



Universidad  
Carlos III de Madrid



This is a postprint version of the following published document:

Venegas, M.; Vega, M.; García-Hernando, N.; Ruiz-Rivas, U. (2016). A simple model to predict the performance of a H<sub>2</sub>O–LiBr absorber operating with a microporous membrane. *Energy*, v. 96, pp. 383-396.  
DOI: 10.1016/j.energy.2015.12.059

© Elsevier 2016



This work is licensed under a Creative Commons Attribution-NonCommercial-NoDerivatives 4.0 International License.

# A simple model to predict the performance of a H<sub>2</sub>O LiBr absorber operating with a microporous membrane

M. Venegas <sup>a, b, c, \*</sup>, M. de Vega <sup>a, c</sup>, N. García-Hernando <sup>a, c</sup>, U. Ruiz-Rivas <sup>a, b</sup>

<sup>a</sup> ISE Research Group, Department of Thermal and Fluids Engineering, Universidad Carlos III de Madrid, Avda. Universidad 30, 28911 Leganés, Madrid, Spain

<sup>b</sup> GTADS Research Group, Department of Thermal and Fluids Engineering, Universidad Carlos III de Madrid, Avda. Universidad 30, 28911 Leganés, Madrid, Spain

<sup>c</sup> Associated Research Unit CSIC-Universidad Carlos III de Madrid, Spain

## Keywords:

Absorption refrigeration  
Membranes  
Absorber  
Water-lithium bromide  
Rectangular microchannels

## A B S T R A C T

A microporous membrane is used in combination with rectangular microchannels in the absorber of an absorption chiller with the aim of reducing the size of this cooling technology. The simulation of the heat and mass transfer between the solution and the vapour phase in a H<sub>2</sub>O LiBr absorber using porous fibres is considered. Heat and mass transfer processes are modelled by means of selected correlations and data gathered from the open literature. This new model is applied for the simulation of the absorber under typical operating conditions of absorption cooling chillers. Absorption rate, heat and mass transfer coefficients, solution concentration, temperatures of the working fluids and pressure potential along the absorption channels are calculated. For the case considered in this study, the absorber channels are of 5 cm length, offering a maximum ratio between cooling capacity of the chiller and absorber volume of 1090 kW/m<sup>3</sup>. This ratio is higher than twice the usual values found in falling film absorbers using conventional circular tubes. The mean absolute error between the model results and the experimental data gathered from the open literature is 8.5%, showing the capability of the model to predict the performance of membrane based absorbers.

## 1. Introduction

Absorption chillers and heat pumps have received growing attention over the last few decades since they allow the use of low grade heat to produce a cooling effect, with no (or minimal) mechanical power consumption. For example, absorption cooling machines can use waste heat or excess heat from solar collectors in the form of hot water for cooling purposes. In buildings, they could help to optimize the solar thermal energy in the summer period reducing the electricity consumption peak due to air conditioning. Nevertheless, their use is limited, compared to mechanical refrigeration systems, because they are generally larger in size, limiting to a great extent their use in low and medium power applications. For instance, a typical value for the volume to refrigeration power ratio in single effect absorption chillers is in the order of 0.04 m<sup>3</sup>/kW (without considering the volume occupied by the cooling

system) for refrigeration capacities between 10 and 30 kW, whereas mechanical compressor systems can have a ratio equal to 0.02 m<sup>3</sup>/kW for the same range of refrigeration capacities (García Hernando et al. [1]).

The absorber is one of the most performance limiting and volume demanding components of absorption systems. In the absorber, the concentrated liquid solution absorbs the refrigerant vapour that comes from the evaporator. The main challenge in designing and operating these devices is to maximize the mass transfer rate by getting as much interfacial area as possible. Several configurations for the liquid and vapour streams in the absorber have been proposed in order to increase the heat and mass transfer in the liquid solution: falling films, bubbles, sprays and droplets, liquid jets and sheets, etc. Nevertheless, all of them present relatively low heat and mass transfer coefficients and lead to large, heavy and rigid heat exchangers. Other authors proposed the use of Plate Heat Exchangers in the absorber or the desorber, in order to increase the cooling capacity to volume ratio in absorption systems (Venegas et al. [2], de Vega et al. [3]). At present, a promising new technology is under study, consisting in the use of membrane contactors in microchannel heat exchangers.

\* Corresponding author. ISE Research Group, Department of Thermal and Fluids Engineering, Universidad Carlos III de Madrid, Avda. Universidad 30, 28911 Leganés, Madrid, Spain. Tel.: +34 916248776; fax: +34 916249430.

E-mail address: [mvenegas@ing.uc3m.es](mailto:mvenegas@ing.uc3m.es) (M. Venegas).

Nomenclature			
$A$	area ( $m^2$ )	$T$	temperature ( $^{\circ}C$ )
$b$	molality (mole $kg_{water}^{-1}$ )	$u$	velocity ( $m s^{-1}$ )
$c_B$	Boltzmann constant ( $J K^{-1}$ )	$U$	global heat transfer coefficient ( $W m^{-2} K^{-1}$ )
$C_p$	specific heat ( $kJ kg^{-1} K^{-1}$ )	$x$	lithium bromide mass fraction ( $kg_{LiBr} kg_s^{-1}$ )
$dz$	discretization length (m)	$z$	length (m)
$D$	diffusion coefficient ( $m^2 s^{-1}$ )	<i>Greek symbols</i>	
$D_h$	hydraulic diameter (m)	$\alpha$	channel aspect ratio, $\alpha = l/e$ .
$d_p$	membrane pore diameter (m)	$\Delta T$	temperature difference ( $^{\circ}C$ )
$e$	height or thickness (m)	$\varepsilon$	porosity
$h$	convective heat transfer coefficient ( $W m^{-2} K^{-1}$ )	$\lambda$	mean free path (m)
$i$	specific enthalpy ( $kJ kg^{-1}$ )	$\mu$	viscosity (Pa s)
$J$	absorption rate ( $kg m^{-2} s^{-1}$ )	$\Xi$	Ackermann factor
$k$	thermal conductivity ( $W K^{-1} m^{-1}$ )	$\rho$	density ( $kg m^{-3}$ )
$K$	mass transfer coefficient ( $kg Pa^{-1} m^{-2} s^{-1}$ )	$\sigma$	molecular collision diameter ( $\text{\AA}$ )
$Kn$	Knudsen number, $Kn = \lambda/d_p$	$\tau$	tortuosity
$l$	width (m)	$\Phi$	heat transfer rate factor
$L$	total length of channels (m)	<i>Subscripts</i>	
$\dot{m}$	mass flow rate ( $kg s^{-1}$ )	$cw$	cooling water
$M$	molecular weight ( $kg mole^{-1}$ )	$l$	liquid
$N$	number of channels	$m$	membrane
$Nu$	Nusselt number, $Nu = hD_h/k$	$ov$	overall
$P$	pressure (Pa)	$s$	solution
$Pr$	Prandtl number, $Pr = \mu C_p/k$	$sat$	saturation
$q$	thermal power (W)	$th$	thermal
$R$	mass transfer resistance ( $kg^{-1} Pa m s$ )	$T$	total
$Re$	Reynolds number, $Re = uD_h\rho/\mu$	$v$	vapour
$R_u$	universal gases constant ( $J mole^{-1} K^{-1}$ )	$va$	vapour absorbed
$Sc$	Schmidt number, $Sc = \mu/\rho D$	$w$	wall
$Sh$	Sherwood number, $Sh = KD_h/D$		

Membrane based absorbers use a microporous polymeric membrane at the solution refrigerant vapour interface. In the membrane, many small diameter pores avoid mixing between vapour and solution, while allow the gas and solution to be in contact. Surface tension prevents the solution from entering the holes, while the gas diffuses to the solution surface through the pores. In the absorber, the gaseous fluid (typically ammonia or water) passes the membrane and is absorbed by the solution ( $NH_3-H_2O$  or  $H_2O-LiBr$  respectively) flowing inside constrained flow passages. The vapour pressure difference through the membrane is the driving force for mass transfer. If the partial pressure of the vapour inside the solution is less than the vapour pressure, it is absorbed at the interface between solution and vapour. Compared with conventional absorption devices, there are several advantages of using microporous fibre modules for vapour absorption. These include larger interfacial area per unit volume, independent control of vapour and liquid flow rates, easier scale up, modular design and compactness.

