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Numerical and experimental analysis of falling-film exchangers used in a LiBr-H₂O interseasonal heat storage system

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ABSTRACT

This paper investigates heat and mass transfer occurring in an interseasonal absorption heat storage system using LiBr/H₂O as the sorption couple. It focuses on the poor performances of the falling film exchangers with vertical tubes, which are characterized by low flow rate compared to conventional absorption machines. A numerical model was developed for the study and validated with specific experimental results. Comparison of the numerical model to experimental results from the heat storage prototype shows the presence of abnormally high thermal resistance between the falling films and the exchanger surfaces. The deterioration in performance appears to originate in the low wetting rate of the surfaces. A new design of the exchangers is proposed to solve this problem and thus attain the desired performance.

INTRODUCTION

Nowadays environmental damage reduction and sustainable energy supply are considered critical topics. In France, the building sector has raised particular interest since it is responsible for 45% of the final energy consumption and accounts for 14% of greenhouse gas emissions. This led France to commit to reducing the energy consumption of buildings by 38% by the year 2020 [1]. The means to achieve this are improving energy efficiency in buildings and to taking advantage of stored heat in favorable periods when the solar resource is strong (summer) to use it in less favorable periods (winter). Nevertheless, the current major systems for heat storage in buildings use the sensible or latent heat capacity of materials composing the building itself, which is usually limited to a few days' heat storage because of thermal losses.

Sorption and thermochemical processes have been widely used for refrigeration applications and different applications [2,4]. However, as indicated by different authors [5-8], during the last 10 years sorption and thermochemical systems have generated a great deal of interest since they can be used in building heating applications given their capability to store energy for long periods, acceptable heat losses and high energy density. As indicated by Wang et al. [7], sorption and chemical reactions offer three to 30 times greater energy storage density than sensible methods.

Different long-term heat storage system prototypes have been constructed and tested in the last few years; these systems are mainly divided into two sorption technology types: solid/gas adsorption and liquid/gas absorption systems.

Zettla et al. [9], for example, describe an open sorption heat storage system for building heat supply based on natural zeolite clinoptilolite impregnated with solutions of varying salt mixtures (on a dry weight basis: 7.5% LiCl/7.5% MgSO₄ and 7.5% MgSO₄/7.5% MgCl₂). An open adsorption drum reactor with a moving bed was used to characterize these materials avoiding overhydration near the air entrance area. Some of the results indicate that in the

adsorption process up to 5 kWh of released heat can be obtained for a batch of approximately 70 kg, and temperatures in the reactor can rise up to 50°C. Experimental charging temperatures on these tests oscillated between 90 and 110°C.

Weber and Dorer [10] and Weber [11] also developed a single-stage closed absorption prototype for long-term heat storage using an NaOH/H₂O couple. The system consists of a reactor and three storage tanks for water, strong solution and weak solution. Two heat exchangers (one for water and the other for the NaOH solution) are placed in the same reactor with a radiation protection located between them. Theoretical results indicated that at a charging temperature of 120°C, the energy storage density was three times higher compared to traditional hot water storage at a discharging temperature of 65–70°C for domestic hot water supply, and about six times higher at a discharging temperature of 40°C for low-temperature space heating. Nevertheless, experimental results indicated that the discharging process went slower than expected.

N'Tsoukpoe et al. [12] constructed a demonstrative prototype based on the long-term absorption storage cycle of a LiBr/H₂O solution. The system was composed of two storage tanks and a reactor with two vertical falling film heat exchangers and had a 8-kWh storage capacity and a 1-kW discharging rate. Despite advantageous charging performance, the discharging process was unsatisfactory due to an inadequate absorber design. Use of intensification heat and mass transfer additives such as the 2-ethyl-1-hexanol (2EH) did not improve discharging performance. Similar behaviour was observed by Fumey et al [8] on their interseasonal absorption heat storage prototype using an aqueous NaOH solution and horizontal shell and tube heat exchangers. Both systems are characterized by very low flow-rates per unit width of the solution falling film compared to conventional absorption machine (the falling film flow-rate is typically 5 times higher with the same exchanger). This very low

flow rate is due to the very high thermal efficiency required at the absorber, which aims to transfer the maximum heat flux between the falling film and the heat transfer fluid.

In this article the system developed by N'Tsoukpoe et al. [12] for building heating applications based on water absorption in a lithium bromide aqueous solution is analyzed using an appropriate model in order to identify and correct the source of the malfunction during the discharging period.

SORPTION HEAT STORAGE SYSTEM

The principle of the long-term heat storage system is similar to an absorption heat pump cycle, although it does not require the four exchange units (absorber, desorber, condenser and evaporator) to work simultaneously since the interseasonal heat storage is designed to work in a discontinuous way (charge in summer and discharge in winter). Consequently, the four heat exchangers can be combined into two reversible falling-film exchangers situated inside the same reactor where one heat exchanger operates as a desorber and the other as a condenser in the charging period or as an absorber and an evaporator in the discharge period, respectively. This modification also requires at least two storage tanks, one for the LiBr solution and the other for water.

Figures 1 and 2 show the functioning and the main components of the storage system. The components of the system are: a reactor (comprising the desorber/absorber and the condenser/evaporator), a LiBr aqueous solution (absorbant) tank and a water (absorbate) storage tank [12, 13]. Both tanks are placed underground.

At the beginning of the charging period (in spring) the solution stored in the solution tank is diluted and at a temperature of about 15°C. The solution is pumped from the solution tank to the generator (desorber) where it is heated by a heat transfer fluid coming from the solar collectors at a temperature above its saturation temperature. It releases vapor before flowing

back to the solution tank. The vapor emitted by the solution condenses in the condenser, which is cooled by a heat transfer fluid (HTF) coming from a heat sink (cooling tower or geothermal source, the latter being at about 15°C). The water leaving the condenser flows to the water tank. During the charging phase the mass of the solution in the solution tank decreases progressively at the same time that the salt concentration increases; on the other hand, the mass of water in the water tank increases.

