Performance Of The Dehumidification Cycle Of A 3-Fluid Liquid Desiccant Membrane Air-Conditioning System

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Abstract

A new 3-fluid liquid-to-air membrane energy exchanger (LAMEE) has been proposed to effectively meet sensible and latent loads. The 3-fluid LAMEE is composed of several adjacent air and solution channels separated by semi-permeable membranes, and refrigerant tubes that are installed within the solution channels to control the temperature of the solution in the LAMEE. In this paper, the performance of a dehumidification cycle of a 3-fluid liquid desiccant membrane air-conditioning (M-LDAC) system is presented and investigated through simulation and experimentation. The coefficient of performance (COP) of the M-LDAC system is evaluated at different inlet refrigerant temperatures. A system COP of 0.70-0.84 is required for neutral air conditions. The 3-fluid M-LDAC system effectively removes sensible and latent loads.

Introduction

Air-conditioning systems are responsible for a significant portion of the total energy consumption in buildings. The cost and energy consumption of air cooling and dehumidification are especially notable in warm and humid climates. For instance, El-Dessouky et al. (2004) reported that space cooling accounts for up to 70% of the total building energy consumption in the Middle East region. Treating outdoor air prior to supplying it to a building is essential in providing adequate thermal comfort and indoor air quality (IAQ). The most commonly used air conditioning system is the conventional air conditioning system (i.e, the vapor compression system). Air dehumidification is responsible for a large portion of energy consumption in vapor compression air conditioning systems, where the air is dehumidified by condensing water from the air by cooling the air to a temperature below its dew point temperature. Thereafter, the air may be reheated before being supplied to the conditioned space. This process consumes a large amount of energy

A new technology that resolves this issue is the liquid desiccant air conditioning (LDAC) system. This system uses a liquid desiccant (an aqueous salt solution) to simultaneously dehumidify and cool the outdoor air. The salt solution cycles through a closed dehumidification-regeneration loop where the solution cools and dehumidifies the air followed by regeneration where the diluted desiccant returns to its initial concentration. The membrane LDAC (M-LDAC) system incorporates a liquid-to-air membrane energy exchanger (LAMEE) as the dehumidifier and regenerator. Similar to a flat plate exchanger, the LAMEE separates two different fluids into channels. The LAMEE implements a vapor permeable membrane that separates the solution and air while allowing for heat and water vapor transfer across the membrane. An M-LDAC system is capable of lowering the cooling energy consumption by up to 47% in a hot and humid climate (Abdel-Salam and Simonson 2013).

Moisture transfer between the desiccant solution and air is accompanied by a phase change of water between liquid and vapor. The resulting phase change energy reduces the temperature differential between the air and solution and negatively impacts cooling and dehumidification capacity. Experimental data, collected by Abdel-Salam et al. (2016b), shows that phase change reduces the cooling and dehumidification of the supply air.

To counter this reduction, Abdel-Salam et al.(2016a) presented and tested an internally cooled/heated LAMEE named the 3-fluid LAMEE. By piping a refrigerant through the solution channel, it is possible to control the solution temperature and minimize the impact of phase change energy. Abdel-Salam et al. (2016a) concluded that keeping the desiccant solution temperature constant improves the sensible and latent effectivenesses of LAMEEs.

Objectives

In this article, the Transient System Simulation program (TRNSYS) is used to investigate the performance of the dehumidification cycle of a 3-fluid M-LDAC system. The objectives of this work are:

i) To develop a 3-fluid LAMEE TRNSYS simulation component that models the dehumidification cycle of the 3-fluid LAMEE designed by Abdel-Salam et al. (2016a).

ii) Use the developed component to evaluate the performance of the 3-fluid M-LDAC system.

The 3-fluid LAMEE LDAC System

Figure 1 presents a labelled schematic and cross-sectional view of the 3-fluid LAMEE and Table A.1, at the end of the paper, summarizes its dimensions. The refrigerant is divided into seven tubes in the solution channel. The refrigerant and solution are in counter flow configuration. Air flows in two channels and is in counter flow

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configuration with the solution. The semi-permeable membranes dividing the air and solution channels allow for heat and moisture transfer while the refrigeration piping (titanium) allows only heat transfer.