Concerning ammonia water systems, Chen et al. [4] theoretically modelled and simulated an absorber using hollow fibre membranes and compared the results to a plate heat exchanger absorber of falling film type. In their model, they considered the ammonia-water concentration across this microporous membrane as the driving potential for mass transfer. The comparisons of the heat and mass transfer coefficients of both devices revealed that the vast interfacial area of the membrane solution is the major reason for achieving higher absorption performance. Schaal et al. [5] experimentally investigated a  $NH_3-H_2O$  absorber using a hollow fibre hydrophobic microporous membrane in order to get basic design data. Ghiasi et al. [6] conducted experimental

research to investigate the absorption of ammonia vapour into the solution, with the main objective of optimizing the absorber within the whole refrigeration system to optimize the system performance.

In  $H_2O-LiBr$  applications, Isfahani and Moghaddam [7] tested an absorber using a superhydrophobic nanofibrous membrane with nominal pore size of 1 micron and 80% porosity. They obtained an absorption rate of about  $0.006 kg/m^2s$ , using channels of 100 micron thickness and a flow velocity of 5 mm/s. Isfahani et al. [8] presented a study on the efficacy of highly porous nanofibrous membranes for application in membrane based absorbers and desorbers. Permeability studies showed that membranes with a pore size greater than about 1 micron are valid for their application in absorbers. Ali and Schwerdt [9] experimentally and analytically investigated the characteristics and properties of commercially available microporous hydrophobic membranes. They evaluated some factors that influence the water vapour (refrigerant) mass transfer flux into a thin lithium bromide-water solution confined in narrow channels. Yu et al. [10] numerically investigated the performance of a membrane based absorber using the  $H_2O-LiBr$  solution. They showed that several folds enhancement in the absorption rate can be achieved with respect to conventional absorbers. When the film thickness was reduced from 150 to 50 microns, the absorption rate increased 3 fold. When the solution velocity was quadrupled, the average absorption rate increased 50%. Bigham et al. [11] showed that mass transport in the microfilm solution could be improved by the implementation of micro scale features on the flow channel surface. Recently a review of membrane contactors applied in absorption refrigeration systems has been done by Asfand and Bourouis [12].

According to the literature review, in order to optimize the heat and mass process to minimize the absorber volume, detailed studies about the role of the relevant parameters in the absorption process are still necessary. In the present investigation, the water lithium bromide solution has been selected. This working pair is adequate for evaporation temperatures higher than 0 °C, and it is the most common solution used for air conditioning applications, according to Henning [13]. Moreover, these systems can be fed with low temperature heat. The present study focuses therefore in the absorber modelling of the heat and mass transfer in the microporous membrane coupled with the cooling of the solution in a H<sub>2</sub>O–LiBr absorber.

## 2. Modelling of the absorption process

The absorber configuration used in the present study is shown in Figs. 1 and 2. It is a plate and frame membrane module with geometrical data and operating variables described in Table 1. The module contains a vapour channel, separated from the solution by an adjacent microporous membrane. The solution is confined in microchannels. The upper side of these channels is the hydrophobic membrane. A metal wall separates the solution channels from the cooling water channels. Water lithium bromide solution is used.

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

- The system operates under steady state conditions.
- The solution, cooling water and refrigerant vapour completely fill their respective channels: no incondensable gases are present and vapour absorption 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, cooling water and vapour are well mixed in each differential volume. Homogeneous temperature and concentration are considered in all points within each respective control volume.
- The heat and mass transfer are one dimensional.
- Pressure is constant along the channels. As a result, the solution equilibrium concentration is only a function of the solution temperature.

As a novelty regarding membrane based absorbers, the heat and mass transfer models consider that temperature and/or concentration are changing along both thermal entrance and fully

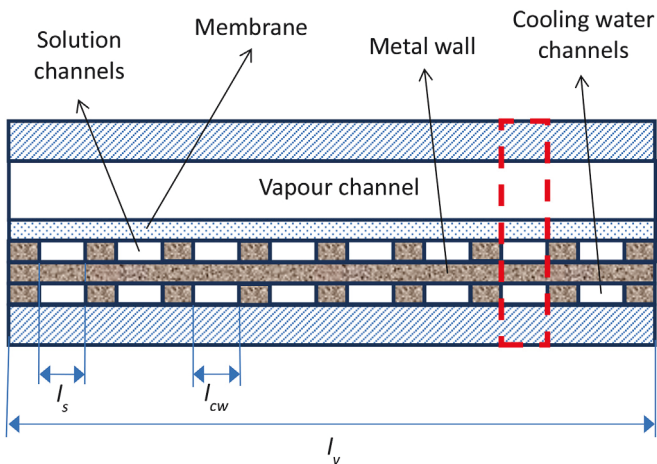


Fig. 1. Cross section of the plate and frame membrane-based absorber.

developed flow regions. Additionally, the influence of mass transfer on heat transfer, which has not been included before in similar investigations, is taken into account.

In the following Sections (2.1 and 2.2), the general equations defining the heat and mass transfer models are presented. In Section (2.3), the correlations and data gathered from the open literature, required to implement the heat and mass transfer models, are grouped together and discussed.

### 2.1. Mass transfer model

The vapour mass flux through the membrane is calculated taking into account the conditions in the bulk water vapour and bulk aqueous solution streams. For the differential element  $j$ , the mass flow rate is:

$$\dot{m}_{va}^j = J^j \cdot A \quad (1)$$

where  $A$  is the heat and mass transfer area:

$$A = l_s \cdot dz \quad (2)$$

and  $J$  is the absorption rate:

$$J^j = \frac{P_v - P_s^j}{R_{ov}^j} \quad (3)$$

$P_v$  and  $P_s$  are the bulk vapour pressure and the water vapour partial pressure corresponding to the bulk solution concentration ( $x$ ) and temperature ( $T_s$ ), according to Ali [14]. The overall mass transfer resistance between the bulk water vapour and bulk aqueous solution ( $R_{ov}$ ) includes the resistance to diffusion through the aqueous solution boundary layer ( $R_s$ ) and the resistance to diffusion of water vapour through the membrane active layer ( $R_m$ ). Both resistances act in series:

$$R_{ov}^j = R_s^j + R_m^j \quad (4)$$

Equations used to calculate  $R_s^j$  and  $R_m^j$  are described in Section 2.3.1.