During the discharging period (in winter) the concentrated solution is pumped from the solution tank to the absorber while the water is pumped from the water tank to the evaporator. The water at the evaporator receives heat from an HTF coming from a heat source such as a geothermal source (which is at about 15°C [13]). The water vapor produced is absorbed by the solution at the absorber and the useful heat produced is transferred to an HTF linked to a loop for dwelling space heating. The diluted solution leaving the absorber returns to the solution tank. Similarly, the residual water at the evaporator returns to the water tank. During the discharging phase, the mass of the solution in the solution tank increases progressively at the same time that the salt concentration decreases; on the other hand, the mass of water in the water tank decreases.

During the year, there may be many cycles of repeated charging and discharging phases, depending on the solar heat availability and the heating needs of the building.

MODELING THE REACTOR

As mentioned in the previous section, the main component of the interseasonal heat storage system is the reactor. Two reversible falling-film exchangers are situated inside this reactor.

In this section a simulation model developed to study the behavior of each heat exchanger inside the reactor of the interseasonal heat storage system is presented.

The one dimensional model considers a metallic plate, an HTF and a falling film, as shown in Figure 3. The falling film (LiBr solution or water) flows on the external surface of the plate as the HTF flows in the isolated canal in contact with the internal plate's surface. Mass and energy exchanges occur between the reactor's vapor and the falling films while the HTF only exchanges heat with the plate.

Different hypotheses have been considered to describe the heat and mass transfer mechanisms inside the falling film. These hypotheses are commonly used in other studies [14-17].

- (1) Noncondensable gases are not present in the vapor, so the resistance to vapor absorption or condensation at the interface of the falling film can be ignored.
- (2) Vapor in the reactor is saturated.
- (3) Convective heat transfer from the liquid phase to the adjacent vapor is ignored.
- (4) The film flow is fully developed steady-state downward and laminar.
- (5) The surface waves on the liquid film flows are not considered.
- (6) The system is in steady-state conditions (each time).
- (7) The vapor absorption or desorption rate is small compared to the mass flow rate of the film.
- (8) Vapor is in equilibrium with the film at the liquid free interface.
- (9) No shear forces are exerted on the liquid by the vapor.
- (10) Fluid velocity is zero at the interface between the plates and the films.
- (11) The physical properties of the liquid film are considered to be constant.
- (12) The film thickness is very small compared to the length of the plate.
- (13) The net pressure force component is very small compared to the body force component.
- (14) The momentum components along the plate are negligible.

Considering previous hypotheses, in the following subsections the model developed for the absorption/desorption, evaporation/condensation heat exchangers is described.

Absorption/desorption heat exchanger

Heat and mass transfer along the plate and the film interface

Considering hypotheses 4, 7, 8, 9, 11 and 12, the film width at each position along the plate can be expressed as:

$$\delta = \sqrt[3]{\frac{3\dot{m}_z\mu}{\rho^2 g L}} \quad (1)$$

where $\dot{m}_z$ is the mass flow of the liquid film at position z and L the width of the plate wetted by the liquid film.

Since at the falling film interface the absolute flux of LiBr is zero due to its low volatility, the mass flux of H₂O absorbed or desorbed per surface unit by the binary mixture of LiBr-H₂O, $\dot{m}''_{H_2O,abs/des}$, can be expressed as:

$$\begin{aligned} \dot{m}''_{H_2O,abs/des} &= \left[\frac{\rho_{st} D_{LiBr/H_2O}}{1 - x_{H_2O}} \frac{\partial x_{H_2O}}{\partial y} \right]_{y=\delta} \\ &= k_{m-st,int} [\rho_{st,int} x_{H_2O,int} - \rho_{st} x_{H_2O}] \end{aligned} \quad (2)$$

where $k_{m-st,int}$ is the vapor mass transfer coefficient at the interface.

If $\dot{m}''_{H_2O,abs/des}$ is positive, vapor absorption occurs at the interface. Conversely, if $\dot{m}''_{H_2O,abs/des}$ is negative, vapor desorption is produced at the interface.

The energy balance at this interface is expressed as:

$$\dot{m}''_{H_2O,abs/des} [h_{vap} - h_{p-H_2O,int}] = \left[\lambda_{st} \frac{\partial T}{\partial y} \right]_{y=\delta} = k_{T-st,int} [T_{st,int} - T_{st}] \quad (3)$$

where h_{p-H_2O} is the partial enthalpy of H₂O in the binary solution and $k_{T-st,int}$ the vapor/film heat convective coefficient at the interface. The left-hand side of equation (3) expresses the heat of vapor absorption at the solution interface.

Along the plate ($y=0$), mass transfer is zero, whereas heat transfer is described by:

$$\dot{q}''_{st,w} = \left[\lambda_{st} \frac{\partial T}{\partial y} \right]_{y=0} = k_{T-st,w} [T_{st} - T_{st,w}] \quad (4)$$

where $k_{T-st/w}$ is the solution heat transfer coefficient at the wall.

Heat and mass transfer coefficients at the vapor/film interface and at the wall/film interface at each position along a vertical plate heat exchanger were determined analytically by Brauner [18]. These coefficients were obtained solving the film governing equations using an integral formulation and expressing equations in a dimensionless form. This approach considered concentration and temperature parabolic profiles across the film that could respect the boundary conditions at each interface.