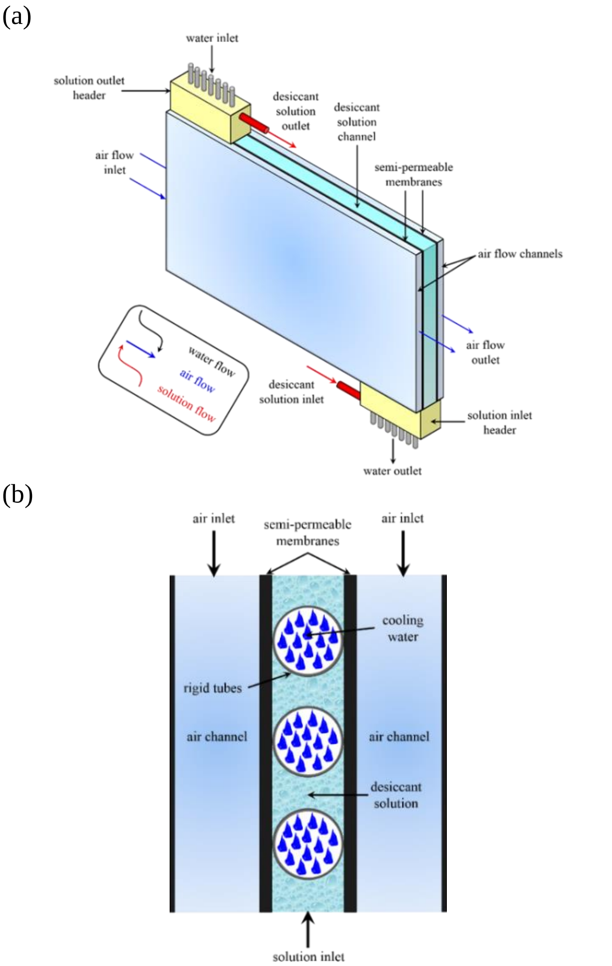


Figure 1: a) side and b) cross-sectional view of the 3-fluid LAMEE (Abdel-Salam et al., 2016a)

Figure 2 corresponds to a cooling and dehumidifying cycle of the 3-fluid M-LDAC system. The liquid desiccant, lithium chloride (LiCl) aqueous solution, cycles through the dehumidifier and cooler. This simplified M-LDAC system assumes solution regeneration does not require input energy. The model serves as a simple and practical means to analyse the dehumidification cycle and verify the TRNSYS model. A chiller cools the salt solution prior to dehumidification to enhance heat and moisture transfer between air and solution. The refrigerant, water in this case, cycles through a cooler and the dehumidifier. Water is used to control the solution temperature. Fan and pumps are used to transport air, solution, and water through the dehumidifier.

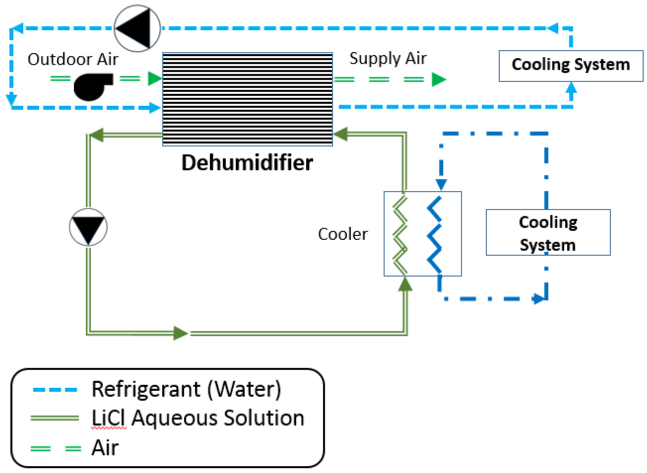


Figure 2: Schematic of the cooling and dehumidification cycle of the 3-fluid M-LDAC system

All three flows are laminar. The cold, intermediate and hot fluid are water, solution, and air, respectively. It is worth mentioning that this simulation component is limited to the configuration in Figure 2.

The work presented in this paper is based on the experimental studies performed by Abdel-Salam et al. (2016a). He designed and tested a 3-fluid LAMEE as a dehumidifier and regenerator. His studies were performed at the component level, that is, studying the 3-fluid LAMEE as a dehumidifier and regenerator (Abdel-Salam et al., 2016c), individually. In the dehumidifier tests, Abdel-Salam et al. (2016a) evaluated the sensible and latent effectiveness at different inlet refrigerant (i.e., water) temperatures and mass flow rates. The dehumidifier simulation model assessed in this paper is designed to replicate this data.