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

$$\dot{m}_s^{j+1} = \dot{m}_s^j + \dot{m}_{va}^j \quad (5)$$

$$\dot{m}_v^{j+1} = \dot{m}_v^j - \dot{m}_{va}^j \quad (6)$$

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

$$x^{j+1} = x^j \frac{\dot{m}_s^j}{\dot{m}_s^{j+1}} \quad (7)$$

### 2.2. Heat transfer model

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

$$(\dot{m}_{va} i_w)^j = q_s^j + q_v^j + q_{cw}^j \quad (8)$$

Left term in Eq. (8) corresponds to the thermal power released during absorption of the vapour flow rate  $\dot{m}_{va}$  into the solution. Right terms are related to the heat transferred to the solution, vapour and cooling water respectively. These can be calculated, for parallel flow, as:

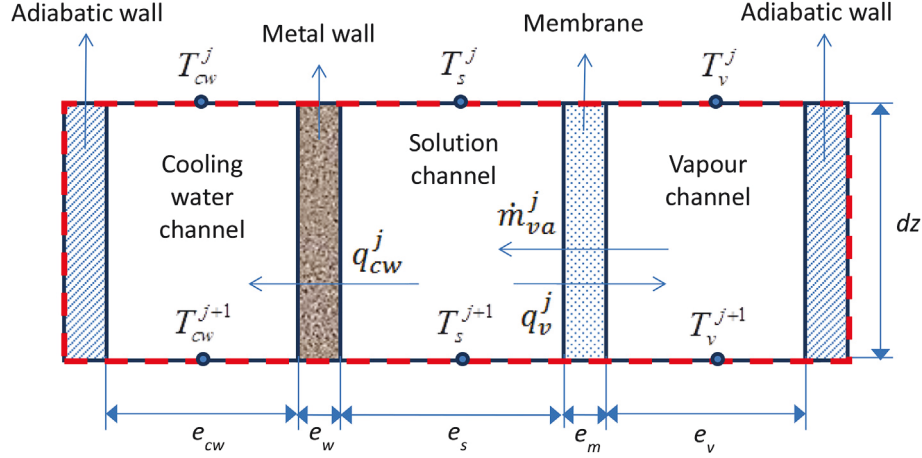


Fig. 2. Differential element of the absorber.

Table 1

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

Parameter	Value
Cooling water inlet temperature, $T_{cw}$ ( $^{\circ}\text{C}$ )	27
Solution inlet temperature, $T_s$ ( $^{\circ}\text{C}$ )	32
Vapour inlet temperature, $T_v$ ( $^{\circ}\text{C}$ )	7
Membrane pore diameter, $d_p$ ( $\mu\text{m}$ )	1
Vapour pressure, $P_v$ (kPa)	1
Porosity, $\varepsilon$	0.8
Cooling water channel height, $e_{cw}$ (mm)	0.15
Metal wall thickness, $e_w$ (mm)	2.7
Solution channel height, $e_s$ (mm)	0.15
Membrane thickness, $e_m$ ( $\mu\text{m}$ )	60
Vapour channel height, $e_v$ (mm)	5
Solution channel centre-to-centre distance, $w_s$ (mm)	1.6
Cooling water channel centre-to-centre distance, $w_{cw}$ (mm)	1.6
Cooling water channel width, $l_{cw}$ (mm)	1.5
Solution channel width, $l_s$ (mm)	1.5
Vapour channel width, $l_v$ (mm)	20.9
Number of solution channels, $N_s$	13
Number of cooling water channels, $N_{cw}$	13
Length of channels, $L$ (mm)	50
Discretization length, $dz$ (mm)	0.22
Total solution mass flow rate at the inlet, $m_{s,T}$ (g/s)	1
Total cooling water mass flow rate, $m_{cw,T}$ (g/s)	0.5
Vapour mass flow rate at the inlet, $m_v$ (g/s)	0.0032
LiBr mass fraction at the inlet, $x$	0.6

$$q_s^j = (\dot{m} \cdot i_s^{j+1}) - (\dot{m} \cdot i_s^j) \quad (9)$$

$$q_v^j = (\dot{m} \cdot i_v^{j+1}) - (\dot{m} \cdot i_v^j) \quad (10)$$

$$q_{cw}^j = \dot{m}_{cw} (i_{cw}^{j+1} - i_{cw}^j) = \dot{m}_{cw} C_{p,cw}^j (T_{cw}^{j+1} - T_{cw}^j) \quad (11)$$

Last term of Eq. (11) considers that cooling water is an incompressible liquid with constant specific heat.

The present model also considers that heat is transferred from the bulk solution channel (where absorption takes place) to both cooling water and vapour channels. In this case the following relations are also valid:

$$q_{cw}^j = U_{s-cw}^j A (T_s^j - T_{cw}^j) \quad (12)$$

$$q_v^j = U_{s-v}^j A (T_s^j - T_v^j) \quad (13)$$

The global heat transfer coefficients in Eqs. (12) and (13) are calculated as:

$$\frac{1}{U_{s-cw}^j} = \frac{1}{h_{cw}^j} + \frac{e_w}{k_w} + \frac{1}{h_{s,cw}^{j,*}} \quad (14)$$

$$\frac{1}{U_{s-v}^j} = \frac{1}{h_v^j} + \frac{e_m}{k_{m,ave}^j} + \frac{1}{h_{s,v}^{j,*}} \quad (15)$$

Equations and data used to calculate the local convection heat transfer coefficients and the thermal conductivities are described in Section 2.3.2.

The concentration and temperatures along the channels cannot be explicitly determined from Eqs. (1)–(15). 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<sup>TM</sup> (Klein [15]). Fig. 3 shows the flowchart used for implementing the heat and mass transfer models.

Main advantages of the present heat and mass transfer models include:

- Less time consuming than detailed numerical codes. Calculation time is less than 30 s.
- Simplicity of the model allows its implementation using any programming language. In this case EES<sup>TM</sup> has been selected, because of its high accurate thermodynamic and transport property database that can be used with the equation solving capability [15].
- No hypothesis about temperature distribution along the cooling water channel is required.
- Heat transfer to vapour channel is not neglected.

### 2.3. Selection of correlations and data

In order to implement the heat and mass transfer models developed in the previous sections, data and correlations available in the open literature were carefully revised and appropriated ones selected, taking into account their adequacy to the simulated conditions.

Thermodynamic properties of water lithium bromide solution have been calculated using EES<sup>TM</sup>, which uses correlations developed by Patek and Klomfar [16] for all properties, except for viscosity and thermal conductivity. In this case, EES uses correlations provided by DiGiulio et al. [17]. Properties of water are also

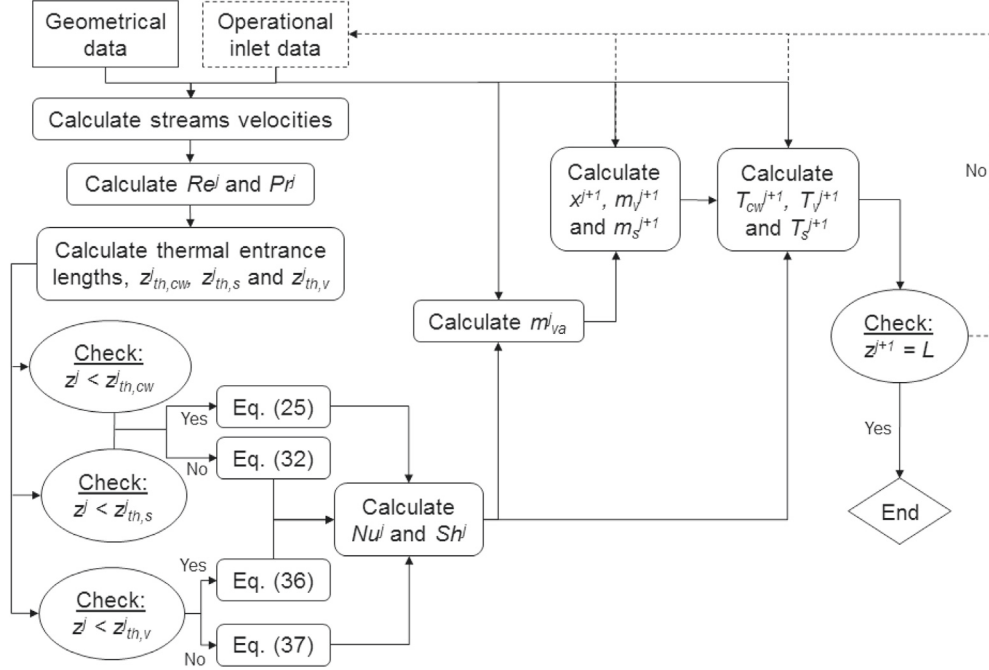


Fig. 3. Flowchart for simulating the heat and mass transfer process along the absorption channel.

calculated using EES: the correlation of Harr et al. [18] for the thermodynamic properties and transport ones using equations of the Electrical Research Association [19]. Thermal conductivity of the metal wall was assumed to be 10 W/mK. This is a typical value for corrosion resistant metals and alloys (Chawla and Gupta [20]).