Brauner [18] expressed the transfer coefficients using nondimensional numbers. Sherwood and Nusselt numbers, as a function of the downstream distance, for the case of isothermal or adiabatic conditions, are defined as follows:

$$Sh_{st,int} = \frac{k_{m-st,int} \delta}{D_{LiBr-H_2O}} \quad (5)$$

$$Nu_{st,int} = \frac{k_{T-st,int} \delta}{\lambda_{st}} \quad (6)$$

$$Nu_{st,w} = \frac{k_{T-st,w} \delta}{\lambda_{st}} \quad (7)$$

For isothermal cases in which the $\frac{x_{H_2O} - x_{H_2O,int}}{x_{H_2O}}$ ratio (nominal driving force of the absorption/desorption process) is near zero, evaluation of the Sherwood and Nusselt numbers with the position along the plate are shown in Figure 4.

In the following section, correlations given by Brauner [18] and plotted in Figure 4 are used to make a volume control mass and energy balance along the heat exchanger as part of the approach used by the model. Brauner [18] considered the case of a solution falling along an isothermal plate, the temperature of the solution at the entrance being equal to the temperature of the plate. The heat and mass transfer correlations describe the spatial evolution of heat and mass transfers along the plate (effects of the developments of the thermal and diffusion boundary layers) as a function of the flow rate. In this configuration, zero heat transfer develops on the upper part of the plate until the thermal boundary layers reach the heat transfer surface. Brauner's heat transfer correlation along the plate in the entrance region was therefore modified here to take into account the heat transfer between the plate and the solution in the entrance region area, the temperature of the solution here differing from the temperature of the plate at the entrance.

Mass and energy balance

The vertical plate exchanger is discretized in n segments. The mass and energy balance is determined on the control volumes and correlations and the hypotheses described in the sections above are used. It must be indicated that as a first approach the system was considered in the co-current condition.

The corresponding balance equations on segment k are shown below.

Energy balance of the LiBr solution film:

$$-\dot{m}_{st-o,k}h_{st-o,k} + \dot{m}_{st-i,k}h_{st-i,k} + \dot{m}_{H_2O,abs/des,k}h_{vap} + k_{T-st,w,k}\Delta zL\left[T_{w,st,k} - \frac{T_{st-o,k} + T_{st-i,k}}{2}\right] = 0 \quad (8)$$

Energy balance at the interface between the LiBr solution film and the water vapor:

$$\dot{m}_{H_2O,abs/des,k}\left[h_{vap} - h_{p-H_2O,int,k}\right] - k_{T-st,w,k}\Delta zL\left[T_{st-int,k} - \frac{T_{st-o,k} + T_{st-i,k}}{2}\right] = 0 \quad (9)$$

Mass balance in the LiBr solution film:

$$-\dot{m}_{st-o,k} + \dot{m}_{st-i,k} + \dot{m}_{H_2O,abs/des,k} = 0 \quad (10)$$

Water mass balance in the LiBr solution film:

$$-\dot{m}_{st-o,k}x_{H_2O-o,k} + \dot{m}_{st-i,k}x_{H_2O-i,k} + \dot{m}_{H_2O,abs/des,k} = 0 \quad (11)$$

Mass transfer at the interface between the LiBr solution film and the water vapor:

$$\dot{m}_{H_2O,abs/des,k} - k_{m-st,int,k}\Delta zL\left[\rho_{st,int,k}x_{H_2O,int,k} - \frac{\rho_{st,o,k} + \rho_{st,i,k}}{2} \frac{x_{H_2O,o,k} + x_{H_2O,i,k}}{2}\right] = 0 \quad (12)$$

Equilibrium condition at the interface between the LiBr solution film and the water vapor:

$$T_{st,int,k} = f(x_{H_2O,int,k}; P_{vap,sat}) \quad (13)$$

Heat transfer between the LiBr solution film and the metallic plate:

$$\frac{\lambda_w}{e_w}\Delta zL[T_{w,st,k} - T_{w,hf,k}] - k_{T-st,w,k}\Delta zL\left[\frac{T_{st,o,k} + T_{st,i,k}}{2} - T_{w,st,k}\right] = 0 \quad (14)$$

Heat transfer between the heat transfer fluid and the metallic plate:

$$k_{T-htf,w,k} \Delta z L \left[\frac{T_{htf,o,k} + T_{htf,i,k}}{2} - T_{w,htf,k} \right] + \frac{\lambda_w}{e_w} \Delta z L [T_{w,htf,k} - T_{w,st,k}] = 0 \quad (15)$$

Energy balance of the HTF:

$$-\dot{m}_{htf,o,k} h_{htf,o,k} + \dot{m}_{htf,i,k} h_{htf,i,k} + k_{T-htf,w,k} \Delta z L \left[T_{w,htf,k} - \frac{T_{htf,o,k} + T_{htf,i,k}}{2} \right] = 0 \quad (16)$$

The heat and mass transfer coefficients in the film depend on its Reynolds number, and thus on the width of the plate wetted by the liquid film L . This parameter influence the heat and mass transfer substantially, as will be seen in the following. The effect of increased transport and the degradation of heat transfers due to the increase of the Reynolds number in laminar flow is considered, but the effects of the development of surface waves on the liquid film are neglected, given that the Reynolds number of the solution is lower than 30. The effects of the surface waves on heat and mass transfer will be discussed below.

The heat transfer coefficient between the HTF and the plate is given by the Colburn correlation (Kakaç and Liu [19]) depending on the HTF flow conditions.

