are membrane-based internally cooled dehumidifier developed generalized equations for evaluating the exit conditions such as LD and air enthalpy (1–2) have been validated with the Wei et al. [3] and the integrated M-cooler developed equations for predicting the water outlet enthalpy (3) has been validated with the Saraireh et al. [4]. The specifications and operating range of the membrane dehumidifier and integrated M-cooler are presented in Table 1. The dimensions for the novel system are considered same as that of the conventional system as mentioned by Wei et. Al. [3] to compare both the system. The dehumidifier vapour pressure effectiveness ($\zeta_{VP,d}$) and thermal effectiveness ($\zeta_{T,d}$) are assumed as 0.65. The experimental results for the change in enthalpy of air, LD, and water are compared with the theoretical results attained from the developed model and observed reasonable agreement with maximum allowable error ±10% as depicted in Figure 3. From validation analysis, it is noticed that proposed model is in good agreement for evaluating performance of conventional and proposed novel systems.



(a) Humid air enthalpy validation



(b) LD enthalpy validation



(c) Water enthalpy validation

Figure 3. Validation of Numerical results with Experimental data for MCID [3,4].

Table 2. Specifications, and physical properties of MCID [3].

Specifications	Unit	Conventional	Novel
Length	cm	20	20
Height	cm	20	20
Width	cm	0.4	0.4
Membrane	–	–	PVDF
Liquid desiccant	–	LiCl	LiCl
Plate thermal conductivity in MCID	W/m-K	–	15
Membrane thickness	cm	–	0.018
Thermal conductivity of membrane	W/m-K	–	0.3
Moisture conductivity of membrane	g/m-s	–	$5.34\ 10^{-4}$

3. Results and discussion

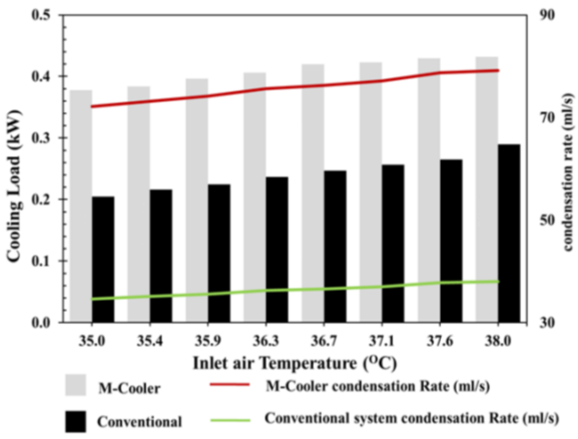
In this study, the proposed model presented in the section 2 is used to assess performance of structured packing and MCID dehumidifier. The parameters chosen for comparing the performance of both systems for air conditioning application are cooling load and condensation rate are considered. The specifications, thermo-physical properties, operating range and parameters presented in Tables 2 and 3 are considered for the present investigation. In the present work, the inlet parameters like temperature, vapour pressure, and mass flow rate of primary/humid air at the dehumidifier, respectively. The inlet parameters, system specifications and their operating range chosen for the current study are enumerated in Tables 2 and 3. The effect of dehumidifier inlet parameters such as humid air and desiccant inlet temperature, specific humidity of air and LD concentration on the performance parameters as well as the performance comparison of both the systems is presented in Figure 4.

Table 3. Operating range and conditions for dehumidifier in conventional/novel systems.