Performance Evaluation

Sensible and latent effectiveness of LAMEEs compares the actual heat and moisture transfer to the theoretical maximum heat and moisture transfer. For Cr* (ratio between heat capacity rates of solution and air) ≥ 1, the sensible (ε_s) and latent effectiveness (ε_l) of a 3-fluid LAMEE are, respectively, defined as (Abdel-Salam et al., 2017):

$$\varepsilon_s = \frac{T_{air,in} - T_{air,out}}{T_{air,in} - T_{w,in}} \tag{1}$$

and

$$\varepsilon_l = \frac{W_{air,in} - W_{air,out}}{W_{air,in} - W_{s\,@\,T_{w,in},C_{sol,in}}}. \tag{2}$$

where, T is temperature (˚C) and W is humidity ratio (g_v/kg_{da}). Subscripts refer to the fluid, either air or water (w) or solution (s), and whether the parameter corresponds to the inlet (in) or outlet (out). Furthermore, $W_{s\,@\,T_{w,in},C_{sol,in}}$ defines the equivalent humidity ratio of the solution at the inlet solution concentration and inlet water temperature.

The LAMEE is characterized by several dimensionless design and operating parameters. The number of heat transfer units (*NTU*) is a design parameter that relates the overall heat transfer coefficient (*U*) and area (*A*) of the membrane to the minimum heat capacity rate (C_{min}).

$$NTU = \frac{UA}{C_{min}}. \quad (3)$$

Ratio of thermal capacity rates (*Cr*) describes the operating conditions of an exchanger. In the 3-fluid LAMEE, *Cr* refers to ratio of heat capacity rates between solution and water while *Cr** is the ratio between solution and air. That is,

$$Cr = \frac{C_{sol}}{C_w} = \frac{(\dot{m}c_p)_{sol}}{(\dot{m}c_p)_w} \quad (4)$$

and

$$Cr^* = \frac{C_{sol}}{C_{air}} = \frac{(\dot{m}c_p)_{sol}}{(\dot{m}c_p)_{air}}. \quad (5)$$

where, C corresponds to thermal capacity rate in kW/K which is the product of mass flow rate, $\dot{m}$ (kg/s), and heat capacity, c_p (kJ/kg/K).

This study uses constant air and solution flow rates ($Cr^* = 1.8$) and varies *Cr* by adjusting the water mass flow rate.

The coefficient of performance (*COP*) describes the efficiency of an air conditioning system and is the ratio between the energy transfer rate from the air (cooling and dehumidification) (*CC*) and energy consumed (*E*),

$$COP = \frac{CC}{E} = \frac{\dot{m}_{\mathrm{air}}\,(h_{\mathrm{air,out}} - h_{\mathrm{air,in}})}{W_{\mathrm{fan}} + W_{pumps} + Q_{\mathrm{w}} + Q_s}. \quad (6)$$

where, cooling capacity and energy consumed are in kW, h is enthalpy in kJ/kg, W refers to power consumed by fans and pumps and Q denotes the power required to cool the water (w) and solution (s).

Cooling capacity represents the energy removed from air through cooling and dehumidifying. The energy consumed is the electric and thermal power required for each cycle. This includes the power required by the fan and pumps (to overcome the pressure drop across the LAMEE) and the chillers.

Methodology

The first objective is to develop an empirical correlation for a 3-fluid LAMEE based on the experimental data collected by Abdel-Salam et al. (2016a). TRNSYS can implement the empirical correlations in a cooling and dehumidification cycle to predict system performance. Using experimental data limits the simulation model to operate solely within the range of the input conditions investigated in the experimental work. This leads to various restrictions. The main restrictions of the model include:

- Inlet solution temperature is constant at approximately room temperature (24.8°C)
- Air, solution, and water are the hot, intermediate, and cold fluids, respectively
- *NTU* is fixed at 1.8
- *Cr** is fixed at 1.8
- Sensible and latent effectiveness are functions of water flow rate when inlet solution conditions, *NTU*, and *Cr** are constant
- Regeneration requires no input energy

The simulation model uses non-linear regression models to describe the effectiveness at changing input parameters. Equations (7) and (8) present the empirical correlations used for sensible and latent effectiveness, respectively;

$$\varepsilon_s = 0.92 + 0.21\log(\dot{m}_w) + 0.0086 \cdot T_{w,in} \quad (7)$$

and

$$\varepsilon_l = 0.84 + 0.12\log(\dot{m}_w) - 0.0018 \cdot T_{w,in}. \quad (8)$$

