Numerical study of the effectiveness of a vertical falling plate film absorber for an absorption chiller

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Abstract:

This study investigates the impact of design and operating conditions on the performance of a falling film grooved plate absorbers integrated in a Lithium Bromide water absorption refrigeration machine. The performance of the component is described using two parameters: the mass efficiency and the absorbed vapour mass flow rate per unit plate width. The study implements a heat and mass transfer model validated with specific experimental data. The absorber effectiveness proposed by previous studies was generalized so that all absorbers' operating conditions could be considered. A parametric study considering variation of solution Reynolds number is conducted. Based on these simulations, global and local transfers are discussed.

Highlights

- Calculation of the mass effectiveness of absorbers
- Heat and mass transfer modeling of grooved plates falling film absorber
- Description of the operating conditions using the ratio of the maximum transferable flows between fluids
- Efficiency Chart describing the Impact of flow rates on the mass effectiveness of absorbers

Keywords

Falling film, flat plate exchanger, absorber, Lithium Bromide, mass effectiveness

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Nomenclature		Subscripts and superscripts	
A	cross-section of the heat transfer fluid, m ²	abs	absorbed
e	thickness, m	$adiab$	adiabatic absorption case
h	enthalpy, J.kg ⁻¹	eq	equilibrium conditions
$\tilde{h}$	partial enthalpy, J.kg ⁻¹	f	solution film
L	length, m	g	groove
w	width, m	htf	heat transfer fluid
m	mass, kg	h	high
$\dot{m}$	mass flow rate, kg.s ⁻¹	H_2O	liquid water
$\dot{M}$	mass flow rate per unit of absorber width, kg.s ⁻¹ .m ⁻¹	i	input
		$int-sol$	interface between the solution film and the vapor
Q	thermal energy, J	iso	isothermal absorption case
R	ratio of the maximum enthalpy variation flow	l	low
T	temperature, °C	max	maximum
$\bar{T}$	mean temperature, °C	min	minimum
x	mass concentration of LiBr in the solution, kg _{LiBr} .kg _{sol} ⁻¹	o	output
$\bar{x}$	mean mass concentration of LiBr in the solution, kg _{LiBr} .kg _{sol} ⁻¹		
		sat	saturated
Greek symbols		sol	solution
α	convective heat transfer coefficient, W.m ⁻² .K ⁻¹	v	water vapor
β	convective mass transfer coefficient, m.s ⁻¹	w	absorber wall
Δz	length of the discretization mesh, m	$w-sol$	interface between the absorber wall and the solution film
ε	effectiveness	$w-htf$	interface between the absorber wall and the heat transfer fluid
λ	thermal conductivity, W.m ⁻¹ .K ⁻¹		
ρ	density, kg.m ⁻³		

1. Introduction

Absorption refrigeration machines are an alternative to the mechanical compression machines used to produce a refrigeration effect (Kim and Ferreira, 2008). These machines operate with a binary working fluid where the most volatile is the refrigerant. The most common solutions used in these systems are the aqueous bromide lithium solution ($\text{H}_2\text{O}/\text{LiBr}$) (water is then the only volatile fluid) for cooling applications higher than 0°C and the ammonia water solution ($\text{NH}_3/\text{H}_2\text{O}$) (NH_3 refrigerant is more volatile than the H_2O sorbent fluid) for temperature applications below 0°C (Altamirano et al 2019).

The $\text{H}_2\text{O}/\text{LiBr}$ absorption machines operate with low internal pressure, especially in the evaporator and the absorber (typically a few kPa). Due to this characteristic, the pressure drop in these elements significantly influences the performance of these machines, which is why they are usually designed with falling film heat and mass exchangers. Among these technologies, the horizontal tubular falling film exchangers are the most commonly used and studied (Babadi and Farhanieh, 2005; Hoffmann et al., 1996; Jeong and Garimella, 2002; Kim et al., 2003; Kyung et al., 2007 ; Altamirano et al 2020)). However, some of these studies investigated vertical tube absorbers (Miller and Keyhani, 2001; Matsuda et al., 1994; Bourouis et al., 2005; Medrano et al., 2002; Yoon et al., 1999). Few studies have been conducted on plate-type falling film absorbers. However, using this type of exchangers is an attractive idea for small-capacity absorption machines. They can be more compact and cheaper than tubular heat exchangers (Flamensbeck et al., 1998; Arzoz et al., 2005; Fujita, 1993; Killion and Garimella, 2003; Kyung et al., 2007; Pérez Reisler, 2007; Takamatsu et al., 2003; Kim and Infante-Ferreira, 2008).