### 2.3.1. Mass transfer

To calculate the mass flux through the membrane, the flow regime must first be determined. Knudsen number ( $Kn$ ) is used to evaluate the transport of water vapour across the membrane: when  $Kn$  is less than 0.01, collisions between gas molecules dominate and viscous flow occurs, resulting in rapid convective transport. When  $Kn$  is equal or larger than 10, collisions between molecules and pore walls are dominant, and the gas transport takes place via Knudsen flow (free molecular regime). Transitional regime exists if  $0.01 < Kn < 10$ . Knudsen number is defined as:

$$Kn = \lambda / d_p \quad (16)$$

where  $\lambda$  is the mean free path of the water vapour (distance between molecular collisions) and  $d_p$  is the membrane pore diameter. In this work, the mean free path is calculated using the equation given by Redhead et al. [21]:

$$\lambda = \frac{c_B T_m}{\sqrt{2\pi\sigma^2 P_m}} \quad (17)$$

where  $c_B$  is the Boltzmann constant ( $1.38 \cdot 10^{-23}$  J/K),  $T_m$  is the absolute temperature (K),  $\sigma$  is the molecular collision diameter (2.641 Å for water vapour) and  $P_m$  is the mean total pressure within the membrane pores (Pa).

For the current study, the calculated Knudsen number value changes very slightly around 13.6. Consequently, free molecular flow occurs through the membrane along the whole absorber.

According to the Dusty Gas model, described by Mason and Malinauskas [22] and applied by Yu et al. [10] to a similar problem, and taking into account that free molecular flow is present in the

membrane, the mass transfer coefficient through the membrane  $K_m$  can be estimated as:

$$K_m = \frac{M}{e_m} \frac{D_e^K}{R_u T_m} \quad 1/R_m \quad (18)$$

where the diffusion term can be evaluated as:

$$D_e^K = \frac{\epsilon d_p}{3\tau} \left( \frac{8R_u T_m}{\pi M} \right)^{0.5} \quad (19)$$

In Eq. (18),  $e_m$  is the membrane thickness. In Eqs. (18) and (19),  $M$  is the molecular weight of water and  $R_u$  is the universal gases constant. In Eq. (19),  $\epsilon$  and  $\tau$  are the porosity and tortuosity of the membrane, respectively. Tortuosity of the membrane is calculated as a function of the membrane porosity, according to Iversen et al. [23]:

$$\tau = \frac{(2 - \epsilon)^2}{\epsilon} \quad (20)$$

Resistance to mass transfer inside the bulk solution  $R_s$  can be calculated according to Ali and Schwerdt [9], as:

$$R_s = \frac{P_{sat}}{\rho_{water} K_s} \quad (21)$$

where  $P_{sat}$  is the saturated water pressure corresponding to the bulk solution temperature,  $\rho_{water}$  is the liquid water density and  $K_s$  is the mass transfer coefficient between the solution–vapour interface and the bulk aqueous solution. A suitable correlation for mass transfer in microchannels has not been found in the open literature. For this reason, the mass transfer coefficient of the solution is calculated using mass and heat transfer analogy, using correlations of Lee and Garimella [24] for the thermal entrance region and Shah and London [25] for the fully developed flow. These correlations are described in the next section.

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 corrected by temperature using the equation described by Mittermaier et al. [26]:

$$D = D(25^\circ\text{C}) \frac{T + 273.15}{298.15} \frac{\mu(25^\circ\text{C})}{\mu(T)} \quad (22)$$

where the diffusion coefficient at 25 °C is calculated as:

$$D(25^\circ\text{C}) = \left( 1.3528 + 0.19881b - 0.036382b^2 + 0.0020299b^3 - 0.000039375b^4 \right) 10^{-9} \quad (23)$$

Parameter  $b$  in Eq. (23) is the molality, obtained from:

$$b = \frac{x}{M_{\text{LiBr}}(1-x)} \quad (24)$$

$M_{\text{LiBr}}$  is the molecular weight of the lithium bromide.

### 2.3.2. Heat transfer

The convection heat transfer coefficients, in Eqs. (14) and (15), along the solution, cooling water and vapour channels, have been calculated taking into account two regions: thermal entrance region and fully developed flow. Correlations to be used were selected also considering the dimensions of the channels.

In the case of the solution and cooling water channels, with  $D_{h,s} = D_{h,cw} = 272.7 \mu\text{m}$ , equations 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. This correlation has the advantage, with respect to others available in the open literature, of allowing the estimation of the heat transfer coefficient as a function of the position along the channel. Also, it covers a sufficiently broad range of aspect ratios and hydraulic diameters.

$$Nu_{th} = \frac{1}{C_1(z^*)^{C_2} + C_3} + C_4, \quad \text{for } 1 \leq \alpha \leq 10, \quad z^* < z_{th}^* \quad (25)$$

The channel aspect ratio is  $\alpha = l/e$ . The dimensionless axial distance  $z^*$  is:

$$z^* = z / (RePrD_h) \quad (26)$$

The dimensionless length of the thermal entrance region is:

$$z_{th}^* = 1.275 \cdot 10^{-6} \alpha^6 + 4.709 \cdot 10^{-5} \alpha^5 - 6.902 \cdot 10^{-4} \alpha^4 + 5.014 \cdot 10^{-3} \alpha^3 - 1.769 \cdot 10^{-2} \alpha^2 + 1.845 \cdot 10^{-2} \alpha + 5.691 \cdot 10^{-2} \quad (27)$$

and:

$$C_1 = 3.122 \cdot 10^{-3} \alpha^3 + 2.435 \cdot 10^{-2} \alpha^2 + 2.143 \cdot 10^{-1} \alpha + 7.325 \quad (28)$$

$$C_2 = 0.6412 \quad (29)$$

$$C_3 = 1.589 \cdot 10^{-4} \alpha^2 - 2.603 \cdot 10^{-3} \alpha + 0.02444 \quad (30)$$

$$C_4 = 7.148 - \frac{13.28}{\alpha} + \frac{15.15}{\alpha^2} - \frac{5.936}{\alpha^3} \quad (31)$$

Convection heat transfer coefficients along the microchannels for the fully developed flow were obtained using the correlation developed by Shah and London [25]. This correlation is valid for rectangular channels, which is the present case:

$$Nu = 8.235 \left( 1 + \frac{2.0421}{\alpha} + \frac{3.0853}{\alpha^2} + \frac{2.4765}{\alpha^3} + \frac{1.0578}{\alpha^4} + \frac{0.1861}{\alpha^5} \right) \quad (32)$$

In the case of the solution channel, the effect of mass transfer on heat transfer was taken into account using a modified heat transfer coefficient. Coefficients obtained using Eqs. (25) and (32) were corrected, multiplying them by the Ackermann factor (Taylor and Krishna [27]):

$$\Xi = \frac{\Phi}{e^{\Phi} - 1} \quad (33)$$

where the heat transfer rate factor  $\Phi$  is defined as:

$$\Phi = J \cdot Cp_v \cdot e_s / k_v \quad (34)$$

The modified convection coefficient, for heat transfer from the solution to the cooling water channel, is incremented because mass transfer is taking place in the same direction of heat transfer. In the case of heat transfer to the vapour channel the coefficient is reduced because the processes occur in opposite directions.

