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Simplified model of a membrane-based rectangular micro-desorber for absorption chillers

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Abstract

The simulation of the heat and mass transfer in a H₂O–LiBr microchannel desorber endowed with a microporous membrane is presented. The heat and mass transfer processes are modelled by means of selected correlations gathered from the open literature. The simulation provides the evolution along the channels of the parameters involved in the process: heat and mass transfer coefficients, solution concentration, temperatures of the working fluids and pressure potential. The calculated values of the desorption rate are compared to experimental data gathered from the open literature. For the case considered in this study, the maximum ratio between cooling capacity of the chiller and desorber volume is 2312 kW m⁻³. This value is more than one order of magnitude higher than the ones found in conventional small-size absorption cooling chillers.

Keywords: Absorption refrigeration; Membranes; Desorber; Water–lithium bromide; Rectangular microchannels

Nomenclature

A	area (m ²)
Bo	Boiling number, $Bo = q''/G_s \cdot i_{lv}$
C_p	specific heat (kJ kg ⁻¹ K ⁻¹)
D	diffusion coefficient (m ² s ⁻¹)
D_h	hydraulic diameter (m)
d_p	membrane pore diameter (m)

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dz	discretisation length (m)
e	height or thickness (m)
G	mass flux, $G = \dot{m}/(l \cdot e)$ ($\text{kg m}^{-2} \text{s}^{-1}$)
h	convective heat transfer coefficient ($\text{W m}^{-2} \text{K}^{-1}$)
i	specific enthalpy (kJ kg^{-1})
J	desorption rate ($\text{kg m}^{-2} \text{s}^{-1}$)
k	mass transfer coefficient (m s^{-1})
K	mass transfer coefficient ($\text{kg Pa}^{-1} \text{m}^{-2} \text{s}^{-1}$)
Kn	Knudsen number
l	width (m)
L	total length of channels (m)
$\dot{m}$	mass flow rate (kg s^{-1})
M	molecular weight (kg mole^{-1})
Nu	Nusselt number, $Nu = hD_h/k$
P	pressure (Pa)
q''	heat flux (W m^{-2})
q	thermal power (W)
R	mass transfer resistance ($\text{kg}^{-1} \text{Pa m}^2 \text{s}$)
R^2	correlation coefficient
Re_s	Solution Reynolds number, $Re_s = G_s D_{h,s}/\mu_s$
Re_{vd}	Vapour Reynolds number, $Re_{vd} = G_s y D_{h,s}/\mu_v$
R_u	universal gases constant ($\text{J mole}^{-1} \text{K}^{-1}$)
Sc	Schmidt number, $Sc = \mu/\rho D$
Sh	Sherwood number, $Sh = kD_h/D$
T	temperature ($^{\circ}\text{C}$)
U	global heat transfer coefficient ($\text{W m}^{-2} \text{K}^{-1}$)
x	lithium bromide mass fraction ($\text{kg}_{\text{LiBr}} \text{kg}^{-1}$)
X_{tt}	Martinelli parameter, $X_{tt} = [(1 - y)/y]^{0.9} [\rho_v/\rho_l]^{0.5} [\mu_l/\mu_v]^{0.1}$
y	vapour mass fraction
z	length (m)

Greek symbols

α	channel aspect ratio, $\alpha = l/e$
ΔT	temperature difference (°C)
ε	porosity
λ	thermal conductivity ($\text{W K}^{-1} \text{m}^{-1}$)
Ξ	Ackermann factor
μ	viscosity (Pa s)
ρ	density (kg m^{-3})
τ	tortuosity
Φ	heat transfer rate factor

Subscripts

e	evaporator
hw	hot water
i	inlet
l	liquid
m	membrane
o	outlet
ov	overall
s	solution
sat	saturation
th	thermal
T	total
v	vapour
vd	vapour desorbed
w	wall

1. Introduction

The contribution to the electricity consumption of the building air-conditioning sector is increasing as the technology is still mainly based on mechanical vapour compression systems. Moreover, in many cases, the peak load in the grid is greatly

influenced by the contribution of these devices. This drawback can be avoided by the use of absorption systems. This technology can employ the solar radiation that occurs simultaneously with the peak cooling demand to feed the generator of the absorption chiller and, in this way, the solar thermal energy in the summer period can help to reduce the electricity consumption peak due to air-conditioning.

This thermally driven cooling technology has been already applied but preferably in large capacity applications. Its use in low and medium power applications is limited as the scale-down to smaller sizes has not been completely achieved. For instance, a typical value for the refrigeration power to volume ratio in single effect absorption chillers is in the order of 25 kW m^{-3} (without considering the volume occupied by the cooling system) for refrigeration capacities between 10 to 30 kW, whereas mechanical compressor systems can have a ratio equal to 50 kW m^{-3} for the same range of refrigeration capacities (García-Hernando et al. [1]). The need of mass and heat exchangers to suppress the conventional mechanical compressor is responsible of this increase in size. In particular, the desorber or generator is one of these limiting and volume demanding components. Traditionally, shell-and-tubes heat exchangers have been used as desorbers, but plate heat exchangers are now being used as well, due to their lower cost and size (de Vega et al. [2], Venegas et al. [3]). At present a promising new technology is under study, consisting in the use of membrane contactors in microchannel heat exchangers.