The absorber is known to be the least efficient of all the subcomponents of absorption machines, mainly because of the relatively low rates of absorption of the refrigerant to the absorbent (Raisul Islam et al., 2006). Therefore, the absorber is generally the most voluminous and most costly exchanger of absorption machines. This key component is usually designed using an empirical approach or a strategy derived from classical heat exchangers such as the ΔT_{lm} method, which is not allowed to take into account the coupled heat and mass transfer taking place in the solution falling film (Matsuda et al., 1994). The effectiveness of exchangers is a performance indicator usually used to characterize the performance of heat exchangers. The development of design tools based on the efficiency of the absorber could help design this exchanger considering the different transfer phenomena that take place in that component.

The absorber effectiveness has already been defined or used in previous studies (Raisul Islam et al., 2006; Kim, 1995; Rogdakis and Papaefthimiou, 2002; Andberg and Vliet, 1983; Kim, 2010; George, 1989). In these studies, the maximum transferable flux is calculated considering that the flow that limits the transfer is always the solution flow. However, under certain operating conditions the limiting flow can be the heat transfer fluid. The goals of the present study were first to propose a definition and a calculation method to determine the effectiveness so as to describe the absorber performance whatever the operating conditions and second to numerically investigate the evolution of the effectiveness of a plate falling film absorber considering several solution and heat transfer fluid flow.

Numerous models have been developed to describe the heat and mass transfer in falling film absorbers and a critical review was proposed by Killion and Garimella (Killion and Garimella, 2001). Among the different approaches, the finite volume technique was chosen for this study. The model takes into account the liquid and vapor phase mass transfer resistances and uses empirical correlations to predict the heat and mass transfer coefficients.

In the first part of this paper, the 1D model developed to describe plate falling film absorber operation is described. In the second part, the absorber effectiveness is defined and the calculation method of the

maximum transferable flux is introduced. Finally, the simulation conditions are described and the main results of the parametric study are presented and discussed.

2. Model of falling film absorbers

2.1. Model description

The model of the absorber was developed to describe the operation of a vertical slotted falling film absorber (Figure 1). The absorber is composed of two metal plates along which the LiBr falling film develops. It is surrounded by saturated water vapor. The absorber plate is cooled by a heat transfer fluid flowing between the plates. The LiBr solution and the heat transfer fluid can flow in the same or in opposite directions: in this paper these configurations are called co-current and counter-current, respectively. The wettability of the plates is ensured by 4-mm-wide grooves. This technique prevents the formation of capillary waves. Heat and mass transfers therefore only develop due to the diffusion process.

Fig. 1: Schematic representation of the plate falling film absorber

The model has already been described in detail in a previous paper (Huaylla et al., 2014). Therefore, only the main hypotheses and equations are presented here. A 1D approach is used for the solution falling film, the solution vapor interface, the metal plate, and the heat transfer fluid (Figure 1). The model involves the application energy balances on these elements as well as mass and species balances for the solution falling film under stationary conditions.

To establish the model the following assumptions are considered:

- The variations of the kinetic and potential energies of fluids are considered negligible.
- The longitudinal heat conduction inside the absorber plate is considered negligible compared to the transversal heat conduction.
- The heat and mass exchanged between the phases are calculated with the average temperatures and concentration of each phase $\left(\bar{T} = (T^i + T^o)/2\right)$ and $\left(\bar{x} = (x^i + x^o)/2\right)$.

- The solution at the vapor/solution interface is considered to be in equilibrium with vapor $(T_i = T_{eq}(P_v; x_i))$.
- Lithium bromide salt has a very low vapor pressure and in this analysis is assumed to be nonvolatile (Yuan and Herold, 2005). Moreover, it is assumed that there is no incondensable gas in the absorber vessel. The gas in the system is therefore assumed to be pure water vapor.
- The solution flow is laminar. The development boundary layers at the vapor solution and wall solution interfaces are considered through the variation of the mass and heat convection transfer coefficient along the absorber.
- The pressure is considered to be homogenous.
- The heat transfer between the falling film and the surrounding environment due to radiation and convection are assumed to be negligible.