The simulation model determines outlet air conditions using these regression models. Effectiveness is only a function of inlet water temperature and mass flow rate because of the constant inlet solution conditions. Similarly, the model determines outlet solution temperature by also implementing a regression model. Equation (9) presents the solution temperature correlation,

$$T_{s,out} = 32.39 - 6.85 \cdot \log(\dot{m}_w) + 0.43 \cdot T_{w,in} - 0.91 \cdot T_{air,in}. \quad (9)$$

By assuming energy conservation, the outlet water conditions are determined from the energy balance equation.

$$\dot{m}_{air}(h_{air,in} - h_{air,out}) + \dot{m}_s(h_{s,in} - h_{s,out}) + \dot{m}_w c_{p,w}(T_{w,in} - T_{w,out}) = 0 \quad (10)$$

There are two cooling systems, for water and solution, in Figure 2. The simple M-LDAC system assumes that the COP of these systems is one. Therefore, the cooling energy is equivalent to the energy required by the system.

Equation (11) determines the pump or fan power.

$$W = Q\Delta P \quad (11)$$

where Q is the volumetric flow rate (m^3/s) supplied by the pump or fan and ΔP is the pressure drop (Pa) across the fluid channel of interest. The Darcy-Weisbach equation (i.e., Equation (12)) defines the pressure drop across a channel (White, 1998). Pressure drop is calculated across the water, solution and air channels using equation (12).

$$\Delta P = f_D \cdot \left(\frac{L_{ex}}{D_h}\right) \cdot \frac{\rho \cdot u^2}{2} \quad (12)$$

where f_D is the friction factor and is defined as $\frac{64}{Re}$ when considering the laminar flow, L_{ex} is the length of channel (m), D_h is hydraulic diameter (m), ρ is the density of the fluid of interest and u is the average fluid flow velocity (m/s).

Table 1 summarizes the inlet conditions considered in this paper. The input air is warm and humid, solution is at room temperature, and the water temperature varies between 10.1°C and 24.6°C. The study considers constant air and solution flow rates and various water mass flow rate, from 1.22 g/s to 9.35 g/s.

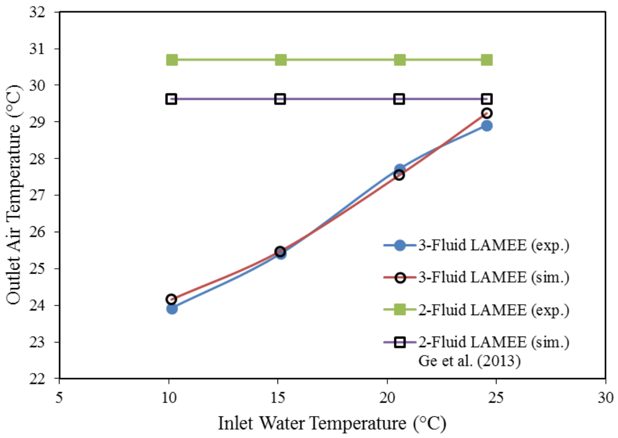
Table 1: Summary of the input conditions used for the 3-fluid LAMEE dehumidifier

	Input Parameter	Value	Unit
Air	Temperature, $T_{air,in}$	35.1	°C
	Humidity Ratio, $W_{air,in}$	17.6	(g_v/kg_{da})
	Mass flow rate	1.19	g/s
Solution	Temperature, $T_{sol,in}$	24.8	°C
	Concentration, C_{sol}	32.5	%
	Mass flow rate	0.76	g/s
Water	Temperature, $T_{w,in}$	10.1, 15.1, 20.6, 24.6	°C
	Mass flow rate	1.22, 1.98, 4.67, 9.35	g/s
Design & Operating Parameters	NTU	1.8	
	Cr*	1.8	
	Cr	0.055, 0.11, 0.26, 0.42	

Results and Discussion

Figure 3 illustrates the agreement between simulated and experimental outlet air temperatures and humidity ratios at different inlet water temperatures.