Membrane contactors can be used in order to create an interface for mass transfer between two phases. The process described in this paper involves the use of a microporous membrane to desorb water vapour from a H_2O –LiBr solution. The membrane is hydrophobic and the H_2O –LiBr solution cannot penetrate inside the membrane pores. Hence, the vapour/liquid interfaces are located at each pore entrance. The water desorption flux from the liquid to the vapour side is driven by the water vapour pressure difference between both phases. This vapour pressure gradient causes a mass-transfer across the membrane. Thus, this process can be able to desorb the vapour below the boiling point of the solution considered. The membrane offers a large interfacial area and presents interesting working conditions, like lower operating temperature and easier solution-vapour separation, as the vapour can be separated without the flow boiling regime, which may contribute to a large pressure drop.

Moreover, the proposed desorber allows easier scale up, modular design and compactness.

The first experimental study available in the open literature regarding the use of membranes in desorbers of absorption chillers was developed by Sudoh et al. [4]. The authors conducted permeation experiments of water vapour in membrane distillation of a H_2O –LiBr solution. The effects of the stirring rate and the temperature difference on the permeation flux in a batch cell were discussed.

Riffat et al. [5] experimentally investigated a novel absorption refrigeration system using membranes in the desorber. They used a solution different to the most widely used in commercial absorption chillers, aqueous potassium-formate, and two membrane modules, a dense membrane and a silicon porous one. The authors concluded that the salt progressively blocks the pores of the silicon membrane, probably implying that porous membranes may not be suitable for salt-based working fluids.

Drost et al. [6] carried out a series of experiments to study the water vapour separation of an ultra-thin film of H_2O –LiBr mixture using a microporous membrane. The desorption rate had a linear relationship with the heat flux, for a given inlet mass fraction (weight percentage of salt) and flow rate. The solution exit mass fraction increased when inlet mass flow rate decreased for a given heat flux.

In an experimental study, Thorud et al. [7] investigated the vapour desorption from a H_2O –LiBr mixture, also using a hydrophobic porous membrane. Results showed that the rate of vapour separated from the solution increased for reduced channel thickness, increased pressure drop across the membrane, and lower solution concentration at the inlet of the desorber. The authors showed also that a lower wall superheat corresponds to a higher heat transfer coefficient. The formation of bubbles in the channel did not decrease the overall performance since the critical heat flux condition was not reached.

Kim et al. [8] presented a numerical study of water desorption from a H_2O –LiBr solution using micro-channel arrays for the desorber and condenser of a miniature heat pump system. These devices were placed in intimate communication using a hydrophobic membrane. The authors concluded that the results demonstrate the feasibility of using miniature absorption refrigeration systems for electronics cooling applications, but they recommended an experimental validation for quantitative support of the theoretical conclusions.

Wang et al. [9] developed an experimental test to study the permeation performance of membranes in a desorber using the H_2O – LiBr mixture. They used a hollow fibre membrane module made of polyvinylidene fluoride, showing a high rejection ratio of the water-lithium bromide solution. The authors concluded that, based on the properties of low temperature driving in membrane distillation and the high contact area of hollow fibre membrane modules, this technique can be used as a novel desorber or as a secondary desorber capable of using low temperature heat, such as solar energy, or to increase the COP of the whole chiller.

Isfahani et al. [10] presented a study on the efficacy of highly porous nanofibrous membranes for application in membrane-based desorbers. Permeability studies showed that membranes with a pore size smaller than 1 micron are valid for application in the desorber. In a following work, Isfahani et al. [11] obtained desorption rates one order of magnitude higher than the results achieved by Thorud et al. [7].

Bigham et al. [12] used a combination of experimental and numerical studies to illustrate the desorption process from a solution flow confined within microchannels. Their numerical model is used to study the impact of surface microstructures on the desorption process.

Recently, a review of membrane contactors applied in absorption refrigeration systems has been presented by Asfand and Bourouis [13]. They concluded that, at low desorber temperatures, desorption rates can be enhanced by direct diffusion. In this way, the absorption cooling systems using membrane contactors can operate at lower generation temperatures.

As it can be observed from the literature previously cited, researchers have paid attention to the use of microporous membrane in desorbers of absorption cooling systems as a highly promising technology, but theoretical studies and experimental results remain scarce. Further, different operating conditions, membrane types, and flow channels configurations have been modelled and tested in each case, rendering difficult to obtain a general and deeper understanding of the thermal phenomena encountered in membrane-based desorbers.

The present study focuses in the modelling of the heat and mass transfer processes in a microporous membrane-based rectangular micro-desorber working with the water-lithium bromide solution, providing a method to optimize the desorption channel length.

The model developed now is based on a previous validated model used to simulate the performance of a membrane-based micro-absorber (Venegas et al. [14]).