The impact of the partial wetting of the surface by the solution on the performance of the exchanger can be roughly estimated by the model. For this purpose, heat and mass transfer are estimated on the basis of a Reynolds number based on the average plate width wetted by the liquid film. The determination of the fin effect affecting the heat transfer between the HTF and the solution requires knowing the distribution of the liquid along the surface. This information is not available for the interseasonal experiment analyzed in this paper. Therefore, two limit cases are considered when partial wetting is analyzed: the optimistic case, which considers a fin efficiency equal to 1 (Figure 5 a), and the pessimistic case, which considers a fin efficiency equal to 0 (Figure 5 b).

Solving procedure

$P_{vap,sat}$ and $T_{vap,sat}$ are assumed to be known. The temperature, water mass fraction and mass flow rate at the entrance are also known as well as the inlet temperature and mass flow rate of the HTF; then equations (8) to (16) define a system of nine equations and nine unknown variables $(\dot{m}_{st,x,o,k}; \dot{m}_{H_2O,abs / des,k}; T_{st,o,k}; T_{st,int,k}; T_{w,st,k}; T_{w,htf,k}; T_{htf,o,k}; x_{H_2O,int,k}; x_{H_2O,o,k})$ for each elementary volume of the exchanger.

A code for simulating the absorption/desorption heat exchanger model was developed in Matlab. The numerical results obtained by this model are presented and validated in the following sections. Thermophysical property correlations for the LiBr solution and water were obtained from work developed by different authors [20-24].

Heat exchanger model for the evaporation/condensation process

In a similar way to the absorption/desorption process, a model was developed for the evaporation/condensation process. The same nodal approach and hypothesis indicated in the previous subsection were used.

The convective heat transfer coefficients: $k_{T-st,w,k}$, $k_{T-st,int,k}$ were calculated previously, again using Brauner's [18] results.

Heat and mass transfers along the evaporator are identified through the values obtained for the variable $\dot{m}_{H_2O,eva / con,k}$ at each position. A positive value of $\dot{m}_{H_2O,eva / con,k}$ indicates that water was evaporated from the water liquid film and a negative value indicates that water was condensed from water vapor (this approach also requires having a liquid water film flow at the entrance of the heat exchanger that is different from zero to avoid inconsistent divisions).

At the condenser, a Nusselt condensation approach [16] is considered.

Heat exchanger model for the evaporation/condensation process

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The convective heat transfer coefficients: $k_{T-st,w,k}$, $k_{T-st,int,k}$ were calculated previously, again using Brauner's [18] results.

Heat and mass transfers along the evaporator are identified through the values obtained for the variable $\dot{m}_{H_2O,eva/con,k}$ at each position. A positive value of $\dot{m}_{H_2O,eva/con,k}$ indicates that water was evaporated from the water liquid film and a negative value indicates that water was condensed from water vapor (this approach also requires having a liquid water film flow at the entrance of the heat exchanger that is different from zero to avoid inconsistent divisions).

At the condenser, a Nusselt condensation approach [16] is considered.

Coupling of the absorption/desorption and evaporation/condensation heat exchangers

A coupling procedure is necessary in the absorption/desorption and evaporation/condensation exchangers. This approach considers that the vapor generated by the evaporator (desorber) is entirely absorbed (condensed) in the absorber (condenser), with the evaporator/absorber (desorber/condenser) working at the same pressure (Figures 3 and 6).

Given the entrance conditions of the LiBr solution film, the HTF and the liquid water film, the model finds the pressure condition $P_{vap,sat}$ that allows obtaining vapor mass flow evaporated (condensed) equal to the vapor mass flow absorbed (desorbed), as indicated in Equation (17).

$$\left(\sum_{k=1}^n \dot{m}_{H_2O,abs/des,k} - \sum_{k=1}^n \dot{m}_{H_2O,eva/con,k} \right)_{(T_{vap-sat}, P_{vap,sat})} \quad (17)$$

Validation of the model

Absorption experimental tests were conducted to validate the model (Figure 7). The experiment consists of a tight vessel filled with saturated water vapor. Two grooved falling film plate exchangers were implanted in the vessel: an absorber using a LiBr solution and an evaporator. The stainless-steel plates are 50 cm high and 50 cm wide. The contact angle between the solution and the stainless-steel plate is close to 90°. The widths of the grooves (4 mm) and the solution flow rate were chosen to ensure the complete wetting of the base of the grooves (as will be developed further), and to develop a two dimensional laminar flow with no wavelets on the surface (the surface tension effects associated with pinning the triple line on the side walls of the grooves prevent the formation of wavelets and ensure a laminar flow regime for the entire Reynolds range studied). The model is therefore compared to experiments corresponding to its condition of use. The LiBr solution and the water are pumped in tanks, distributed along the absorber and evaporator plates and collected at the bottom of the plates before returning back to the tanks. Coriolis mass-flow meters measure the concentration and the flow rate of the solution at the inlet and outlet of the absorber. The vapor mass flow is also measured using mass-flow meters at the liquid inlet and outlet of the evaporator. The amount of solution within the LiBr tank is sufficiently large (51 L) compared to the solution mass flow rate (110 kg h^{-1}) to consider the LiBr mass fraction as constant over the test duration (~ 30 min). The relative uncertainty of the solution mass flow rate, the solution concentration and the absorbed mass flow are respectively 0.2%, 0.08% and 0.4%.

Different experimental tests in absorption/evaporation and desorption/condensation operating modes were conducted. Figure 8 shows the comparison between the experimental and simulated results for the absorbed water mass flow of the solution for different inlet solution LiBr mass fractions and flow rates. The average inlet solution temperature was 26°C, the inlet solution mass fraction range was $0.54 < x_{\text{LiBr},i} < 0.59 \pm 0,006$, the inlet solution Reynolds number range was $78 \leq Re \leq 81$, the absorber HTF inlet temperature was 25°C, the

absorber HTF inlet mass flow was 300 kg h^{-1} . The vapor pressure inside the reactor during the tests was around 13.5 mbar. The average deviation between the simulated and the experimental values was about 7%.