(a)



(b)

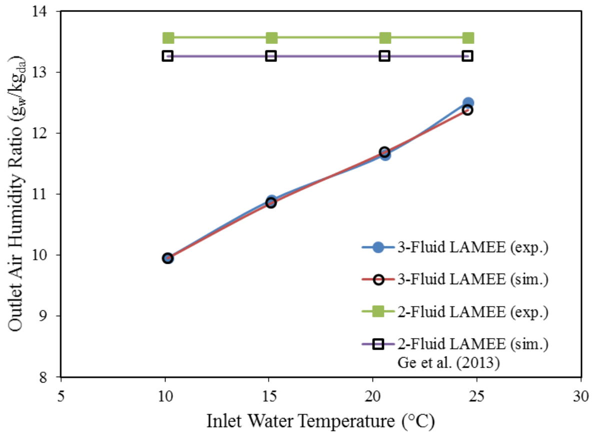


Figure 3: Simulated and experimental data of the a) outlet air temperature and b) outlet air humidity ratio at varying inlet water temperatures

The 2-fluid LAMEE simulation results serve as a benchmark for appropriate agreement between a simulation model and experimental data. Ge et al. (2013) created this 2-fluid LAMEE TRNSYS analytical component by modifying a heat and mass transfer analytical model for a hollow fibre membrane shell-and-tube energy exchanger, developed by Zhang (2011).

The simulated outlet air temperature and humidity ratio agree with their experimental counterpart as shown by the coinciding trend lines in Figure 3. Supply air conditions worsen when raising the inlet water temperature. That is, there is less potential for heat and moisture transfer and the supply air is more warm and humid. The supply air at a lower inlet water temperature is comfortable but cannot do any cooling. That is, the system provides neutral air. Out of the data points presented, the outlet air temperature and humidity ratio of 24°C and 10 g_w/kg_{da}, respectively, is the closest to thermal comfort.

Figure 4 presents the coefficient of performance for the experimental and simulated 2-fluid and 3-fluid M-LDAC systems at various inlet water temperatures. Since studies were limited to the component level, evaluating the experimental COP required using input and output data to determine the energy consumption and cooling capacity. The experimental and simulated COP of the 2-fluid M-LDAC system are evaluated at the same input conditions as the 3-fluid M-LDAC system.

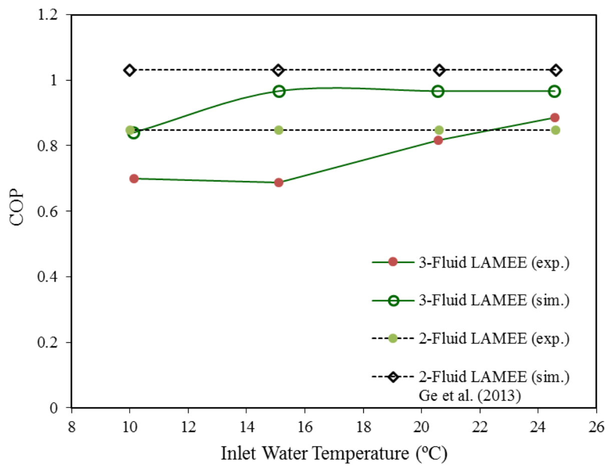


Figure 4: COP for the experimental and simulated 2-fluid and 3-fluid LAMEEs at various inlet water temperatures

The COP increases as the inlet refrigerant temperature increases because less cooling energy is required to return the refrigerant to its initial temperature. The temperature change across the water piping decreases as the inlet water temperature approaches the solution and air temperature. Subsequently, there is less heat transferred to the water and the cooling system requires less energy to cool the water to its initial temperature. However, as the inlet refrigerant temperature rises, the supply air becomes warmer and more humid (as shown in Figure 3 a) and b)).

The simulated COP of the 3-fluid M-LDAC exhibits an asymptotic trend. At higher inlet water temperatures, COP increases and approaches 0.97. Conversely, the COP drops at lower inlet water temperatures. Conditioning outdoor air to a temperature of 24°C and humidity ratio of 9.97 g_w/kg_{da} requires an inlet water temperature of 10°C and has a COP of 0.84. Figure 4 illustrates that the experimental COP of the 3-fluid M-LDAC system demonstrates asymptotic behaviour at cooler inlet water temperatures compared to the COP of the simulated system, which continues to decrease. The experimental COP increases with warmer inlet water temperatures. The COP is 0.70 when inlet water temperature is 10°C and air is supplied at 24°C and 9.97 g_w/kg_{da}.