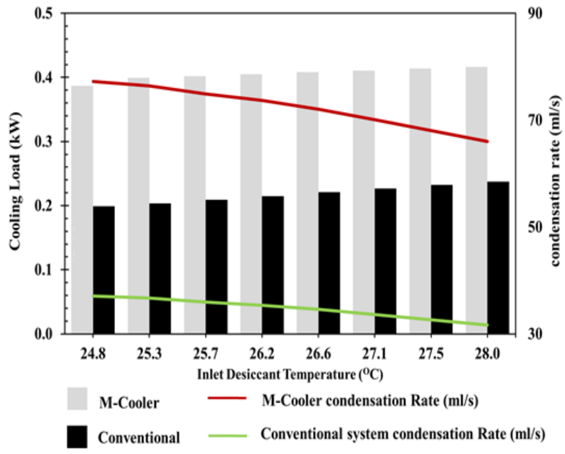
Operating parameters	Units	Operating range	Inlet condition
Air inlet temperature in conventional system	℃	35-38	38
Humid air inlet temperature in novel system	℃	35-38	38
Liquid desiccant temperature	℃	24.8-28	24.8
Conventional inlet air specific humidity	g_{wv}/kg_{da}	20-31	31
Humid air inlet temperature in novel system	g_{wv}/kg_{da}	20-31	31
Liquid desiccant concentration	kg_{LiCl}/kg_{sol}	0.3-0.4	0.35
Inlet cooling water temperature in MCID	℃	17-20.2	17
Eckert number	–	0.02-0.18	0.02
Prandtl number	–	1-12	4

3.1. Cooling load and condensation rate

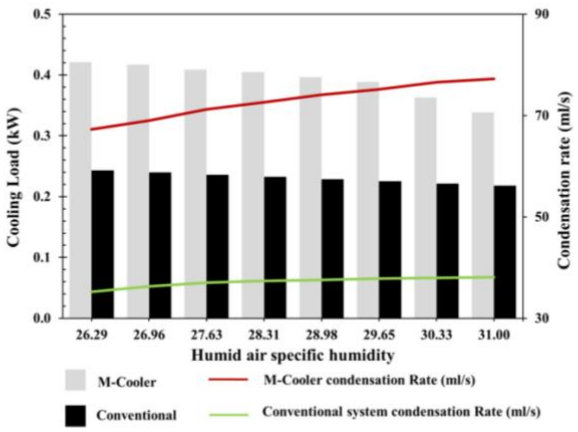
Figure 4 represents the impact of dehumidifier inlet parameters which are humid air and LD inlet temperature, humid air specific humidity, desiccant concentration, and LD to fluid flow ratio on direct evaporative cooler (DEC) cooling load (kW) and dehumidifier condensation rate for both the conventional and novel systems. From Figure 4a and 4f, it has been noticed as inlet humid air temperature is increased from 35 ℃ to 38 ℃, the cooling load on both systems increases by 29.3% and 12.5%, respectively. This happens because as inlet humid air temperature at the entrance of dehumidifier increases, exit temperature of the dehumidified air from the dehumidifier also increases. As a result, cooling load of the DEC increases for conventional and novel systems. In Figure 4b and 4f, it is found that as the LD inlet temperature increases from 24.8 ℃ to 28 ℃, the cooling load of both the systems tends to increase by 16.3% and 7.1%, respectively. This is due to increase in humid air inlet temperature with increase in LD inlet temperature while the interaction between the LD and humid air takes place. Thus, DEC cooling load increases. It is understood from Figure 4c and 4f that, with rise in humid air specific humidity from 26.3 g_{wv}/kg_{da} to 31 g_{wv}/kg_{da}, the DEC cooling load capacity of both the systems decreases by 19.6% and 10.4%, respectively. It happens due to increase in condensation rate and decrease in temperature across the dehumidifier. As illustrated in Figure 4d and 4f, as the LD concentration is increased from 0.3 $kg_{LiCl}/kg_{sol.}$ to 0.4 $kg_{LiCl}/kg_{sol.}$, the DEC cooling load decreases by 22.3% and 13.3% for both the systems. This indicates that with increase in LD concentration, moisture absorption rate as well as temperature drop increases which in turn reduces the conventional and novel systems DEC cooling load.