The mass, species, and energy balance of the solution film is expressed as follows:

$$\dot{m}_{sol}^i - \dot{m}_{sol}^o + \dot{m}_{abs} = 0 \quad (1)$$

$$\dot{m}_{sol}^i x_{sol}^i - \dot{m}_{sol}^o x_{sol}^o = 0 \quad (2)$$

$$\dot{m}_{sol}^i h_{sol}^i - \dot{m}_{sol}^o h_{sol}^o + \dot{m}_{abs} h_v^{sat} + \alpha_{w-sol} \Delta z L_{w-sol} (T_{w-sol} - \bar{T}_{sol}) = 0 \quad (3)$$

The mass and energy balance at the interface between the vapor and the solution film is expressed as follows:

$$\dot{m}_{abs} - \beta_{int-sol} \rho_{H_2O} \Delta z L_{int-sol} (\bar{x}_{sol} - x_{int-sol}) = 0 \quad (4)$$

$$\dot{m}_{abs} (h_v^{sat} - \tilde{h}_{H_2O,sol}) - \alpha_{int-sol} \Delta z L_{int-sol} (T_{int-sol} - \bar{T}_{sol}) = 0 \quad (5)$$

The equilibrium condition at the interface between the vapor and the solution film reads:

$$x_{int-sol} = x_{eq}(P_v; T_{int-sol}) \quad (6)$$

The energy balance of the heat transfer fluid reads:

$$\dot{m}_{htf} (h_{htf}^i - h_{htf}^o) + \alpha_{w-htf} \Delta z L_{w-htf} (T_{w-htf} - \bar{T}_{htf}) = 0 \quad (7)$$

The energy balance at the interface between the solution film and the absorber wall is written:

$$\alpha_{w-sol} \Delta z L_{w-sol} (\bar{T}_{sol} - T_{w-sol}) + \frac{\lambda_w}{e_w} \Delta z L_{w-sol} (\bar{T}_{htf-sol} - T_{w-sol}) = 0 \quad (8)$$

The energy balance at the interface between the heat transfer fluid and the absorber wall is written as follows:

$$\alpha_{w-htf} \Delta z l_{w-htf} (\bar{T}_{htf} - T_{w-htf}) + \frac{\lambda_w}{e_w} \Delta z L_{w-sol} (T_{w-sol} - \bar{T}_{htf-sol}) = 0 \quad (9)$$

The convective heat and mass transfer coefficients at the solution film interfaces (α_{w-sol} , $\alpha_{int-sol}$ and $\beta_{int-sol}$) are calculated using the correlations obtained by Brauner for a laminar LiBr solution film flowing over a constant temperature plate surface (Brauner, 1991; Grossman, 1983). The calculation of these coefficients requires the determination of the solution film thickness. This is calculated later using the equation developed by Carey (Carey, 2008) for a laminar film considering a parabolic velocity profile. The convective heat transfer coefficient between the heat transfer fluid and the absorber plate (α_{w-htf}) is calculated using the Dittus Boelter equation for an internal flow when the flow is turbulent (Incropera, 2011). When the heat transfer flow is laminar, that heat transfer coefficient (α_{w-htf}) is calculated by considering the establishment of the flow with the correlations described in (Incropera, 2011; Lee and Garimella, 2006). No transitional flow for the heat transfer fluid is considered in this model: the flow evolves from laminar directly to turbulent (and reversely) at a fixed Reynolds number.

2.2. Model validation

The 1D heat and mass transfer model of water vapour absorption in a laminar lithium bromide solution falling film was compared to experimental results in adiabatic and cooled absorption conditions (Michel et al 2017). Two grooved flat plate absorber (220 and 420 mm long respectively) have been characterized. The design of the grooves is presented in part 4. A wide range of operating conditions have been investigated by varying the inlet temperature and inlet mass fraction of the solution, as well as mass flow rate and the inlet temperature of the heat transfer fluid. The summary of the results is presented in Figure 2

The numerical model slightly overestimates the absorbed mass. Except for a few rare points the difference between the modeling and the experiment is less than 20%. For solution Reynolds number higher than 50, the model overestimated the absorbed mass by a mean 8.7% for the 220-mm plate and 3.6% for the 420-mm plate. For Reynolds numbers lower than 50, higher deviations between experimental and numerical results are observed (Michel et al 2017). Indeed, in this case, the model overestimates the absorbed mass. These overestimations may be attributed low wetting impacts. For low flow rates, dry regions develop on the absorber plates that do not participate in the absorption phenomenon (Huaylla et al (2019)). This wetting phenomenon has already been observed in other experiments (Mortazavi et al. (2015), B. Fumey et al. (2015), Huaylla et al (2019)) and is not considered by the model.

Fig. 2: Comparison of the experimental and simulated results

Although the model slightly overestimates the absorbed steam flow rate, it faithfully reproduces the impact of operating conditions and plate length effects. This model having been validated, it is used in the rest of the study to map the evolution of the absorber's performance according to its geometrical characteristics and operating conditions. These maps represent interesting tools for machine design.