2. Modelling of the desorption process

The configuration considered for the desorber in the present study is a plate-and-frame membrane module. It consists of the contact membrane, separating the vapour and the H_2O –LiBr solution confined in rectangular microchannels. The heating is provided by hot water circulating also in microchannels and separated from the solution by a wall interface. A cross-section of the proposed micro-desorber is shown in Fig.1. The geometrical characteristics and operating variables are provided in Table 1.

In order to simplify the problem, the following assumptions are considered:

- The system operates under steady state conditions.
- The solution, hot water and refrigerant vapour completely fill their respective channels: no incondensable gases are present and vapour desorption takes place instantaneously.
- The plate-and-frame membrane module is adiabatic with respect to its surroundings.
- An equilibrium state exists at the vapour/liquid interface.
- Solution, hot water and vapour are well-mixed in each differential volume. Homogeneous temperature and concentration are considered in all points within each respective control volume. A similar assumption of well-mixed fluid was already used by Abolhasani et al. [15] in a similar problem. Given the small dimensions of each discretized control volume and the movement of the solution produced by bubble generation, motion and escape across the membrane, it could be a reasonable assumption to consider a well-mixed solution.
- The heat and mass transfer processes are one dimensional.
- Pressure is constant along the channels. As a result, the solution equilibrium concentration is only a function of the solution temperature.

In this work, the heat and mass transfer processes in the membrane-based desorber have been modelled using mass and energy balances. Equations used in the model are

directly based on fundamental physics or empirical correlations derived from experiments. A similar methodology has been extensively used in the literature in membrane distillation.

The mass and energy balances, and all the equations used, are discretised using differential elements, according to Fig. 2. The discretisation and the iterative procedure employed allow the calculation of the bulk temperature and concentration of each next element by adjusting it with the exchanged mass and energy of the previous one. By calculating the temperature and concentration in each element, the temperature and concentration profiles along the channel can be predicted.

In the model the bubble formation and growth is considered independently for each control volume. The rate of vapour generation in each control volume is considered to be equal to the rate of vapour flowing across the membrane at the end of that control volume. No storage of vapour in the solution channel is considered from one control volume to the subsequent. According to Isfahani et al. [11], during the direct diffusion desorption mode, the vapour desorbed closely matches the vapour flow through the membrane. However, in the boiling regime, some vapour is not able to escape across the membrane and remains in the solution channel. The same authors state that the difference between vapour desorbed and vapour crossing the membrane reaches a maximum of 15%. It could be one of the reasons for the overprediction of the present model respect to experimental data of Isfahani et al. [11]. The development of a model including the percentage of vapour stored in the solution channel is a challenging task. Further experimental research is required to improve insights on this process because it depends on the operating conditions (as shown in Isfahani et al. [11]) and also of the geometric characteristics of the channels and the membrane.

The main advantage of the present model is that it can provide an immediate answer, reducing computational cost and offering relevant information of the heat and mass transfer phenomena. Other advantages of the present model include:

- Less computational time than detailed CFD models. Calculation time is less than 31 seconds.
- Simplicity of the model allows its implementation using any programming language.
- No hypothesis about temperature distribution along the heating surface and outlet boundary conditions are required.

The present model, up to the authors' knowledge, is the first non-CFD model developed and validated using experimental data of a membrane-based microchannel desorber. The heat transfer model considers that temperature is changing along both thermal entrance and fully developed flow regions for the single-phase flow. Solution mass transfer resistance is taken into account and desorption taking place in the boiling region is evaluated.

2.1. Mass transfer model

Mass rate balances for solution and vapour give the mass flow rates in the differential element $j+1$:

$$\dot{m}_s^{j+1} = \dot{m}_s^j - \dot{m}_{vd}^j \quad (1)$$

$$\dot{m}_v^{j+1} = \dot{m}_v^j + \dot{m}_{vd}^j \quad (2)$$

Mass fraction of lithium bromide in the solution is calculated from:

$$x^{j+1} = x^j \frac{\dot{m}_s^j}{\dot{m}_s^{j+1}} \quad (3)$$

Mass flow rate of vapour desorbed from the solution and transported across the membrane is proportional to the desorption rate J and the heat transfer area A for each of the j differential control volumes:

$$\dot{m}_{vd}^j = J^j \cdot A \quad (4)$$

$$A = l_s \cdot dz \quad (5)$$

The driving force of the process is the difference of vapour pressure and therefore the desorption rate can be expressed as:

$$J^j = K^j (P_s^j - P_v) \quad (6)$$

P_s is the water vapour partial pressure corresponding to the bulk solution concentration (x) and temperature (T_s), and P_v is the bulk vapour pressure.

To obtain the desorption rate the classical resistances-in-series model is followed. In this case, the overall mass transfer coefficient between the bulk water vapour and bulk aqueous solution (K) includes the resistance to diffusion through the aqueous solution boundary layer (K_b) and the resistance to diffusion of water vapour through the membrane active layer (K_m).