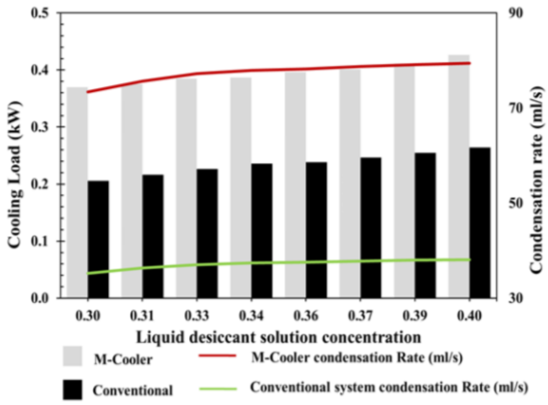
(a) Humid air temperature



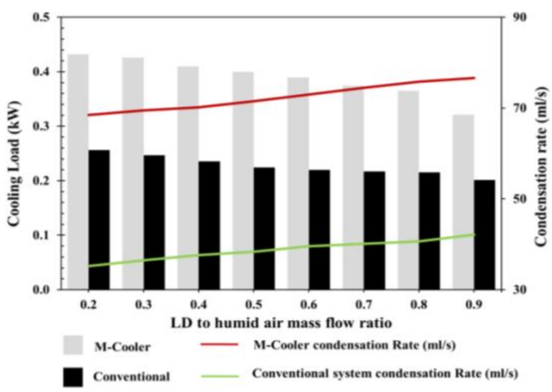
(b) Desiccant temperature



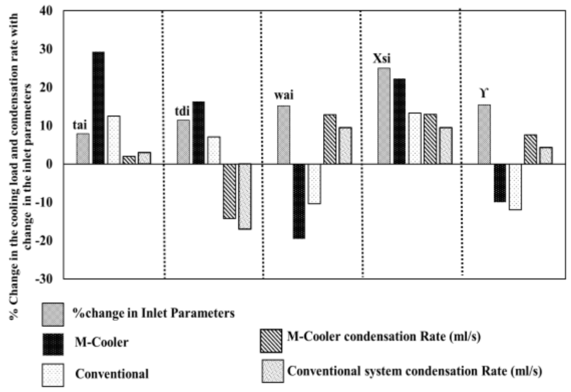
(c) Humid air specific humidity



(d) Desiccant concentration



(e) LD to humid air mass flow ratio



(f) Percentage variation in the cooling load and condensation rate of the conventional and novel systems with increase in the operating parameters of MCID

Figure 4. Influence of inlet parameters on cooling load and condensation rate of M-Cooler and conventional dehumidifier systems.

As shown in Figure 4e and 4f, it is realized that with increase in LD to humid air flow ratio from 0.2 to 0.9, the DEC cooling load decreases by 12% and 10% for both the systems. This occurs because as the flow ratio increases, the moisture removal rate and temperature drop increases. Subsequently, the cooling load on DEC decreases for both the systems.

As depicted in Figures 4a and 4f, it is analyzed that with increment in humid air temperature from 35 ℃ to 38 ℃, there is no phenomenal change in condensation rate for both the systems. It signifies that raise in humid air temperature does not have predominant role on condensation rate of both the systems and it is in line with the trends indicated in the literature [9]. As the LD temperature increases from 24.8℃ to 28℃, the condensation rate for both the systems decreases by 17% and 14.3% for conventional and novel system (Figures 4b and 4f). It is due to decrease in temperature difference between the LD and the humid air which further reduces the vapour pressure gradient. This decrease results in reduction of condensation rate for both the conventional and the novel dehumidifiers. In Figures 4c and 4f., it is analyzed that as the specific humidity increases from 26.3 g_{wv}/kg_{da} to 31 g_{wv}/kg_{da}, the condensation rate increases by 9.5% and 12.9% for both the systems. This arises because as the specific humidity increases, the moisture level present in the humid air increases. Thus, the strong LD has more scope for moisture absorption from the humid air. Consequently, the condensation rate increases. The raise in LD concentration from 0.3 $kg_{LiCl}/kg_{sol.}$ to 0.4 $kg_{LiCl}/kg_{sol.}$, the condensation rate increases by 4.3% and 7.6% for conventional and novel systems as shown in Figures 4d and 4f. This is due to increase vapor absorption from the moist air due to increase in LD concentration (i.e., vapour pressure gradient raises with raise in LD concentration). As depicted in Figures 4e and 4f, it is found that with raise in LD to humid air mass flow ratio from 0.2 to 0.9, the condensation rate increases by 8% and 10% for the both the systems, respectively. It occurs due to rapid increase in interaction of fresh and strong LD with the humid/moist air as the flow ratio increases. This increases the moisture carrying capacity of the LD and ultimately result in upsurge condensation rate. Further, from Figures 4f it is also observed that MCID of novel system achieves higher condensation rate compared to structured packing based direct contact dehumidifier of the conventional system. This reveals the fact that by incorporating Maisotsenko cycle will improve the overall dehumidification performance and reduces the environmental, chemical, and thermal losses during moist air and LD interaction in the dehumidifier.