3. Definition of absorber effectiveness

As heat exchangers, the effectiveness of absorbers (ε) can be defined by the ratio between the effective transfer flux and the maximum transferable flux. Absorbers are both heat and mass exchangers, which is why two effectiveness levels can be defined: one to characterize the heat exchange and one to characterize the mass exchange (Raisul Islam et al., 2006).

Thermal effectiveness (ε_{th}) is defined as the ratio between the heat rate collected by the heat transfer fluid ($\dot{Q}_{htf}$) and the maximum heat rate that can be collected by this fluid ($\dot{Q}_{htf}^{max}$).

$$\varepsilon_{th} = \frac{\dot{Q}_{htf}}{\dot{Q}_{htf}^{max}} \quad (10)$$

Mass effectiveness (ε_m) is defined as the ratio between the mass flow rate of vapor absorbed by the solution film ($\dot{m}_{abs}$) and the maximum mass flow rate of vapor that can be absorbed ($\dot{m}_{abs}^{max}$).

$$\varepsilon_m = \frac{\dot{m}_{abs}}{\dot{m}_{abs}^{max}} \quad (11)$$

When the solution enters and exits the absorber at equilibrium with the vapour water, the absorbed mass flowrate and the heat rate collected by the heat transfer fluid are proportional and related with by the sorption heat. The thermal and mass effectiveness are then similar. The situation is different when the solution at the inlet and outlet of the absorber is not at equilibrium with the vapour water. Take the case of a solution entering the absorber in an overheated state. This solution must be cooled before it can absorb vapour water. The exchanged heat rate with the heat transfer fluid cools the solution along the

plate. Conversely, superheated solution at the inlet is subject to desorption before it reaching its equilibrium state and being able to absorb vapor. The mass effectiveness is then negative at the inlet. It increases downstream throughout the plate and tends towards thermal efficiency.. In such an operating configuration, thermal efficiency is then higher than mass efficiency. If the solution enters the absorber in a sub-cooled state, the reverse will be true. Absorption systems can be used as frigorific machines as well as heat pumps; the useful effects are the heat exchanged at the evaporator (directly proportional to the vapor absorbed by the absorber) and the heat rejected at the absorber (collected by the heat transfer fluid). It therefore seems advantageous to use thermal effectiveness (ε_{th}) to characterize the performance of an absorber used in a heat pump and mass effectiveness (ε_m) for an absorber implanted in a frigorific machine.

The maximum transferable flux in an absorber is achieved in an infinitely long counter-flow absorber. The approach to determine the maximum transferable heat flux ($\dot{Q}_{hff}^{max}$) and the maximum mass absorption flux ($\dot{m}_{abs}^{max}$) are similar; therefore, only the second one is developed in this paper. In previous studies (Raisul Islam et al., 2006; Kim, 1995; Rogdakis and Papaefthimiou, 2002; Andberg and Vliet, 1983; Patnaik, 1993; Grossman, 1995), the maximum absorbed flow rate is calculated considering an ideal device where the solution is cooled to the inlet temperature of the heat transfer fluid. The outlet of the solution is thus in equilibrium with the lowest temperature source of the absorber, ($T^{min} = T_{hff}^i$) so the variation of the LiBr concentration in the solution is maximum. This definition assumes that the flow of the heat transfer fluid is always sufficient to impose the solution's outlet temperature. However, under certain operating conditions (low heat transfer fluid flow rate compared to the solution flow rate and/or high inlet solution concentration and/or high vapor pressure) the heat transfer fluid cannot cool the solution at its inlet temperature. In these configurations, for an infinitely long counter-flow absorber the outlet temperature of the heat transfer fluid is imposed by the solution.

The operating modes can be determined using the maximum transferable heat flux of the two fluids. In fact, the fluid that has the highest maximum transferable heat flux is the one that imposed the outlet temperature of the other one. The solution's maximum transferable heat flux is obtained when its outlet reaches the lowest temperature in the absorber (T^{min}), i.e., when it reaches the inlet temperature of the heat transfer fluid as in an isothermal absorption ($T_{sol}^o = T^{min} = T_{hff}^i$) (Figure 3a). The maximum transferable heat flux of the heat transfer fluid is obtained when its outlet reaches the absorber's highest possible temperature (T^{max}), i.e., when it reaches (Figure 3b):

- the outlet temperature of the solution considering an adiabatic absorption ($T^{max} = T_{sol}^{o, adiab} = T_{hff}^o$) if the inlet solution equilibrium pressure is lower than the vapor pressure ($P_{sol, eq}^i(T_{sol}^i; x_{sol}^i) < P_v$);
- the inlet temperature of the solution ($T^{max} = T_{sol}^i = T_{hff}^o$) if the inlet solution equilibrium pressure is equal to or higher than the vapor pressure ($P_{sol, eq}^i(T_{sol}^i; x_{sol}^i) \geq P_v$).