The vapour channel is a conventional rectangular channel with  $D_{h,v} = 8.1 \text{ mm}$ . In this case, the dimensionless thermal entry length is calculated using the following correlation, derived in this research from data tabulated by Shah and London [25] for the case of simultaneously developing flow:

$$z_{th,v}^* = \frac{0.205}{\alpha_v^2} + \frac{0.5282}{\alpha_v} + 0.0169, \quad \text{with } R^2 = 1 \quad (35)$$

Data tabulated also by Shah and London [25], for the case of constant heat flux, were used to derive the following correlation for  $Nu$  in this developing region. It is valid for the aspect ratio of the vapour channel used in the present study,  $\alpha_v = 4$ .

$$Nu_{th,v} = 3 \cdot 10^{-5} (z_v^*)^2 + 0.0307 z_v^* + 5.2901, \quad \text{with } R^2 = 0.999 \quad (36)$$

Finally, for the thermally developed flow, a correlation was also derived from data tabulated by Shah and London [25] for the case of constant heat flux:

$$Nu_v = \frac{0.037}{\alpha_v^2} + \frac{0.7639}{\alpha_v} + 2.8036, \quad \text{for } 1 \leq \alpha_v \leq 10, \quad \text{with } R^2 = 0.998 \quad (37)$$

The average thermal conductivity of the membrane,  $k_{m,ave}$  in Eq. (15), is calculated using the equation given by Martínez and Rodríguez Maroto [28]:

$$k_{m,ave} = \epsilon k_v + (1 - \epsilon) k_m \quad (38)$$

In Eq. (38),  $k_v$  is the thermal conductivity of the vapour inside the membrane pores, while  $k_m$  is the thermal conductivity of the membrane solid material. A polytetrafluoroethylene (PTFE) membrane has been selected, with thermal conductivity equal to 0.22 W/mK, the same used by Ali [14].

## 2.4. Model validation

Validation of the model has been performed comparing the absorption rate predicted with the experimental data reported by Isfahani and Moghaddam [7]. In the experimental case, the solution and cooling water channels measure 1 and 4 mm in width respectively, while two different heights for the solution channel, 0.1 and 0.16 mm, were used. The cooling water channel height was 0.4 mm. An important issue is that the correlations used in the present model remain valid in all the cases.

Figs. 4–7 show a comparison between the absorption rate predicted by the model, using the data of Isfahani and Moghaddam [7], and their experimental results. Fig. 4 shows the variation of the absorption rate as a function of the vapour pressure and the solution channels height. A similar trend is obtained in both cases. The almost parallel lines show the good prediction of the model concerning the vapour pressure increase influence: the absorption rate increases because the vapour pressure potential rises. Also, a better absorber performance is obtained in both cases for the less deep channel. This outcome coincides with simulation results presented in the next section about the influence of the solution channel aspect ratio, i.e. higher aspect ratios contribute to higher absorption ratios.

The influence of the cooling water inlet temperature on the absorption rate is represented in Fig. 5 for the two solution channel heights of the experiments. Again, the model predicts a tendency similar to the experiments in both cases. The increase in the cooling water inlet temperature reduces the driving force for heat transfer between the solution and the cooling water, leading to an increase of the solution temperature. Consequently, the solution water vapour pressure increases, which reduces the vapour pressure potential for mass transfer. As observed in Figs. 4 and 5, the difference between the simulation results and the experiments increases when the solution channel height decreases. One reason for this higher difference can be attributed to the tolerance in the channels manufacturing, 10  $\mu\text{m}$  in both cases, representing up to 10% of the channel height in the case of the 100  $\mu\text{m}$  channel.

The combined influence of the solution mass flow rate and the vapour pressure on the absorption ratio is shown in Fig. 6. Again, small differences exist between the experiments and the model results and similar trends are observed in all the cases. As observed,

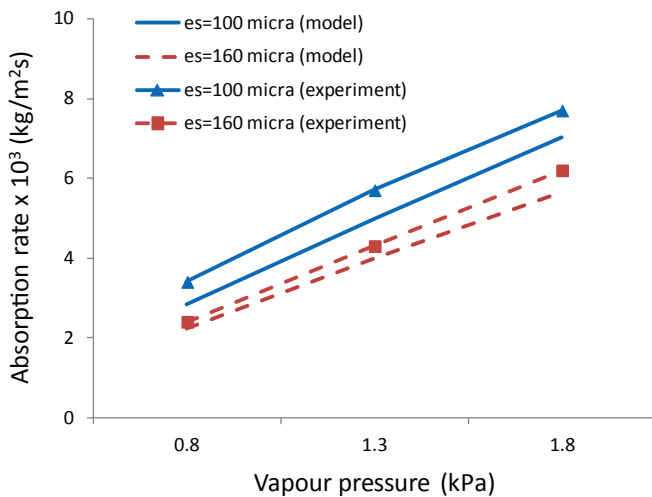


Fig. 4. Absorption rate as a function of the vapour pressure and the solution channel height ( $e_s$ ). Comparison between model and experimental results of Isfahani and Moghaddam [7].

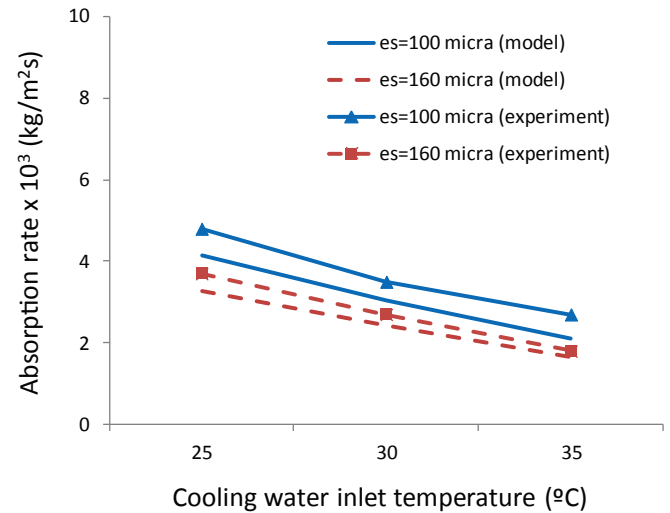


Fig. 5. Absorption rate as a function of the cooling water inlet temperature and the solution channel height ( $e_s$ ). Comparison between model and experimental results of Isfahani and Moghaddam [7].

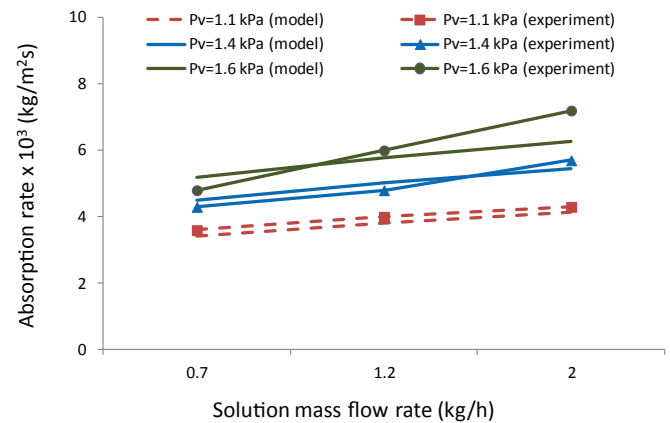


Fig. 6. Absorption rate as a function of the solution mass flow rate and the vapour pressure ( $P_v$ ). Comparison between model and experimental results of Isfahani and Moghaddam [7].