From Figures 4 it is also noticed that specific humidity and LD temperature has significant impact on the cooling load and condensation rate compared to other dehumidifier inlet parameters. Moreover, it is also determined that cooling load is low and condensation rate is high for MCID compared to structured packing dehumidifier. This indicates that novel system has superior performance in comparison with conventional system. Further, it signifies that incorporation of M-cycle in the dehumidifier improves the performance as well as water vapour absorption rate.

3.2. Eckert and Prandtl number

Figure 5 shows variation of LD outlet temperature with increase in Prandtl number (Pr) and Eckert number (Ec). Pr is varied from 1 to 12, the Pr is considered as higher than 1 since thermal boundary layer thickness is lower than hydrodynamic boundary layer thickness for LiCl whereas it is considered less than 12 because the value calculated by Conde [11] for LiCl is observed to be maximum of 12. The Eckert number (Ec) is calculated using the Eq. 17 where the maximum velocity ranges from 0.5 m/s to 1.5 m/s hence, Ec is observed to vary from 0.02 to 0.18. The LD temperature increases with increase in Pr and Ec (Eq. 16). As Ec and Pr is increased from 0.02 to 0.18 and 1 to 12, respectively, the LD outlet temperature increases by 7.8% and 18%, respectively. This is due to increase in the velocity of the LD giving it less time to interact with the cooling plate which results in the LD outlet temperature to increase with increase in Ec and Pr.

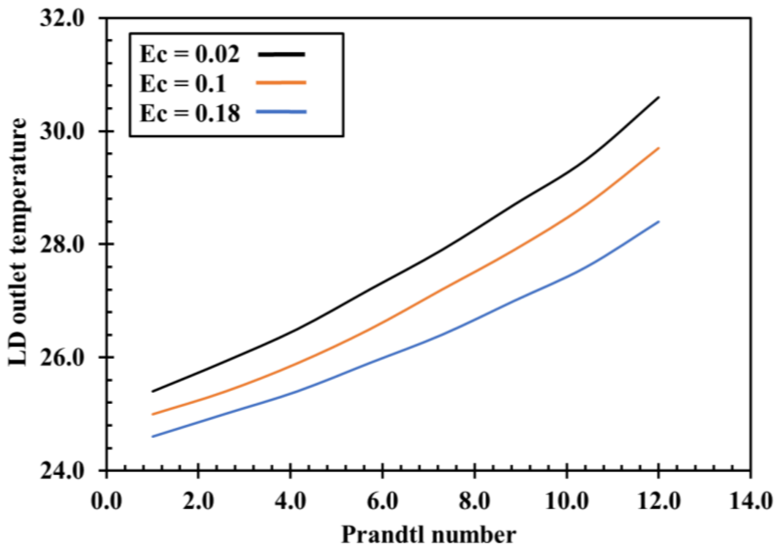


Figure 5. Variation of LD outlet temperature with increase in Prandtl and Eckert number.

4. Conclusions

From the performance comparison of structured packing dehumidifier and MCID following conclusions are obtained,

- Within any specified conditions specified, it is noticed that MCID provides superior vapor absorption rate than structured packing-based dehumidifier.
- It is observed that dehumidifier specific humidity and LD temperature has significant effect on performance of structured packing dehumidifier and MCID.
- Incorporating MCID system in the novel air conditioning cum desalination and the novel drying cum desalination systems, performance of traditional system can be significantly enhanced.
- It is analyzed that on increasing the Eckert and Prandtl number the outlet temperature of LD is increased.

Thermal model developed in the present study can be used effectively for performance evaluation of LD based multipurpose thermal systems. Further, this investigation opens a new window for the advancement of conventional LDAC/dryer system for extracting pure water from humid air.

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