The mass transfer coefficient in the solution K_b can be calculated according to Ali and Schwerdt [16], as:

$$K_b^j = \left(\frac{P_{sat}^j}{\rho_{water}^j k_s^j} \right)^{-1} \quad (7)$$

where P_{sat} is the saturated water pressure corresponding to the bulk solution temperature, ρ_{water} is the liquid water density and k_s is the mass transfer coefficient between the solution-vapour interface and the bulk aqueous solution.

Mass transfer coefficient k_s has been calculated taking into account two main regions:

- Single phase flow, characterized by a direct diffusion mode. It occurs if $T_s - T_{sat} < 0$. Correlation of Lvque (Norris et al. [17]) has been used to predict mass transfer in this region.

$$Sh = 1.85 \left(Re_s Sc_s \frac{D_h}{z} \right)^{1/3} \quad (8)$$

- Two-phase flow, where mass transfer takes place in the boiling mode. It occurs if $T_s - T_{sat} \geq 0$. Correlation of Ganapathy et al. [18] has been used in this zone.

$$Sh = 1.689 \cdot 10^{-4} Re_{vd}^{0.223} Re_s^{0.829} Sc_s^{1.766} \quad (9)$$

This selection was performed after a thorough review of mass transfer correlations available for multiphase flows in microchannels.

The use of Sherwood and Schmidt dimensionless numbers to obtain the solution mass transfer coefficient requires knowledge of the diffusion coefficient. In this study, it was calculated and corrected by temperature using the equations described by Mittermaier et al. [19].

In the case of the membrane, the dusty-gas model is used to describe the vapour transport within the membrane pores. The corresponding mass transfer coefficient through the membrane (K_m) is estimated as described in Mason and Malinauskas [20]. In the present work, the membrane works in the transitional regime (the Knudsen number ranges from 0.01 to 10) and K_m accounts for two terms, corresponding to the free molecular flow and the viscous flow:

$$K_m = \frac{M}{e_m} \left(\frac{D_e^K}{R_u T_m} + \frac{P_m B_0}{R_u T_m \mu_v} \right) \quad (10)$$

The diffusion term can be evaluated as:

$$D_e^K = \frac{\varepsilon d_p}{3\tau} \left(\frac{8 R_u T_m}{\pi M} \right)^{0.5} \quad (11)$$

and the molar viscous flow contribution can be calculated from:

$$B_0 = \frac{\varepsilon d_p^2}{32\tau} \quad (12)$$

The tortuosity of the membrane, τ , is calculated as a function of the membrane porosity, according to Iversen et al. [21]:

$$\tau = \frac{(2-\varepsilon)^2}{\varepsilon} \quad (13)$$

2.2. Heat transfer model

The energy rate balance applied to the differential element j is written as:

$$q_{hw}^j = (\dot{m}_{vd} i_{lv})^j + q_s^j + q_v^j \quad (14)$$

Left term in Eq. (14) corresponds to the thermal power transferred by the hot water. Right terms are related to the heat absorbed during desorption of the vapour flow rate $\dot{m}_{vd}$ from the solution, and the heat stored in the solution and the vapour, respectively. These can be calculated, for parallel flow, as:

$$q_s^j = (\dot{m} \cdot i)_s^{j+1} - (\dot{m} \cdot i)_s^j \quad (15)$$

$$q_{hw}^j = \dot{m}_{hw} (i_{hw}^j - i_{hw}^{j+1}) = \dot{m}_{hw} C p_{hw}^j (T_{hw}^j - T_{hw}^{j+1}) \quad (16)$$

$$q_v^j = (\dot{m} \cdot i)_v^{j+1} - (\dot{m} \cdot i)_v^j \quad (17)$$

The heat is transferred from the hot water channel to the bulk solution channel (where desorption takes place) across the metallic wall and from the solution to the vapour across the membrane:

$$q_{hw}^j = U_{s,hw}^j A (T_{hw}^j - T_s^j) \quad (18)$$

$$q_v^j = U_{s,v}^j A (T_s^j - T_v^j) \quad (19)$$

The global heat transfer coefficients in Eqs. (18) and (19) are calculated as:

$$\frac{1}{U_{s,hw}^j} = \frac{1}{h_{hw}^j} + \frac{e_w}{\lambda_w^j} + \frac{1}{h_{s,hw}^j} \quad (20)$$

$$\frac{1}{U_{s,v}^j} = \frac{1}{h_v^j} + \frac{e_m}{\lambda_{m,ave}^j} + \frac{1}{h_{s,v}^j} \quad (21)$$

The average thermal conductivity of the membrane $\lambda_{m,ave}$ is calculated using the classical parallel model, according to Martínez and Rodríguez-Maroto [22]:

$$\lambda_{m,ave} = \varepsilon \lambda_v + (1 - \varepsilon) \lambda_m \quad (22)$$

In Eq. (22), λ_v is the thermal conductivity of the vapour inside the membrane pores, while λ_m is the thermal conductivity of the membrane solid material. In this study, PTFE is selected and its thermal conductivity is calculated using the correlation of Sperati et al. [23]:

$$\lambda_m = (4.86 \cdot 10^{-4} T_m) + 0.253 \quad (23)$$

In Eqs. (20) and (21), the convection heat transfer coefficients along the solution, hot water and vapour channels have been calculated using the procedures described in the next subsections 2.2.1 and 2.2.2.