Discrepancy in the experimental and simulated COP, for the 3-fluid M-LDAC system, is a result of disagreement in outlet water temperatures. The simulation assumes complete energy conservation and determines outlet water temperature using the energy balance equation (i.e., Equation (10)). Therefore, the measured water temperature accounts for the energy imbalance observed in experiment. The largest energy imbalance occurs at an inlet water temperature of 15°C that can be seen in Figure 4 as the point with the largest discrepancy between the simulated and experimental COPs. The model is optimistic and predicts that the simulated outlet water temperature is cooler than that of the experiment. Therefore, the water needs less cooling energy in the simulation compared to the experiment which results in a higher COP in the simulation.

The difference between simulated and experimental COP of the 2-fluid M-LDAC system is 20%. The 3-fluid M-LDAC model is in better agreement than this except at the inlet water temperature of 15°C. At this temperature, the difference is 34%. Otherwise, the COP difference is below 17%

The 3-fluid system cools and dehumidifies the air much more than the 2-fluid system at lower inlet water temperatures but the COP drops. A comparison of COPs is only relevant when the systems are supplying air with similar conditions. Figures 3(a) and 3(b) illustrate that the 3-fluid M-LDAC system supplies air at similar temperatures and humidity as the 2-fluid system when the inlet water temperature is 25°C. The COP of the 2-fluid and 3-fluid M-LDAC systems can be compared at this point. Figure 4 illustrates that the 3-fluid M-LDAC system (COP=0.89) performs more efficiently than the 2-fluid system (COP=0.85) in the experimental results. Additionally, Figures 3(a) and 3(b) show the supply air from the 2-fluid M-LDAC system is less desirable (i.e., more warm and humid). This system requires additional input energy to further cool and dehumidify the air, thereby decreasing the system COP and increasing the gap between the 2 and 3-fluid M-LDAC system experimental COPs.

The simulation models predict better performance than the experiment data measured on the 2-fluid M-LDAC system as presented in Figure 4. The 2-fluid and 3-fluid COPs are 1.03 and 0.97, respectively. Figure 3(a) shows the supply air temperatures is approximately 29°C for both systems while Figure 3(b) shows that the simulated 3-fluid system dehumidifies more effectively. The 2-fluid system requires more energy to dehumidify to a similar humidity as its 3-fluid counterpart. Subsequently, the COP will decrease. It is possible to conclude that the simulated systems have similar COPs.

Conclusion

The TRNSYS empirical model of the 3-fluid LAMEE accurately predicts output water, solution, and air conditions. The simulated results agree with the available experimental data under similar input conditions. The major limitations of the simulation model include:

i. Operating (*Cr**) and design (*NTU*) parameters are both fixed at 1.8
ii. Limited to the 3-fluid LAMEE designed by Abdel-Salam et al. (2016a).

With this model, it is possible to simulate the simple 3-fluid M-LDAC system. Decreasing inlet water temperature and increasing water mass flow rate enhances cooling and dehumidifying but decreases COP. At the specified input conditions, the 3-fluid M-LDAC system demonstrates a COP of 0.70-0.84 to obtain neutral air conditions. The experimental 3-fluid M-LDAC system performed better than the experimental 2-fluid M-LDAC system while the simulated 2-fluid and 3-fluid systems performed similarly.

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Appendix A

Table A.1: Summary of the 3-fluid LAMEE dimensions (Abdel-Salam et al., 2016a)

	Parameter	Value	Unit
Exchanger	flow configuration (solution-air)	counter-cross	-
	flow configuration (solution-refrigerant)	counter	-
	Length	470	mm
	Height	100	mm
	exchanger solution entrance ratio*	0.11	-
	nominal air channel width (δ_{air})	5	mm
	nominal solution channel width (δ_{sol})	4.2	mm
	number of air channels	2	-
	number of solution channels	1	-
	mass (empty)	1.7	kg
	desiccant solution	LiCl	-
Membrane	Thickness	0.3	mm
	mass resistance (R_m)	38	s/m
	liquid penetration pressure	124	kPa
Refrigeration tubes	Refrigerant	water	-
	tube material	titanium	-
	number of tubes	7	-
	tube length	660	mm
	inner diameter	2.362	mm
	outer diameter	3.175	mm
	Thickness	0.4	mm
	spacing between tubes	9.7	mm
	thermal conductivity	21	(W/(m·K))

**entrance ratio is defined as the ratio between the length of the inlet/outlet desiccant solution header and the length of the exchanger* (Vali 2009).