The heat transfer between the absorbant falling film and the heat transfer fluid depends on either the heat released by absorption or the thermal resistance of the falling film. The higher the flow rate, the higher the absorption rate and therefore the higher the heat obtained by the film due to absorption. In contrast, the higher the flow rate, the higher the thermal resistance between the interface of the film (where absorption heat is released) and the heat transfer fluid due to the thickening of the film. An optimum mass flow exists, leading to a maximum heat flux transferred to the heat transfer fluid. This optimum depends on the length of the plate. It is around $1.44 \text{ kg.h}^{-1}.\text{cm}^{-1}$ for lithium bromide falling films flowing over 50-cm-high vertical surface exchangers.

APPLICATION TO INTERSEASONAL HEAT STORAGE

Experimental tests on a constructed prototype of the interseasonal sorption heat storage system described previously were conducted by N'Tsoukpoe and coworkers [12, 25]. The simulation model is used to better understand their experimental results.

Interseasonal heat storage - Experimental setup

The prototype consists of three main components: a LiBr solution tank, a water tank and a reactor (Figure 9a). Inside the prototype reactor, two shell and tube exchangers are placed. At each heat exchanger, the LiBr solution or water flows on the tube's internal surface while the HTF flows on the tube's external surface (shell side). The tube is in $\text{CuZn}_{22}\text{Al}_2$ brass. Figure 9b shows the distribution part for the films flowing along the vertical internal surface of the tubes, where at each tube top three 0.4-mm injection points were drilled.

Vapor produced by the desorption/evaporation process flows through the top or bottom of each tube to the condenser/absorber (Figure 9c).

The prototype is instrumented to measure temperature, pressure and the mass fraction of the fluids. Each heat exchanger is connected to a thermal module that can provide a controlled flow rate and temperature for the HTF. The module connected to the desorber represents the solar collectors during the charging tests and the building during the discharge tests. The module connected to the condenser/evaporator simulates a geothermal exchanger. Finally, two additional modules were installed to keep the storage tanks in constant surrounding temperature conditions [12, 25].

Experimental results obtained with the interseasonal sorption heat storage system described were compared to the model in desorption/condensation functioning mode (charge) and in absorption/evaporation operating mode (discharge).

Comparing the model in desorption/condensation operating mode

The experimental inlet conditions on the desorber and condenser for the LiBr solution falling film and the HTF are described in Table 1. The inlet conditions chosen correspond approximately to the conditions required by the system to work in charge mode. The experimental LiBr mass concentration varied between 54% and 56%; these concentrations are within the system's working range, which varies between 54% and 60% (higher concentrations could imply crystallization of the solution on the desorber).

The experimental inlet conditions mentioned in Table 1 were used as inlet conditions in the simulation model. In both cases, experimentation and simulation, the movement of the HTF with regard to the falling films was in counter-current. Figure 10 shows the comparison between experimental and simulation results for the LiBr solution film and HTF leaving the

reactor. The parameters compared are the LiBr solution film temperature and mass fraction at the reactor's outlet as well as the HTF outlet temperature at the desorber and condenser.

At the beginning of the test, the solution tank is filled with homogeneous solution ($x = 54.2\%$; $T_{st,i} = 10^{\circ}\text{C}$). The desorption process is active since an approximately 1% concentration difference occurs between the inlet and the outlet of the desorber. The diluted solution is pumped at the top of the tank and the concentrated solution is re-injected at the bottom of the tank. The tank works in a quasi-plug-flow mode, as can be seen in Figure 10a and 10b: an abrupt concentration modification appears at the inlet of the desorber 1:04 h after the beginning of the test, corresponding approximately to the time needed for a particle to shift from the inlet to the outlet of the reservoir if no mixing occurs. The increase of the solution's inlet temperature and concentration impact the heat transfer between the solution and the HTF (Figure 10c). However, it seems to have a negligible effect on mass transfers within the solution since heat transfer at the evaporator is not affected, demonstrating a constant evaporation rate (Figure 10d).

Numerical simulations considering completely wetted surfaces (S1_100%_S2_100%) were compared to the experimental results (S1 refers to the wetting rate of the desorber surface; S2 refers to the wetting rate of the condenser surface). The qualitative changes of the variables are reproduced. However, numerical simulations overestimate the performance of the system. This can be explained by the low wetting rate of the desorber and evaporator surfaces. Dry patches are known to develop at low flow rates and to have a great impact on heat transfer (Roques and Thome [26]). Consequently, partial wetted surfaces of "S1_60%" and "S1_12%" were considered in the simulations. For cases in which the wetted surface is partially wetted, a fin effect appears. Given that the distribution of the liquid film on the surface is unknown, two limit cases are considered (Figure 6): the optimistic case denoted "1F" considers a fin efficiency equal to 1, whereas the pessimistic case denoted "2F" considers a fin efficiency