2.2.1. Solution channels

In the solution channels, of hydraulic diameter $D_{h,s} = 272.7 \mu\text{m}$, the flow is divided into two main regions:

- Single phase flow: It occurs if $T_s - T_{sat} < 0$. Here the flow is subdivided into two regions:
 - Thermal entrance region: Correlations of Lee and Garimella [24] for microchannels were used to estimate the length of the thermal entrance region and the convection heat transfer coefficients in this region. With this correlation the heat transfer coefficient can be evaluated in each position along the channel.
 - Fully developed flow: Convection heat transfer coefficients along the microchannels were obtained using the correlation developed by Shah and London [25].
- Two-phase flow: It occurs if $T_s - T_{sat} \geq 0$. The composite correlation developed by Bertsch et al. [26] is used to calculate the heat transfer coefficients. The critical pressure of the solution was estimated according to the method of Lydersen, described in Sladkov and Pobedonostseva [27]. The critical pressure of the lithium bromide was not found in the open literature consulted. For this reason its critical pressure was supposed equal to that of lithium, 68.9 MPa, taken from Weast et al. [28]. The influence of the critical pressure is negligible in the case considered in the present study. If a critical pressure of 10.34 MPa (critical value for Br) is used, the average desorption rate modifies

only 0.047%. In the present study the surface roughness (according to DIN 4762) was supposed equal to 1 following criteria stated by Bertsch et al. [26] for cases where the surface roughness is unknown.

The convection heat transfer coefficients obtained were multiplied by the Ackermann factor (Taylor and Krishna [29]) to take into account the effect of mass transfer on heat transfer. This factor is:

$$\Xi = \frac{\Phi}{e^{\Phi}-1} \quad (24)$$

where the heat transfer rate factor Φ is defined as:

$$\Phi = J \cdot C p_v \cdot e_s / \lambda_v \quad (25)$$

2.2.2. Hot water and vapour channels

In the case of the hot water channels, with $D_{h,hw} = 272.7 \mu\text{m}$, the same equations of Lee and Garimella [24] and Shah and London [25] employed for the single phase flow of the solution were used.

The vapour channel is a conventional rectangular channel with $D_{h,v} = 8 \text{ mm}$. In this case, the dimensionless thermal entry length is calculated using the correlation derived by Venegas et al. [14] using data tabulated by Shah and London [25] for the case of simultaneously developing flow. Data tabulated also by Shah and London [25], for the case of constant heat flux, were used by Venegas et al. [14] to derive the correlation for Nu in this developing region. It is valid for the aspect ratio of the vapour channel used in the present work, $\alpha_v = 4$. Finally, for the thermally developed flow, the correlation derived by the same authors from data tabulated by Shah and London [25] for the case of constant heat flux is used.

2.3. Resolution

The concentration and temperatures along the channels cannot be explicitly determined from Eqs. (1)–(25). For this reason, the above set of equations should be solved iteratively. They have been compiled in a computer code developed by the authors using Engineering Equation Solver software, EES™ (Klein [30]). Fig. 3 shows the flowchart used for implementing the heat and mass transfer models.

The thermodynamic and transport properties were calculated using correlations provided by EESTM. Thermodynamic properties for the water-lithium bromide solution are the ones of Harr et al. [31]. The viscosity and thermal conductivity are computed from Lee et al. [32] and DiGuilio et al. [33], respectively. Thermodynamic properties of water have been calculated using correlations developed by Patek and Klomfar [34], while its transport properties are calculated using equations of the Electrical Research Association [35]. Metal of the wall interface was considered AISI 316 and its thermal conductivity was calculated using correlation of Ho and Chu [36]. Surface tension of the water-lithium bromide solution was calculated using the following correlation, derived in this research from data tabulated by Herold et al. [37], as a function of the solution concentration ($50 \leq x \leq 60$) and temperature ($70 \text{ }^{\circ}\text{C} \leq T \leq 90 \text{ }^{\circ}\text{C}$):

$$\sigma = 8.48330476 \cdot 10^{-2} + 2.586 \cdot 10^{-2}x + 7.6 \cdot 10^{-3}x^2 - 3.98752381 \cdot 10^{-4}T + 1.24761905 \cdot 10^{-6}T^2, \text{ with } R^2 = 0.9995 \quad (26)$$

2.4. Model validation

Validation of the model has been performed comparing the desorption rate predicted with the experimental data reported by Isfahani et al. [11]. In the experimental case, the solution channel measures 1 mm and 0.2 mm in width and height, respectively. An important issue is that the correlations used in the present model remain valid in that case.

Figs. 4 and 5 show a comparison between the desorption rate predicted by the model, using the operating and design data of Isfahani et al. [11], and their experimental results up to a wall temperature of 120 °C. Fig. 4 shows the variation of the desorption rate as a function of the vapour pressure and the wall temperature. In both model and experiments, when the vapour pressure increases the desorption rate decreases. Also, a better desorber performance is obtained in both cases for higher wall temperatures. Only larger discrepancies are found for the lower vapour pressure.