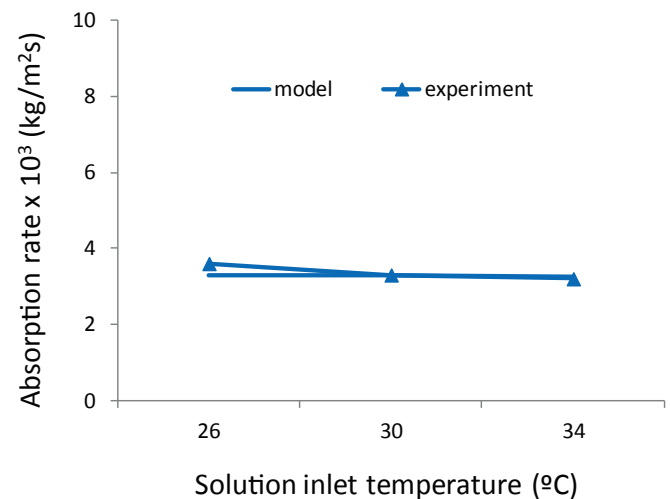


Fig. 7. Absorption rate as a function of the solution inlet temperature. Comparison between model and experimental results of Isfahani and Moghaddam [7].



the increase of the solution mass flow rate tends to increase the mass transfer. As discussed earlier in relation to Fig. 4, a higher vapour pressure contributes to increase the absorption rate.

Fig. 7 shows the effect of the solution inlet temperature on the absorption rate. Temperature increase approaches the solution to its equilibrium state, so the driving force for mass transfer decreases. This effect translates into a slight decrease of the absorption rate, as observed in the figure. Both model and experimental data show a similar tendency, and small differences exist between them.

The mean absolute error of the model predictions respect to all experimental data represented in Figs. 4–7 is 8.5%. This low difference demonstrates the value of the model to perform a good prediction of the miniaturized membrane based absorber performance. Also, it allows evaluating the influence of individual parameters on the absorption rate.

### 3. Results and discussion

In Fig. 8 the variation of the absorption rate along the channel is represented simultaneously with the solution concentration change. The decrease in LiBr mass fraction describes the water vapour absorption process. At the channel entrance, when higher pressure potential and overall mass transfer coefficient are available (Fig. 9), the absorption rate is higher. This value tends to decrease as the solution absorbs the vapour along the channel. Absorption rates of  $3.2 \cdot 10^{-3} \text{ kg/m}^2\text{s}$  on average are obtained in the channel.

As observed in Fig. 10, major resistance to mass transfer moves from membrane to solution side. In the first 2 cm, absorption is controlled by the membrane resistance. From this point, solution mass transfer limits the absorption process. These results have not been found in the literature of microporous membranes applied to absorption chillers, to our knowledge. This reveals the importance of decreasing the mass transfer resistance of both, membrane and solution, to improve the absorption rate. Membrane resistance is controlled mainly by the maximum pore diameter allowable to avoid aqueous solution penetration into the membrane pores, as discussed by Ali and Schwerdt [9]. One way to reduce the solution mass transfer resistance could be to decrease the solution thickness, which is corroborated by the heat and mass transfer analogy used in the present model. According to this analogy, solution mass transfer resistance decreases if the aspect ratio of the solution channels rises.

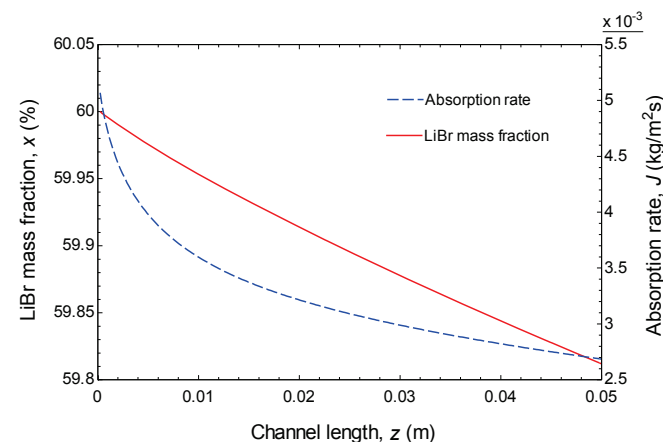


Fig. 8. Evolution of the LiBr mass fraction and absorption rate along the absorption channel.

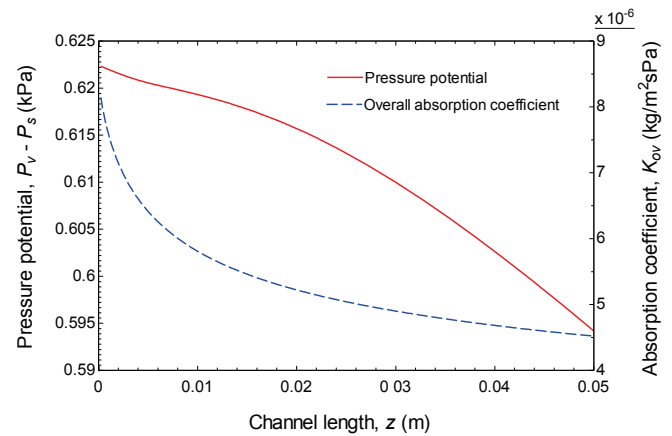


Fig. 9. Evolution of the pressure potential and the overall absorption coefficient along the absorption channel.

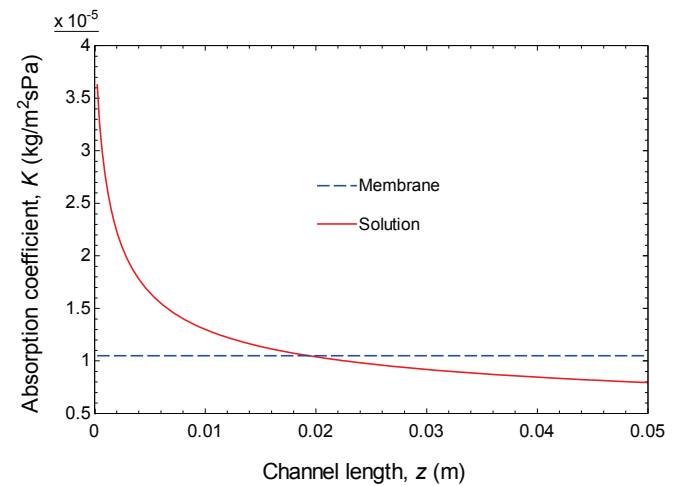


Fig. 10. Evolution of the absorption coefficients along the absorption channel.

Variation of the solution and cooling water temperatures along the absorption channel is shown in Fig. 11. Temperature of the solution is almost constant initially, due to heat transfer to cooling water, until 2 cm length approximately. From this point, the release of the absorption heat tends to increase both the solution and cooling water temperatures. The first 2 cm of constant solution temperature observed in the figure directly relates to the maximum observed in the absorption parameters represented in Figs. 8–10. It implies that, the lower the solution temperature is, the higher the potential for mass transfer will be.

Fig. 12 represents the evolution of the thermal power stored in the solution and transferred to the cooling water and vapour along the absorption channel. In the first 2 cm of the channel (corresponding to the solution temperature variation commented in relation to Fig. 11) most of the heat is transferred to the cooling water. After this point, heat is also stored in the solution and, associated to it, is the drop off observed in the absorption rate in Fig. 8. Along the whole absorber, heat is effectively transferred to the cooling water and this heat transfer is always higher than the heat stored in the solution. The contribution of the power transferred to the vapour is almost two orders of magnitude smaller.

Very high convection heat transfer coefficients are predicted in the present study for the solution and cooling water flows, as shown in Fig. 13. In these cases the thermal entrance

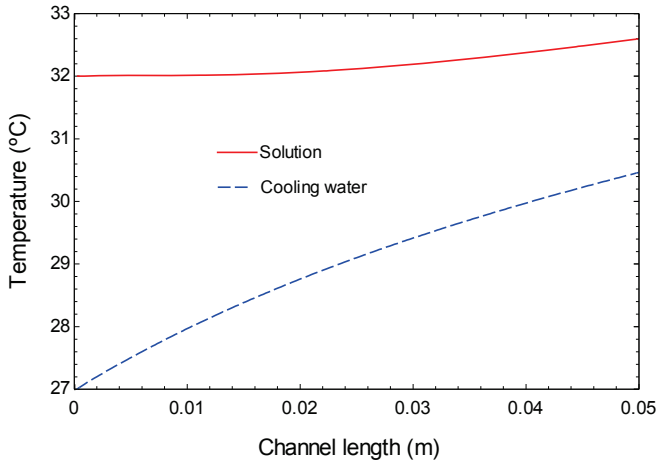


Fig. 11. Evolution of the solution and cooling water temperatures along the absorption channel.