equal to 0. The simulated results indicate that for “S1_12%”, the optimistic case (1F) presents a better heat transfer across the desorber’s metallic exchange surface compared to the pessimistic case (2F). This is observed in Figures 10a, 10b and 10c where outlet-inlet temperatures and desorbed mass differences are larger in case 1F than in case 2F. Heat transfer fluid temperatures at the exit of the desorber and the condenser are then correctly predicted by the simulation whereas the temperature and the concentration of the solution at the exit are respectively overestimated by 10K and 0.8%. The model does not consider all the physics involved in the absorption process, especially 2D and 3D instability that can develop in falling films and rivulets, the spatial distribution of the film in case of partial wetting, the presence of noncondensable gases, the curvature effects of the wall, etc. Most of these phenomena are second-order effects compared to the wetting rate: considering the Reynolds range of the flows inside the tubes, the thickness of the falling films is sufficiently small (<0.4 mm) to ignore the curvature effects (tube diameter, 16 mm) as well as the intensification of heat and mass transfer due to the waviness of the flows (Gambaryan-Roisman et al [27]) (the order of magnitude of the intensification factor is estimated in the range 10–15% (Yoshimura et al. [28])). The noncondensable gas rate has not been estimated in the experiment and is assumed to be sufficiently small to have no significant impact on heat and mass transfers. The distribution of the falling film along the surface has an impact on the fin efficiency, as can be seen in Figure 10 (difference between the results obtained with fin efficiency equal to 0 or 1). However, the simple model is able to reproduce the tendencies and the orders of magnitude of the heat and mass transfer, showing that the wetting rate is a key parameter of the system.

The hypothesis that the LiBr solution has low wettability on the exchange surface is in agreement with studies conducted by Drelich et al. [29], which indicate that to have high wettability on surfaces, usually a chemical surface treatment must be applied. The hypothesis

of low LiBr solution wettability on the heat exchangers' brass metallic surfaces with no treatment is further described, which is in agreement with the simulation results.

Comparing the model in absorption/evaporation operating mode

As in the previous section, the experimental results obtained by N'Tsoukpoe and co-workers. [12, 24] in absorption/evaporation operating mode of the interseasonal sorption heat storage system prototype are compared to our simulation model; the results are shown in Figure 11.

Experimental inlet conditions on the absorber and evaporator for the LiBr solution falling film, water film and the HTF are described in Table 2. The inlet conditions chosen correspond approximately to the conditions required by the system to work in discharge mode. The experimental LiBr mass concentration varied between 55% and 54%.

As in desorption, the absorption process is effective since an approximately 1% concentration difference occurs between the inlet and the outlet of the absorber. The concentrated solution is pumped at the top of the reservoir but contrary to the previous case, the diluted solution is re-injected at the top of the reservoir in the form of a plunging jet. The mixing zone is limited to the top of the reservoir, which works in a quasi-perfectly mixed mode, as can be seen in Figure 11b (linear decrease of the concentration with time), the heat losses to the surroundings leading to an increase followed by a stabilization of the solution temperature, leaving the tank to be injected in the absorber (Fig. 11a). The heat transfer between the solution and the HTF is limited since the temperature difference of the HTF between the inlet and outlet is about 0.2°C, whereas it should be higher than 5°C in case of perfect wetting film (Fig. 11c) (the oscillation of the temperature of the HTF flowing into the absorber enclosed between 26 and 26.5°C is due to regulation problems and not to a physical phenomenon). The heat transfer between the water and the HTF at the evaporator is also very

limited compared to the one expected with an entirely wetted surface: the temperature difference of the HTF between the inlet and outlet is about 0.6°C, whereas it should be about 2.8°C in case of perfect wetting film (Figure 11d).

As for the charging mode, numerical simulations considering completely wetted surfaces (S1_100%_S2_100%) substantially overestimate the heat transfer with the HTF (S1 refers to the wetting rate of the absorber surface; S2 refers to the wetting rate of the evaporator surface). Better matching between experimental and simulated results is obtained for partial wetting of the heat transfer surfaces. The concentration of the solution at the exit of the absorber predicted by the model agree with the experimental measurements considering their uncertainty ranges assuming a wetting rate equal to 20% and a fin efficiency equal to 1. The relative difference of the solution heating $\left(\frac{\Delta T_{st,abs-sim} - \Delta T_{st,abs-exp}}{\Delta T_{st,abs-sim}} \right)$ for this configuration is 10%. Comparisons between simulation and experiments are worst for the heat transfer fluids at the absorber and the evaporator: The temperature of the heat transfer fluid at the exit of the absorber is overestimated by 1.5K, leading to a relative difference between simulation and experiment of around 80%. The temperature of the heat transfer fluid at the exit of the evaporator is underestimated by 0,4K leading to a relative difference between simulation and experiment of around 40%. Such differences on temperature are characteristic of an underestimation of the heat transfer within the plates. This can be seen when comparing experiment and modeling using fin efficiency equal to 0. The results globally fit better : the relative differences of the concentration, the temperature of the solution, the temperature of the heat transfer fluid at the absorber and the temperature of the heat transfer fluid at the evaporator are respectively equal to 30%, 10%, 40% and 30%.

Higher temperature differences of the HTF are observed in desorption/condensation mode compared with the absorption/evaporation mode, even if the wetting rate is smaller

compared to the other mode. This is due to the temperature differences between the falling films and the HTF in the absorption/evaporation operating mode, which are significantly smaller than in desorption/condensation mode: the temperature difference between the solution and the HTF is about 4.5 times higher during desorption compared to absorption. Heat transfers between the fluids as well as the HTF temperature difference between inlet and outlet decrease, leading to higher sensitivity of the results to measurement uncertainties. The cumulative influence of all the parameters can explain a large part of the differences between the model and the experimental results. The measurement uncertainties also lead to deviations with the model. Nevertheless, the influence of the wetting rate on the performance of the exchangers is similar to that obtained in desorption/condensation.

The comparison of the experimental results with the heat and mass transfer model show the large influence of the wetting rate on the performance of the system. Other parameters also impact the performance of the system, such as the hydrodynamic instabilities of the falling films and the liquid distribution along the surface, but they appear to be second-order parameters in the system's condition of use.