The influence of the solution mass flow rate on the desorption rate is represented in Fig. 5 as a function of the wall temperature. Again, the model predicts a tendency similar to the experiments. The increase of the solution mass flow rate has a negligible influence on mass transfer, while the rise in the wall temperature (also observed in Fig. 4) enhances the driving force for mass transfer, due to an increase of the solution temperature.

The mean absolute error of the model predictions respect to all the experimental data represented in Figs. 4 and 5 is 26.4%. It has to be pointed out that Isfahani et al. [11] also presented a model which deviates from their experimental results, and the authors suggested some factors that could affect this behaviour:

- Solution used in the experiments contains additives that could impact the solution properties.
- Surface temperature and liquid film thickness can be different to values specified.
- Mass fraction of salt in the solution was not constant because it changed from 48% to 51%.

3. Results and discussion

3.1. Evolution along the channel

For the initial conditions considered in Table 1, the inlet solution temperature is slightly higher than the saturation temperature corresponding to the inlet solution concentration and pressure. For this reason the solution starts boiling since the channel inlet. In Fig. 6 the pressure potential, which is the driving force of the process, is shown, simultaneously with the solution concentration change. The increase in LiBr mass fraction describes the water vapour desorption process. In the first centimetre the pressure potential increases rapidly, but as the concentration augments, the difference in pressure tends to decrease.

Fig. 7 shows the overall mass transfer coefficient obtained. This overall desorption coefficient is the result of the combination of membrane and solution mass transfer coefficients that are also represented. As it is observed in Fig. 7, major resistance to mass transfer (lower desorption coefficient) is obtained in the membrane for the operating conditions in Table 1. This means that desorption is controlled by the membrane resistance, revealing the importance of optimizing its geometrical properties to improve the desorption rate. One limitation of the membrane is the maximum pore diameter allowable to avoid aqueous solution penetration into the membrane pores, as discussed by Ali and Schwerdt [16]. These theoretical results have not been found in the open literature of microporous membranes applied to absorption chillers, to our knowledge.

In Fig. 8, the resulting desorption rate is shown. When the higher pressure potential is available the desorption rate is higher. This value tends to decrease as the solution desorbs the vapour along the channel, and the driving potential (the difference in pressure) decreases. This is confirmed by the evolution of the pressure potential shown in Fig. 6. Desorption rates of $8.6 \cdot 10^{-3} \text{ kg m}^{-2} \text{ s}^{-1}$ on average are obtained in the channel.

Variation of the solution and hot water temperatures along the channels is shown in Fig. 9. Temperature of the solution increases rapidly, due to heat transfer from the hot water, until 1 cm length approximately. From this point, the heat transferred from the hot water is mostly used to evaporate the refrigerant from the solution. The first centimetre of rapidly increasing solution temperature observed in Fig. 9 directly relates to the region of increasing desorption rate observed in Fig. 8. It implies that, the higher the solution temperature is, the higher the potential for mass transfer will be. Beyond 1 cm length the solution temperature remains high, but the solution concentration is higher. For this reason the corresponding solution vapour pressure tends to decrease, reducing the potential for mass transfer. The continuous increase in the solution temperature is limited in real systems by the hot water inlet temperature and the efficiency of the heat exchange process.

Laminar flow is expected to occur in all the cases. In the conditions evaluated, Reynolds numbers range between 137–155, 21–23 and 0–56 for the hot water, solution and vapour, respectively. Velocity of the hot water and solution are practically constant, at 18.4 and 11.1 cm s^{-1} respectively. Vapour velocity increases up to 2.3 m s^{-1} .

Very high convection heat transfer coefficients are predicted in the present study for the solution and hot water flows, as shown in Fig. 10. In this case the thermal entrance region of the hot water flow is very tiny, of about 1.6 mm length. The strong change observed in the figure for the water heat transfer coefficient corresponds to this point. Additionally, in the fully developed flow region, as the aspect ratio of the channel is constant, no significant change is observed in the convection coefficient. Regarding solution flow, the modified heat transfer coefficient (taking into account the effect of mass transfer) is shown. The modified convection coefficient is higher than without mass transfer because desorption takes place in the same direction of heat transfer. In the case of the vapour flow, very low convection coefficients are predicted, 14.4 $\text{W m}^{-2} \text{ K}^{-1}$ on average. Along the major part of the vapour channel the flow can be considered as a thermally developing flow.

Fig. 11 represents the evolution of the thermal power stored in the solution and transferred by the hot water along the desorption channel. In the first part of the channel most of the heat is transferred to the solution, corresponding to the rapid increase in the solution temperature observed in Fig. 9. After 1.7 cm length, heat transferred by the hot water is exclusively used to vaporise the refrigerant. Along the whole desorber, the contribution of the thermal power transferred to the vapour is almost two orders of magnitude smaller.