Figure 3 shows the temperature trend. As the heat flux is not constant in the wall, the temperatures do not evolve linearly with the length. However, for sake of simplicity linear evolutions are plotted in that figures.

Fig. 3: Schematic representation of temperature profiles of the heat transfer fluid and the solution for an infinitely long counter-flow absorber if the solution is the limiting flow (a) and if the heat transfer fluid is the limiting flow (b)

The outlet state (temperature and concentration) of the fluids can be calculated by solving the equation system composed of the mass, species, and energy balances of the absorber (eq. 12, eq. 13, and eq. 14).

$$\dot{m}_{sol}^i - \dot{m}_{sol}^o + \dot{m}_{abs} = 0 \quad (12)$$

$$\dot{m}_{sol}^i x_{sol}^i - \dot{m}_{sol}^o x_{sol\ eq}^o(T_{sol}^o; P_v) = 0 \quad (13)$$

$$\dot{m}_{sol}^i h_{sol}^i - \dot{m}_{sol}^o h_{sol\ eq}^o(T_{sol}^o; P_v) + \dot{m}_{abs} h_v^{sat} + \dot{Q}_{htf} = 0 \quad (14)$$

The resolution of the equation system with $T_{sol}^o = T_{htf}^i$ makes it possible to calculate the heat flux transferred to the heat transfer fluid in the isothermal case $(\dot{Q}_{htf}^{iso})$, whereas the resolution of the equation system with $\dot{Q}_{htf} = 0$ allows one to determine the outlet temperature of the solution in the adiabatic case $(T_{sol}^{o\,adiab})$.

Based on these limiting cases, the transition between the two operating modes can be determined by the ratio between the maximum transferable heat flux of the heat transfer fluid $(\dot{Q}_{htf}^{max})$ and of the solution $(\dot{Q}_{sol}^{max})$.

$$R = \frac{\dot{Q}_{htf}^{max}}{\dot{Q}_{sol}^{max}} = \frac{\dot{m}_{htf} [h_{htf}^o(T_{htf}^{max}) - h_{htf}^i]}{\dot{m}_{sol}^i h_{sol}^i - \dot{m}_{sol}^{o\,iso} h_{sol}^{o\,iso}(T_{htf}^i; x_{sol}^{o\,iso}) + \dot{m}_{abs}^{iso} h_v^{sat}} \quad (15)$$

where

$$T_{htf}^{max} = \max[T_{sol}^i; T_{sol}^{o\,adiab}] \quad (16)$$

When the transition between the two operating modes has been determined, the maximum absorbed flow rate $(\dot{m}_{abs}^{max})$ can be calculated for any operating condition of the absorber by solving the equation system composed of eqs. 12, 13, and 17.

$$\dot{m}_{sol}^i h_{sol}^i - \dot{m}_{sol}^o h_{sol}^o(T_{sol}^o; P_v) + \dot{m}_{abs} h_v^{sat} + \dot{m}_{htf} [h_{htf}^i - h_{htf}^o(T_{htf}^o)] = 0 \quad (17)$$

If $R < 1$, the outlet temperature of the heat transfer fluid is imposed by the solution film and is equal to the maximum temperature of the absorber $(T_{htf}^o = T_{sol}^{max})$. If $R \geq 1$ the outlet temperature of the solution is imposed as equal to the inlet temperature of the heat transfer fluid $(T_{sol}^o = T_{htf}^i)$.

From the previous equation systems, the maximum absorbed flow rate can be calculated for definite inlet conditions. To be independent of the inlet concentration and temperature conditions, a dimensionless maximum absorbed flow rate (ψ) is proposed:

$$\psi = \frac{\dot{m}_{abs} - \dot{m}_{abs}^{adiab}}{\dot{m}_{abs}^{iso} - \dot{m}_{abs}^{adiab}} \quad (18)$$

Fig. 4: Example of the dimensionless maximum absorbed flow rate (ψ) versus the ratio of the maximum enthalpy variation flow (R)

Figure 4 shows an example of how the dimensionless maximum absorbed flow rate (ψ) evolves as a function of the ratio of the maximum enthalpy variation flow (R). At R lower than 10^{-2} the maximum absorbed flow is close to that obtained considering an adiabatic absorption. This means that in this condition the flow rate of the heat transfer fluid is too low compared to the flow rate of the solution to have a significant impact on absorber behavior. With an R between 10^{-2} and 1, the heat transfer fluid significantly cools the solution so the maximum absorbed flow rate increases to reach its highest possible value when R equals 1. When R exceeds 1, the outlet temperature of the solution is equal to the inlet temperature of the heat transfer fluid, i.e., the maximum absorbed flow stays at its highest possible value.