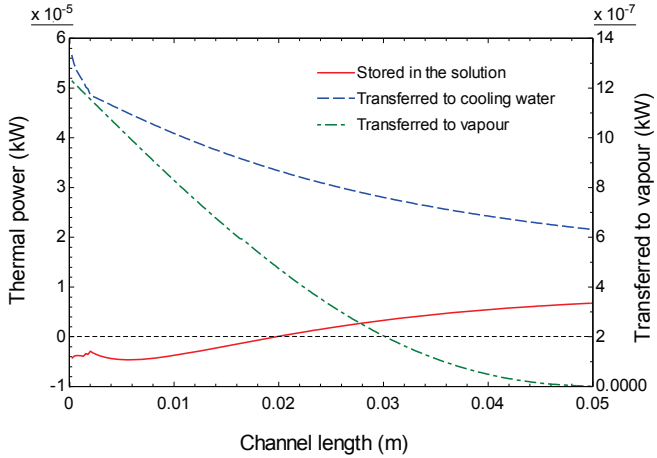


Fig. 12. Evolution of the thermal power transferred to the cooling water and vapour and stored in the solution along the absorption channel.

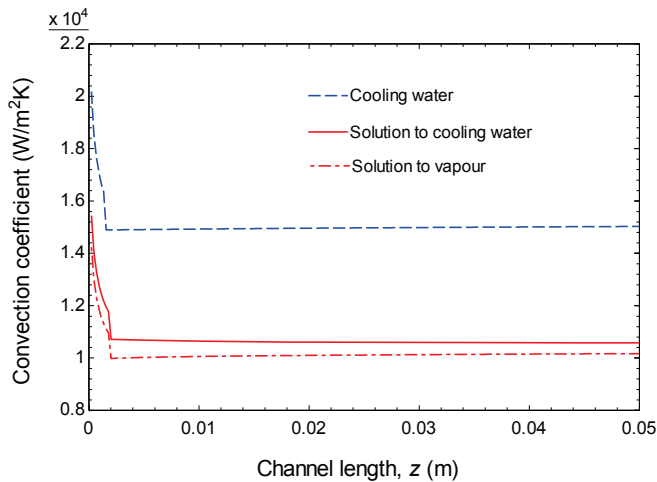


Fig. 13. Evolution of convection coefficients along the absorption channel.

region is about 2 mm in both cases. The strong change observed in the figure for the heat transfer coefficients corresponds to this point. Additionally, in the fully developed flow region, as the aspect ratio of the channels is constant, the slight increase observed in the convection coefficient of cooling water flow is determined by the change in the thermal conductivity of the fluid. Regarding the solution flow, the modified heat transfer coefficients (taking into account the effect of mass transfer) are shown. As commented before, convection coefficients for heat transfer to cooling water are higher than those to vapour because absorption takes place in the same direction of heat transfer. In the case of the vapour flow, very low convection coefficients are predicted, between 11.4 and 12 W/m<sup>2</sup>K. Along the whole vapour channel the flow is still thermally developing.

Laminar flow is expected to occur in all the cases. In the conditions evaluated, Reynolds numbers range between 54 and 60, 14–16 and 0.4–27 for the cooling water, solution and vapour, respectively. Velocity of the cooling water and solution are practically constant, at 17.2 and 20.1 cm/s respectively. Vapour velocity reduces from 4 m/s to 7 cm/s.

The cooling power of an absorption chiller equipped with the absorber modelled in the present work is represented in Fig. 14 as a function of the channel length. Also, the ratio between this cooling power and the absorber volume is shown. As observed, the ratio suddenly increases with the channel length, reaching its maximum value at 3 mm length. From this point, the ratio slowly decreases. Hence, in order to optimize the absorber size, when designing the absorber it is not recommended to use very long channels, because the increase of the absorber size is 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 absorber volume is used to optimize the size of membrane based absorbers.

The maximum value of the ratio between cooling power and absorber volume is 1090 kW/m<sup>3</sup>. If the module proposed in Fig. 1 is constructed forming a parallel system, in such a way that the vapour and cooling water channels are shared by the adjacent modules, this ratio can be increased almost twice. This ratio is higher than values found in optimized horizontal falling film absorbers using the same solution, as modelled by Jeong and Garimella [29]. In that study, the ratio reaches maximum values of

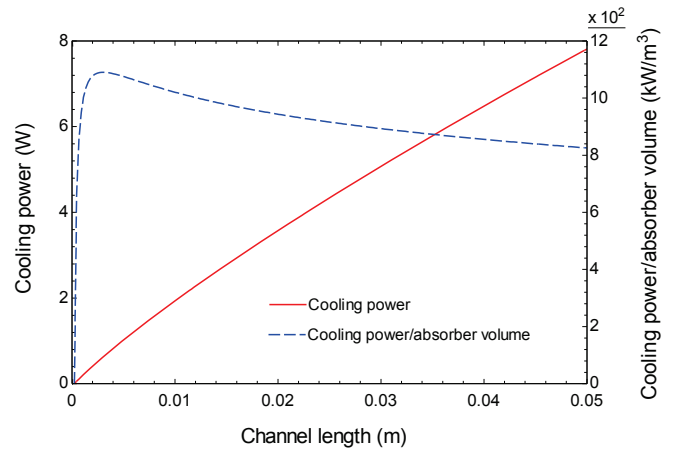


Fig. 14. Evolution of the cooling power and the ratio between the cooling power and the absorber volume along the absorption channel.

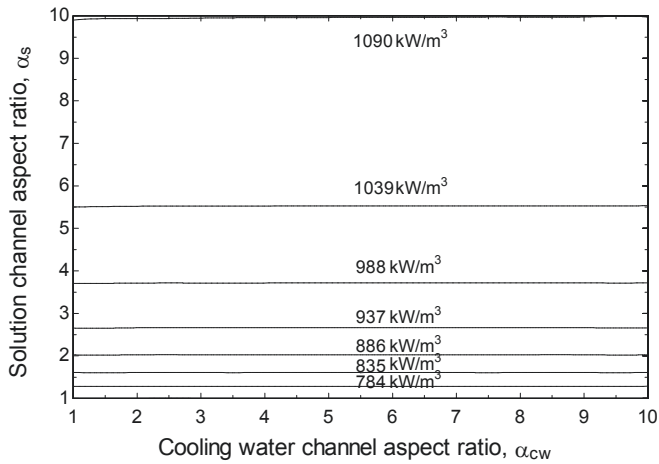


Fig. 15. Ratio between the cooling power and the absorber volume for different aspect ratios of the solution and cooling water channels.

938.3 kW/m<sup>3</sup> for tubes of 3.175 mm diameter, 655.6 kW/m<sup>3</sup> for tubes of 6.35 mm diameter, and 450.7 kW/m<sup>3</sup> for conventional tubes of 15.88 mm diameter. Additionally, wetting of small diameter tubes can be a problem and the cooling capacity can be reduced by about 30% if the one half of the total tube surface is wet (Jeong and Garimella [29]).

The model also allows optimizing the geometry of the absorber channels. Therefore, the influence of the solution and cooling water channels aspect ratios on the performance of the absorber is evaluated in Fig. 15. As it is observed, the aspect ratio of the cooling water channel has a negligible influence on the absorber size. However, the cooling power of the chiller equipped with this absorber notably increases when the solution channels aspect ratio augments. The maximum ratio between this cooling power and the absorber volume is obtained for the maximum value evaluated,  $\alpha_s = 10$ .