The cooling power of an absorption chiller equipped with the desorber modelled in the present work is represented in Fig. 12 as a function of the channel length. The equation used to calculate this cooling power is:

$$q_e = \dot{m}_{vd}(h_o - h_i) \quad (27)$$

The enthalpy at the outlet of the evaporator, h_o , is calculated considering saturated vapour at a temperature of 5 °C. Enthalpy at the inlet, h_i , taking into account isenthalpic expansion in the valve, corresponds to saturated liquid at the pressure in the vapour channel of the desorber.

Also, the ratio between this cooling power and the desorber volume is shown in Fig. 12. As observed, the ratio suddenly increases with the channel length, reaching one half of the maximum value at 2 mm length approximately. From this point, the ratio increases more slowly and the maximum value (optimal value) is reached at 2.3 cm length for the geometrical and operating conditions given in Table 1. So, in order to optimize the desorber size, when designing the desorber it is not recommended to use very long channels, because the increase of the desorber size can be higher than the increment obtained in the cooling power. This is an important conclusion derived from the results of the present model. To the best of our knowledge, it is the first time the ratio between the cooling power and the desorber volume is used to optimize the size of membrane-based desorbers.

The maximum value of the ratio between cooling power and desorber volume is 2312 kW m⁻³. This ratio is higher than values found in desorbers of small-size absorption chillers (4 to 15 kW of cooling capacity) using the same solution, according to data offered by Asdrubali et al. [38]. In that study, the ratio between the cooling capacity and chiller volume is given, reaching a maximum value of 9.9 kW m⁻³. Estimating the generator volume as one tenth of the chiller volume, the ratio between the cooling power and desorber volume reaches 99 kW m⁻³. This value is more than one order of magnitude lower

than the values found in the present work. When the comparison is performed with absorption chillers of the highest capacity available in the market, higher than 4 MW, the ratios obtained are about one half of those reached in this study. For this reason, the membrane-based micro-desorbers are specially recommended for low-capacity absorption chillers.

Fig. 13 shows the evolution of COP as a function of the desorber length. As it is observed the COP continuously increases, but its increment is negligible beyond one fifth of the channel length. The ratio of the cooling power and the electric power shows the same evolution of the cooling power, as the pump consumption is constant. It depends only of the solution mass flow rate, temperature and concentration at the desorber inlet, and the operating pressure difference between evaporator and desorber. These variables are constant for the case evaluated in the present study, corresponding to a power consumption of 1.45 mW.

3.2. Comparison between direct diffusion and boiling desorption modes

With the aim of showing the effect of boiling on desorption, Fig. 14 shows the change of the desorption coefficient from direct diffusion mode to boiling regime along the channel (at about 2 mm length). In this case the solution pressure has been increased up to 6.2 kPa. The rest of geometrical and operating data are those of Table 1. The dominant mass transfer mode in this case is the boiling one, because the solution inlet subcooling is low. For the same reason, the dominant heat transfer mode is the one associated to boiling. Also, it has been evaluated which contribution (nucleate or convective) prevails during boiling. With this aim, data predicted with the model has been analyzed according to the criterion of Thonon et al. [39], using the Boiling number and the Martinelli parameter X_{tt} :

$$Bo \cdot X_{tt} > 0.15 \cdot 10^{-3} \Rightarrow \text{Nucleate boiling}$$

$$Bo \cdot X_{tt} < 0.15 \cdot 10^{-3} \Rightarrow \text{Convective boiling}$$

As result, all boiling data fall in the nucleate boiling zone, with $Bo \cdot X_{tt}$ always higher than $2 \cdot 10^{-3}$.

A parametric analysis also has been performed to evaluate the effect of the operating and geometrical parameters on the average desorption rate. Figs. 15 to 17 show the effect of the inlet solution and hot water mass flow rates, inlet solution temperature, vapour

pressure and solution channel height and length. The rest of variables take the values presented in Table 1, except the solution pressure that has been increased up to 15.5 kPa in the direct diffusion mode. The effect of these variables is shown separately for the boiling and direct diffusion desorption modes. Comparison between both desorption modes reveals that, as expected and also shown in Fig. 14, boiling mode allows obtaining higher desorption rates.

As observed in Fig. 15, the increase of the channel length decreases the average desorption rate because mass transfer rate reduces as the solution concentration increases. Solution channel height also influences notably desorption. Thinner channels increase the ratio between mass transfer area and solution volume, increasing mass transfer. Solution channel height has a higher relative influence on the direct diffusion mode, while the channel length impacts more during the boiling regime. In both figures also it is observed that the influence of the channel length and channel height on the desorption ratio reduces as these quantities increase.

Results shown in Fig. 16 allow deriving optimum hot water mass flow rates for each condition of solution mass flow rate. For the maximum desorption ratios shown, optimum (minimum) hot water flow rates are about 1 and 1.3 g s⁻¹ for the direct diffusion and boiling mode respectively. The decrease observed in the optimum hot water flow rates as the desorption rate reduces is associated to a slight decrease/increase in the solution mass flow rate for the direct diffusion/boiling regime.