Nevertheless, the design of the exchangers' internal falling film prevents visualization of the flow (the falling films develop along the internal surface of 14-mm-diameter tubes in a low pressure environment). To confirm this hypothesis, wetting tests on brass and stainless-steel surfaces were performed, as described in the following section.

Discussion

Different absorption experiments were conducted previously involving falling films on vertical tubes. Medrano et al. [30] studied absorption of water vapor in falling film of water–lithium bromide inside a vertical tube ($D_i = 22.1$ mm). They carried out wetting tests starting with high flow rates, which were reduced at constant intervals until the film broke down, which

was observed at a Reynolds number of about 40. Takamatsu et al. [31] observed the breakdown of the LiBr aqueous solution liquid films covering the internal surface of copper tubes ($D_i = 16$ mm) at a Reynolds number close to 32. Considering the diameter of the exchanger ($D_i = 16$ mm), the Reynolds number ($Re < 30$), the liquid distribution at the entrance of the tube (the liquid is distributed through three injection holes (0.4 mm in diameter) or overflow if the holes are not sufficient) and the operating mode (no prior procedure is applied to ensure complete wetting of the surface at the beginning of the tests), the development of rivulets instead of uniform falling film on the internal surface of the tubes as predicted by the model makes sense. Wetting rates of 12% or 20% lead to rivulet width L_w equal to 1.8 and 2.9 mm, respectively, in case three similar rivulets are formed in each tube. The width of the rivulets is smaller than the capillary length (the capillary length $L_{cap} = \sqrt{\frac{\sigma}{\rho g}}$ is close to 2.25 mm) so the shape of the cross section can reasonably be assumed to be almost circular. The mean thickness of the rivulet can be estimated assuming uniform liquid distribution over the wetted area of the tube and parabolic velocity profile (laminar regime): $\delta \approx \sqrt[3]{\frac{3\dot{m}_a \mu}{\rho^2 g L}}$. The mean thickness of uniform falling film is between 0.4 and 0.5 mm for absorption and desorption conditions. This average thickness is small compared to the average thickness that should be obtained with the cylindrical shape of the rivulet mentioned above and the contact angle equal to 90° : the mean thickness of the rivulet is then close to that obtained on a flat surface ($\delta_{fp} = L\pi/4$). It is close to $\delta = 0.7$ mm for the desorption condition and close to $\delta = 1.15$ mm for the absorption condition. The average thickness of the rivulets estimated using the wetting area may be obtained with a spherical shape, considering the contact angle smaller than 90° , and thus better wettability properties of the surface.

The wetting rate depends on many parameters such as the contact angle, the surface structuration, the flow rate, the liquid distribution, the temperature of the plate, the width of the plate (or the diameter of the tube), etc. The lowest flow rate needed to ensure that the surface remains covered by a continuous thin liquid film increases with the reduction of the contact angle (El Genk and Saber [32], Lee et al. [33]). The contact angle between water or lithium bromide solution with nonoxidized metal plates is typically enclosed between 80 and 90°. The wettability can be improved by chemical treatment or oxidization affecting the surface energies or the low-scale roughness of the surface, as reported by Drelich et al. [29]. Chemical attacks can occur in operation, improving the performance of the exchangers. This is typically the case when using copper or brass materials and LiBr solutions, as will be explained below. The operating conditions such as the temperature difference between the film and the surface also impact the wetting rate through Marangoni effects (Zang et al. [34], Budiman et al [35]). The oxidation of the brass surface by the solution is in agreement with the estimations of the film thickness mentioned previously and was confirmed after the test by visual observation of the exchanger surface. Nevertheless, the increase of the wettability due to oxidation is not sufficient to obtain a high wetting surface.

The validation of the assumptions related to the development of solution rivulets on the internal surface of the tubes have led to wettability experiments on vertical plates. As mentioned above, the width of the expected rivulets is small and their average thickness negligible compared to the radius of the tube. Therefore curvature effects can be neglected. The experiment involves a vertical flat plate. The wettability performance of water and LiBr solution have been investigated on three different plates 10 cm wide and 50 cm high: a stainless-steel plate, an nonoxidized brass plate, and a brass plate oxidized with a LiBr solution for 3 days. The wettability performance of the plates was studied using an experimental setup described in Figure 12. The LiBr solution or pure water is pumped into the tank, distributed

along the plates before returning back to the tank. A Coriolis mass-flow meter measures the density, the temperature and the flow rate of the liquid. The concentration of the solution is calculated from density and temperature measurements (Yuan and Herold [22]). Falling film visualizations are performed using a CCD camera located in front of the plate. The wetting rate is determined using the ImageJ image-processing software [36]. All tests were made at atmospheric pressure.

The static contact angle estimated using sessile drop between water and the oxidized brass surface is smaller and close to 60° . The wetting rate is known to be controlled by the advancing contact angle during the wetting process, and the receding contact angle during the de-wetting process, leading to hysteresis effects. The wetting rate, defined as the ratio of the wetted area related to the entire surface, was determined at an increasing flow rate up to about $2 \text{ kg.h}^{-1}.\text{cm}^{-1}$ and at a decreasing flow rate down to zero.

Falling films developing on the vertical plates on the flow range studied are characterized by the development of several rivulets that can merge along the plate (Figure 13).