4. Simulation conditions and results

4.1. Simulation conditions

From the model described in the following section a simulation tool was developed to numerically study the behavior and the effectiveness of vertical grooved falling film absorbers.

The simulations presented in this study were performed considering operating parameters for the absorber close to one implanted in a single-effect absorption machine used for solar air conditioning (Figure 5).

To determine the standard operating parameters of this machine, the following operating conditions and assumptions are considered:

- The source temperatures are set as mentioned in Figure 5.
- The temperature differences between the inlet and outlet of the heat transfer fluids are equal to 5 K.
- The temperature peaks between the working fluids (solution or water) and the heat transfer fluids are equal to 5 K.
- The solution heat exchanger effectiveness is equal to 0.8.

- The pressure drops in the components and connecting pipes are not considered.
- The solution and the vapor are assumed to be in equilibrium in the generator at 80 °C and the condenser pressure.
- The water at the condenser outlet is considered a saturated liquid at 40 °C.
- The vapor at the evaporator outlet is assumed to be in equilibrium with the liquid at 7 °C.

The thermodynamic properties of the solution and the liquid water (for the heat transfer fluid) considered for these simulations are derived from the free-energy correlation presented in (Yuan and Herold, 2005). The transport properties (dynamic viscosity, thermal conductivity, diffusivity) of the solution and the water are dependent on the temperature and the LiBr concentration according to the correlation developed in (Herold, 1997; Gierow and Jernqvist, 1993). The liquid density is calculated by correlation gave in (Hellmann and Grossman, 1996). As for the water vapor, the transport properties are determined with the correlation described in (Florides, 2003).

According to these assumptions, the machine and specifically the absorber operating conditions were determined and are summarized in Figure 5 and Table 1.

Fig. 5: Main machine operating conditions

Inlet temperature of the heat transfer fluid (T_{htf}^i)	30 °C
Inlet temperature of the solution (T_{sol}^i)	48 °C
Inlet concentration of the solution (x_{sol}^i)	58 %
Absorber vapor pressure (P_v)	1 kPa

Table 1: Operating conditions of the falling film absorber

The absorber considered for these simulations is a stainless steel vertical plate with a grooved failing surface (Figure 6). This element is cooled by water flowing inside rectangular internal channels. The characteristics of this absorber are summarized in Table 2.

Fig. 6: Schematic cross-sectional view of the absorber

Configuration	Co-flow
Absorber length (L),	0,5 m
Thickness of the stainless steel wall (e_w)	3.10^{-3} m
Conductivity of the stainless steel wall (λ_w)	$16 \text{ W.K}^{-1}.\text{m}^{-1}$
Width of the slotted failing surface grooves (w_g)	4.10^{-3} m
Width of the heat transfer fluid channels (w_{htf})	4.10^{-3} m
Thickness of the heat transfer fluid channels (e_{htf})	2.10^{-3} m

Table 2: Main design characteristics of the absorber

To study the impacts of the solution Reynold number on the heat and mass transfer and on the effectiveness, a parametric study was conducted considering different solution flow rates $25 \leq Re_{sol} \leq 200$ with a nominal case at 25.

To be independent of the absorber width, the absorbed mass flow rate and the heat flux transfer at the heat transfer fluid are expressed per unit of width of the slotted failing surface grooves defined by eq. 20 and eq. 21, respectively.

$$\dot{M}_{abs} = \frac{\dot{m}_{abs}}{N_g w_g} \quad (20)$$

$$\dot{q}_{htf} = \frac{\dot{Q}_{htf}}{N_g w_g} \quad (21)$$

The transitional Reynolds number from laminar to turbulent flow (and inversely) for the heat transfer fluid is usually included between 2300 and 4000. It is set to $Re_{htf} = 4000$ in the present study to take an unfavorable position.

In the following sections, the behavior of the absorber for the nominal case and the impacts of the evolution of each parameter are analyzed and discussed.

4.2. Behavior of the absorber for the nominal operating and design conditions

The numerical model provides the temperature, solution mass flow rate, and LiBr mass fraction profile of the fluids along the absorber. It is also permeated to determine the local absorbed mass flow rate ($\dot{M}_{abs}$) and the heat transfer to the heat transfer fluid ($\dot{q}_{htf}$).