#### 4. Conclusions

In the present paper, a simulation of the heat and mass transfer processes taking place in a miniaturized absorber using membrane technology has been developed. The model has been validated comparing the results with experimental data available in the literature. Novel results have been presented, showing the variation along the channel of: membrane and solution mass transfer coefficients, thermal powers transferred to cooling water and vapour and stored in the solution, convection heat transfer coefficients in the cooling water and solution, cooling power and the ratio between the cooling power and the absorber volume. Also, as a novelty, the influence of the aspect ratios of the cooling water and solution channels on the ratio between the cooling power and the absorber volume is given. The following conclusions have been derived:

- The first 2 cm of the absorption channels are the most active for mass transfer. Here the solution is sufficiently cool and is able to efficiently absorb the water vapour. As the solution heats, its absorption potential decreases.
- Mass fraction changes of about 0.2% and average absorption rates of  $3.2 \cdot 10^{-3}$  kg/m<sup>2</sup>s are obtained for the conditions simulated in the present study using an absorber of 5 cm length. Mass transfer coefficients of about  $5.2 \cdot 10^{-6}$  kg/m<sup>2</sup>sPa are obtained.

- The simulated absorber has a maximum value of the ratio between cooling capacity of the chiller and absorber volume of 1090 kW/m<sup>3</sup>. This value is more than twice the one obtained using falling film absorbers of conventional diameter tubes. Additionally, dimensions of the absorber equipped with membranes can be reduced at about one half of the value here obtained using a parallel combination of the proposed module.
- In order to maximize the ratio between the cooling capacity of the chiller and the absorber volume, when designing the absorber it is not recommended to use very long channels, because the increase of the absorber size is higher than the increment obtained in the cooling power.
- The aspect ratio of the solution channel has a high influence on the absorber performance. Keeping constant the channel width, when the channel height decreases the absorption rate increases and the absorber size reduces. This demonstrates the importance of reducing the solution film thickness.
- Comparison between the model predictions and the experimental results shows a very good agreement. The mean absolute error of the model with respect to experimental data is 8.5%. This demonstrates the applicability of the model to predict the performance of miniaturized membrane based absorbers.
- The influence of geometrical and operational parameters on the absorption rate can be evaluated using the model presented. These parameters include the solution channel height, solution and cooling water inlet temperatures, vapour pressure and solution mass flow rate. Therefore, the model can be used to optimize the geometry and operating conditions of the absorber.

#### Acknowledgements

The financial support of this study by the Ministerio de Economía y Competitividad of Spain through the research grant ENE2013 43131 R is greatly appreciated.

#### References

- [1] García-Hernando N, Almendros-Ibanez JA, Ruiz G, de Vega M. On the pressure drop in plate heat exchangers used as desorbers in absorption chillers. *Energy Convers Manag* 2011;52:1520–5.
- [2] Venegas M, Arzoz D, Rodríguez P, Izquierdo M. Heat and mass transfer in LiNO<sub>3</sub>-NH<sub>3</sub> spray absorption system. *Int Commun Heat Mass Transf* 2003;30:805–15.
- [3] de Vega M, Almendros-Ibanez JA, Ruiz G. Performance of a LiBr-water absorption chiller operating with plate heat exchangers. *Energy Convers Manag* 2006;47:3393–407.
- [4] Chen J, Chang H, Chen S-R. Simulation study of a hybrid absorber-heat exchanger using hollow fiber membrane module for the ammonia-water absorption cycle. *Int J Refrig* 2006;29:1043–52.
- [5] Schaal F, Weimer T, Hasse H. Membrane contactors for absorption refrigeration. Seoul, Korea: International Sorption Heat Pump Conference; 2008.
- [6] Ghiasi C, Marti-Calatayud MC, Weimer T, Ziegler F. Ammonia-water absorption heat pumps with membrane absorber. 2009. IIR Conference: Ammonia Refrigeration Technology, Ohrid.
- [7] Isfahani RN, Moghaddam S. Absorption characteristics of lithium bromide (LiBr) solution constrained by superhydrophobic nanofibrous structures. *Int J Heat Mass Transf* 2013;63:82–90.
- [8] Isfahani RN, Sampath K, Moghaddam S. Nanofibrous membrane-based absorption refrigeration system. *Int J Refrig* 2013;36:2297–307.
- [9] Ali AH, Schwerdt P. Characteristics of the membrane utilized in a compact absorber for lithium bromide-water absorption chillers. *Int J Refrig* 2009;32:1886–96.
- [10] Yu D, Chung J, Moghaddam S. Parametric study of water vapour absorption into a constrained thin film of lithium bromide solution. *Int J Heat Mass Transf* 2012;55:5687–95.
- [11] Bigham S, Yu D, Chugh D, Moghaddam S. Moving beyond the limits of mass transport in liquid absorbent microfilms through the implementation of surface-induced vortices. *Energy* 2014;65:621–30.
- [12] Asfand F, Bourouis M. A review of membrane contactors applied in absorption refrigeration systems. *Renew Sustain Energy Rev* 2015;45:173–91.
- [13] Henning HM. Solar assisted air conditioning of buildings – an overview. *Appl Therm Eng* 2007;27:1734–49.

- [14] Ali AHH. Design of a compact absorber with a hydrophobic membrane contactor at the liquid vapor interface for lithium bromide water absorption chillers. *Appl Energy* 2010;87:1112–21.
- [15] Klein SA. Engineering equation solver. Academic Professional, 1992–2014, V9,698-3D. 2014.
- [16] Patek J, Klomfar J. A computationally effective formulation of the thermodynamic properties of LiBr-H<sub>2</sub>O from 273 to 500 K over full composition range. *Int J Refrig* 2006;29:566–78.
- [17] DiGuilio RM, Lee RJ, Jeter SM, Teja AS. Properties of lithium bromide-water solutions at high temperatures and concentrations I thermal conductivity. *ASHRAE Trans* 1990;RP-527:702–8.
- [18] Harr L, Gallagher JS, Kell GS. NBS/NRC steam tables. Hemisphere Publishing Co; 1984.
- [19] Electrical Research Association Steam Tables. Thermodynamic properties of water and steam; viscosity of water and steam, thermal conductivity of water and steam. London: Edward Arnold Publishers; 1967.
- [20] Chawla SL, Gupta RK. Materials selection for corrosion control. *ASM Int* 1993: 272.
- [21] Redhead PA, Hobson JP, Kornelsen EV. The physical basis of ultrahigh vacuum. Chapman and Hall; 1968.
- [22] Mason EA, Malinauskas AP. Gas transport in porous media: the dusty-gas model. Elsevier; 1983.
- [23] Iversen SB, Bhatia VK, Dam-Johansen K, Jonsson G. Characterization of microporous membranes for use in membrane contactors. *J Membr Sci* 1997;130:205–17.
- [24] Lee P-S, Garimella SV. Thermally developing flow and heat transfer in rectangular microchannels of different aspect ratios. *Int J Heat Mass Transf* 2006;49:3060–7.
- [25] Shah RK, London AL. Laminar flow forced convection in ducts, in: a source book for compact heat exchanger analytical data. *Advances in heat transfer*. New York: Academic Press; 1978. Suppl. 1.
- [26] Mittermaier M, Schulze P, Ziegler F. A numerical model for combined heat and mass transfer in a laminar liquid falling film with simplified hydrodynamics. *Int J Heat Mass Transf* 2014;70:990–1002.
- [27] Taylor R, Krishna R. Multicomponent mass transfer. New York: John Wiley & Sons; 1993.
- [28] Martínez L, Rodríguez-Maroto JM. Characterization of membrane distillation modules and analysis of mass flux enhancement by channel spacers. *J Membr Sci* 2006;274:123–37.
- [29] Jeong S, Garimella S. Optimal design of compact horizontal tube LiBr/water absorbers. *HVAC&R Res* 2005;11:27–44.