Finally, Fig. 17 shows the increment produced in the desorption rate with the increase of the solution inlet temperature and vapour pressure. A linear influence of both variables on the desorption rate is observed. Raises observed in the desorption rate remain invariable for constant increments in temperature or pressure, except for pressures below 4.5 kPa in the boiling mode. In this last case the influence of the solution inlet temperature is reduced.

4. Conclusions

In the present paper, a simulation of the heat and mass transfer processes taking place in a miniaturized desorber using membrane technology has been developed. For the studied configuration, a desorber of 5 cm length, mass fraction changes of about 0.9% and average desorption rates of 8.6·10⁻³ kg m⁻² s⁻¹ are obtained. Comparison between the

model predictions and the experimental results shows a reasonable agreement, being the mean absolute error of the model with respect to experimental data equal to 26.4%. This demonstrates the applicability of the model to predict the performance of miniaturized membrane-based desorbers.

The influence of operating and geometrical variables on the desorption rate has been evaluated using the model presented. These variables include the inlet solution temperature, vapour pressure, solution and hot water mass flow rates and solution channel height and length. Therefore, the model can be used to optimize geometrical and operating conditions of the desorber both in the direct diffusion and boiling modes.

The simulation shows the variation along the channel of membrane and solution mass transfer coefficients, thermal power transferred to solution, convection heat transfer coefficients in the hot water and solution, cooling power, ratio between the cooling power and the desorber volume and COP of the refrigeration cycle, which have not been presented before.

The region of the channels where the desorption rate increases very rapidly corresponds to the first centimetre, where the solution is heated. Beyond this point, the solution temperature remains almost constant, the solution increases its weight percentage of salt and the desorption potential decreases.

According to the results, in order to maximize the ratio between the cooling capacity of the chiller and the desorber volume, when designing the desorber it is not recommended to use very long channels, because the increase of the desorber size is higher than the increment obtained in the cooling power. In the studied configuration, the simulated desorber has a maximum value of the ratio between cooling capacity of the chiller and desorber volume of 2312 kW m^{-3} . This value is more than one order of magnitude higher than values found in small-size absorption chillers. For this reason, the configuration of membrane-based rectangular micro-desorber here proposed is specially recommended for low-capacity absorption chillers.

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Fig. 2. Differential element of the desorber.

Fig. 3. Flowchart for simulating the heat and mass transfer process along the desorption channel.

Fig. 4. Desorption rate as a function of the wall temperature for different vapour pressures. Comparison between model and experimental results of Isfahani et al. [11].

Fig. 5. Desorption rate as a function of the wall temperature for different solution mass flow rates. Comparison between model and experimental results of Isfahani et al. [11].

Fig. 6. Evolution of the LiBr mass fraction and pressure potential along the channel.

Fig. 7. Evolution of the solution, membrane and overall desorption coefficients along the channel.

Fig. 8. Evolution of the desorption rate along the channel.

Fig. 9. Evolution of the solution and hot water temperatures along the channel.

Fig. 10. Evolution of solution and hot water convection coefficients along the channel.

Fig. 11. Evolution of the thermal powers transferred by the hot water and stored in the solution along the channel.

Fig. 12. Evolution of the cooling power and the ratio between the cooling power and the desorber volume along the channel.

Fig. 13. Evolution of the COP along the channel.

Fig. 14. Evolution of the desorption coefficients along the channel in the case of change from direct diffusion to boiling mode.

Fig. 15. Average desorption rate as a function of the solution channel height and length. a) Direct diffusion. b) Boiling.

Fig. 16. Average desorption rate as a function of the solution and hot water mass flow rates. a) Direct diffusion. b) Boiling.

Fig. 17. Average desorption rate as a function of the inlet solution temperature and vapour pressure. a) Direct diffusion. b) Boiling.

Table 1. Data considered for the simulation, corresponding to the schemes of the desorber represented in Figs. 1 and 2.

Parameter	Value
Hot water inlet temperature, T_{hw} (°C)	90
Solution inlet temperature, T_s (°C)	70
Membrane pore diameter, d_p (μm)	1
Solution pressure, P_s (kPa)	5.5
Vapour pressure, P_v (kPa)	5.5
Porosity, ϵ	0.8
Hot water channel height, e_{hw} (mm)	0.15
Wall thickness, e_w (mm)	2.7
Solution channel height, e_s (mm)	0.15
Membrane thickness, e_m (μm)	60
Vapour channel height, e_v (mm)	5
Solution channel centre-to-centre distance, w_s (mm)	1.6
Hot water channel centre-to-centre distance, w_{hw} (mm)	1.6
Hot water channel width, l_{hw} (mm)	1.5
Solution channel width, l_s (mm)	1.5
Vapour channel width, l_v (mm)	20
Length of channels, L (mm)	50
Discretization length, dz (mm)	0.29
Total solution mass flow rate at the inlet, $\dot{m}_{s,T}$ (g s ⁻¹)	0.5
Total hot water mass flow rate, $\dot{m}_{hw,T}$ (g s ⁻¹)	0.5
Vapour mass flow rate at the inlet, $\dot{m}_v$ (g s ⁻¹)	0
LiBr mass fraction at the inlet, x	0.55