Figure 7a shows the changes in temperature and the LiBr mass fraction and Figure 7b shows the profiles of the local absorbed mass flow rate ($\dot{M}_{abs}$) and the heat transfer to the heat transfer fluid along the absorber in the nominal design and operating conditions (Table 2).

Overall the temperature and the LiBr mass fraction of the solution decreases, whereas the temperature of the heat transfer fluid increases along the absorber (Figure 7a).

At the absorber inlet, with the temperature and concentration condition used for this study, the equilibrium pressure of the solution was higher than the absorber pressure ($P_{sol}^{eq}(T_{sol}^i; x_{sol}^i) > P_v$). To reach the equilibrium condition, the solution desorbs water along the first few millimeters of the absorber ($\dot{M}_{abs} < 0$) (Figure 7b). These inlet conditions are characterized by a solution temperature higher than the interface temperature ($T_{sol}^i > T_{int}^i$) and a solution LiBr mass fraction lower than the interface mass fraction ($x_{sol}^i < x_{int}^i$) (Figure 7). Close to the absorber inlet the temperature of the solution ($\bar{T}_{sol}$) decreases rapidly because of the endothermic phenomenon of desorption and the cooling due to the flow

of the heat transfer fluid in co-flow with the solution ($\dot{q}_{htf}$). Indeed, close to the absorber inlet the cooling of the absorber ($\dot{q}_{htf}$) is high due to the high convective heat transfer coefficient between the heat transfer fluid and the absorber plate (entrance length), and the large temperature difference between the solution and the heat transfer fluid. This desorption generated a slight increase in the LiBr solution mass fraction ($\bar{x}_{sol}$). As result of the decrease in the solution temperature (and the slight increase of the solution LiBr mass fraction) the equilibrium pressure of the solution becomes lower than the vapor pressure and the solution absorbs water vapor ($\dot{M}_{abs} > 0$). Due to substantial cooling and the high LiBr mass fraction, the absorbed mass flow rate increases to reach a maximum close to $L=3$ cm. Then the lowering of the cooling (decrease in the temperature difference between the heat transfer fluid and the solution and lowering of the convective heat transfer coefficient between the heat transfer fluid and the absorber plate) and the rise in the LiBr mass fraction along the absorber lower the absorption potential and thus decrease the absorbed mass flow rate. During the absorption process, the interface temperature is higher than the solution temperature ($T_{int} > \bar{T}_{sol}$) because of the exothermic phenomenon of absorption that takes place at the interface and the cooling rate due to the flow of the heat transfer fluid. Throughout the absorption process, the interface LiBr mass fraction is lower than the solution mass fraction because of the absorption that takes place at the interface and because of the water mass diffusion inside the falling film.

Fig. 7: Profiles of temperatures and LiBr mass fractions (a) and heat transfer to the heat transfer fluid and the absorbed mass flow rate (b) along the absorber for the nominal case.

4.3. Influence of the solution flow rates

Fig. 8: Absorbed mass flow rate (a) and mass effectiveness versus (b) the R ratio for different solution inlet Reynolds numbers in co-flow configuration

Figure 8a shows the absorbed flow rate versus the R ratio for different inlet Reynolds numbers of the solution for a co-flow absorber. In this study, the inlet Reynolds number of the solution is set by adjusting only the inlet mass flow rate of the solution (the solution inlet properties are set to be constant).

For all the solution Reynolds numbers the absorbed mass increases as the R ratio rises because of the increase in cooling power, reducing the global heat transfer resistance. When the transfer fluid flow rate tends to be null (the R ratio also tends to be null), it works like an adiabatic absorber. Once the transfer fluid flow rate is very high (the R ratio tends to infinity), it works like an isothermal absorber. With the inlet solution temperature and concentration condition used for this study, the equilibrium pressure of the solution is higher than the absorber pressure ($P_{sol}^{eq}(T_{sol}^i; x_{sol}^i) > P_v$) (as already discussed in the Part 4.2). To reach the equilibrium state, the solution has to desorb a part of its water. This is why for all the simulated cases at a low R ratio (close to the adiabatic case) the absorbed flow rate reaches zero and becomes negative (desorption process). For $R < 1$, the limiting flow is the heat transfer fluid; therefore,

variations of this fluid flow rate have a significant impact on the absorbed flow rate. The gap due to the laminar/turbulent transition of the heat transfer fluid appears at a lower R ratio when increasing the solution Reynolds number because of the associated increase in the solution mass flow rate. Because of this flow regime transition, the convective heat transfer coefficient between the absorber wall and the heat transfer fluid (α_{w-htf}) changes significantly. This coefficient is higher in turbulent than laminar flow. The solution cooling provided by the heat transfer fluid is therefore more efficient in turbulent flow, which is characterized by a notable increase in the absorbed flow rate at the flow transition. When $R > 1$ and $Re_{htf} \leq 4000$ the limiting fluid is the solution, and the heat transfer fluid flow is laminar in this operating area; the increase in the absorbed flow rate as the heat transfer fluid flow rate rises is due to the decreases in its outlet temperature and to the increases in its thermal establishing length (that reducing the thermal resistance between the heat transfer fluid and the falling plate). Nearing the transition point, the effect of the change in R on the absorbed flow is less and less significant. In the turbulent flow area of the heat transfer fluid, although the increases in the heat transfer flow rate raise the heat transfer coefficient between the absorber wall and the heat transfer fluid (α_{w-htf}), the thermal resistance caused by this transfer is low enough compared to the other transfer resistances to have no significant impact on the absorbed mass.

The increase in the solution Reynolds number tends

- to raise the mass transfer at the vapor/solution interface due the thickening of the mass boundary layers.
- to increases the film thickness, and the thermal resistance between the interface and the heat transfer fluid. This reduces also the equilibrium concentration at the interface, and thus the driving concentration gradient between the interface and the film. .

Because of these two antagonist phenomena, there is an optimal solution Reynolds number in terms of mass transfer. For example, at R equal to 100 with the operative conditions used for this study, the optimal solution Reynolds number is between 25 and 100.

The mass effectiveness values (ε_m) corresponding to these simulations are plotted in Figure 8b. As described in Section 3, mass effectiveness is defined as the ratio between the absorbed vapour mass flow rate ($\dot{m}_{abs}$) and the maximum absorbable vapour mass flow rate achieved in an infinitely long counter-flow absorber ($\dot{m}_{abs}^{max}$). At a low R ratio, with the operative conditions considered for this study the absorbed flow rate is negative (desorption), so the absorption mass effectiveness is null. Close to the limit absorption R ratio (where ε_m became >0), the maximum absorbed flow is low so mass effectiveness for all configurations rises sharply to reach their maximum value. At an R ratio equal to 1, the maximum absorbed flow ($\dot{m}_{abs}^{max}$) reaches its peak value, the absorption potential is thus maximum. This is why at an R ratio equal to 1 there is a local minimum value for all solution flow rates. The gap observed in the absorbed flow at the laminar/turbulent transition of the heat transfer fluid (Figure 8a) generates a gap in the effectiveness at the corresponding R ratio.

For R ratio larger than 1, the increase of the heat transfer fluid Reynolds number increases the absorbed vapour mass flowrate but do not affect the maximum mass flow rate of vapor that can be absorbed. Therefore, the mass effectiveness increases with the increase of the heat transfer fluid Reynolds number. For R ratio lower than 1, the decrease of the heat transfer fluid Reynolds number decreases the absorbed vapour mass flowrate but reduces also the maximum mass flow rate of vapor that can be absorbed. Due

to these two antagonist phenomena, one can observe a mass effectiveness increase with the decrease of the heat transfer fluid Reynolds number.

5. Conclusion

The effectiveness of an absorber is an indicator of performance that shows the gap between the real transfers and those considering an ideal component. Although absorber effectiveness has already been defined and used in a previous study, this definition cannot describe all the ideal operating conditions of that component.

In this paper the effectiveness of the absorber was generalized to be used for any limiting flow (solution or heat transfer fluid). Absorbers are heat and mass exchangers and therefore the effectiveness can be established to characterize the performance of either of these transfers. These two effectiveness concepts are discussed and directions for use are given.

A model developed to describe the behavior of a plate falling film absorber was introduced. A parametric study was conducted with a simulation tool based on the model presented. This study analyzes the impact of the variation of the solution inlet Reynolds number (Re_{sol}), on the absorbed mass ($\dot{m}_{abs}$) and the mass effectiveness (ε_m) of the absorber. This study highlights the following points:

- There is an optimal solution Reynolds number that maximizes the absorbed mass flow (in this study $25 < Re_f^{op} < 100$).
- Increasing the solution inlet flow rate (increasing solution Reynold number) has a negative impact on the absorber mass effectiveness.
- Whatever the solution Reynolds number, the maximum mass effectiveness is obtained when the heat transfer fluid is the limiting flow but the maximum absorbed mass flow is reached when the heat transfer flow rate tends to infinity

The results of this analysis can be used to guide the design of the falling plate absorber. More generally, the definition of the effectiveness and the model proposed in this study can be used to produce an effectiveness abacus to establish the design of an appropriate absorber.

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