The changes in the wetting rate as a function of the flow rate for both plates are shown in Figure 14. The wetting rate of the water falling film on nonoxidized plates (stainless steel plate or brass plate) is limited to 12%. The wetting of the stainless-steel plate increases regularly with the flow rate and reaches 12% for a mass flow rate of $1.2 \text{ kg.h}^{-1}.\text{cm}^{-1}$. It remains nearly constant for the mass flow rate between 1.2 and $2.5 \text{ kg.h}^{-1}.\text{cm}^{-1}$. No significant hysteresis is observed when decreasing the flow rate. The wetting rate of the nonoxidized brass plate increases with the mass flow rate in increments: it increases with the flow rate up to 5% for a mass flow rate of $0.5 \text{ kg.h}^{-1}.\text{cm}^{-1}$. It remains nearly constant for the mass flow rate included between 0.5 and $1.5 \text{ kg.h}^{-1}.\text{cm}^{-1}$ and increases again to reach a new level of about 8% for mass

flow rates between 2 and 2.5 kg.h⁻¹.cm⁻¹. The evolution of the wetting rate is much more regular for decreasing flow rates.

The wetting rate of water falling films on the oxidized brass plate increases in increments as a function of the water flow rate. Its amplitude is four times higher than for the nonoxidized brass plate. This greater ability to wet the surface is due to the reduction of the contact angle, as mentioned above. The plate remains at the same wetting rate when reducing the mass flow rate until the mass flow rate becomes smaller than 0.5 kg.h⁻¹.cm⁻¹. Then the wetting rate decreases abruptly. This behavior shows a high hysteresis in the apparent contact angle that can be attributed to the development of a microporous layer on the surface during oxidization.

The water mass flow rate per unit width of the tube exchanger was about 0.23 kg.h⁻¹.cm⁻¹ during the absorption experiments presented above. This flow rate leads to a wetting rate close to 10% of the internal surface of the tube in increasing flow rate conditions and about 25% of the internal surface of the tube in decreasing flow rate conditions (the mass flow being previously carried up to 1,5 kg.h⁻¹.cm⁻¹). Such values are consistent with the wetting rate estimated with the model, i.e. 20% (Figure 11).

The wetting rate of the LiBr solution on nonoxidized plates (stainless steel plate or brass plate) is about twice as high as the one obtained with water. The differences between the stainless steel plate and the brass plate are relatively small, even if the wetting rate of the brass plate increases incrementally rather than the stainless steel plate, as with water. The wetting rate increases regularly with the flow rate up to 20% for a mass flow rate of 2 kg.h⁻¹.cm⁻¹. The plate remains at the same wetted level when reducing the solution mass flow rate until the mass flow rate becomes smaller than 0.25 kg.h⁻¹.cm⁻¹. Then the wetting rate decreases abruptly. This behavior shows a high hysteresis in the apparent contact angle that can be attributed to a salt deposition on the surface.

The solution mass flow rate per unit width of the tube exchanger is about $0.6 \text{ kg.h}^{-1}.\text{cm}^{-1}$ during the desorption experiments described above, and about $1 \text{ kg.h}^{-1}.\text{cm}^{-1}$ during absorption experiments. These flow rates lead to wetting rates close to 10% and 15% of the internal surface of the tube in the increasing flow rate and about 25% of the internal surface of the tube in decreasing flow rate conditions (the mass flow being previously carried up to $2 \text{ kg.h}^{-1}.\text{cm}^{-1}$). Such values are consistent with the wetting rate estimated with the model in desorption mode (i.e. 12% , figure 10) and in absorption mode (ie 20%, Figure 11).

As mentioned previously, in the experimental tests reported by N'Tsoukpoe and coworkers [13-25], shell and tube heat exchangers were used for the sorption and evaporation tests where the aqueous LiBr solution and water flowed on the inner surface of the metallic tubes. The material used for these tubes was brass ($\text{CuZn}_{22}\text{Al}_2$). The film distribution in this system was certainly not optimal since it consisted of only three injection points (0.4 mm in diameter) located at the top of each brass tube, and the maximum normalized flow rate on the inner surfaces was limited to $1.5 \text{ kg.h}^{-1}.\text{cm}^{-1}$.

Even if the estimation of the wetting rate using the model 2D steady-state laminar model is coarse, it shows that the efficiency of the system can be significantly improved by increasing the wetting rate. The next section is devoted to the development of an exchanger geometry, ensuring a high wetting rate at a low flow rate as needed by the application.

New exchanger design

Building heating is provided by the absorption of water vapor generated by a water falling film at the evaporator, by the LiBr solution falling along the absorber surface. The heat transferred to the HTF at the absorber depends on the efficiencies of both the evaporator and absorber. The efficiencies of these falling film exchangers increase with the increase of the wetted area (increase of the liquid–vapor interface) and the reduction of the film thickness. For

high flow rates, the surface of the evaporator and the absorber can be entirely wetted by a thick film. The reduction of the flow rate induces a reduction of the film thickness, increasing the exchanger efficiency. When the film becomes too thin, film breakdown appears, reducing the wetted area and lowering the efficiency of the exchanger. In the present study, and contrary to air conditioning absorption heat pumps, the objective is not to absorb the maximum amount of vapor, but to maximize the heat transfer from the solution falling film to the HTF at the absorber. For this purpose, the mass flow rate per unit width of the falling film has to be low enough to maximize the thermal efficiency of the absorber, and high enough to ensure acceptable exchanger compactness. An optimization of the inlet conditions therefore has to be found.

For flow rates close to the optimum flowrate ($1.44 \text{ kg.h}^{-1}.\text{cm}^{-1}$ for 50-cm-high plate exchangers), the wetting rate of solution falling films is close 15%, as shown above, leading to very low absorber efficiency. Macro-structuration of the exchanger surface was therefore developed to increase the wetting rate. It consists in grooves machined on the surface to limit the formation of rivulets and take advantage of surface tension effects. Grooved surfaces are commonly used in chemical engineering and in heat transfer engineering to maintain liquid films over surfaces. For this purpose, five grooved plates characterized by grooves whose width was equal to 0.5, 1, 2, 4 and 8 mm were machined on stainless steel plates and tested (for all these plates, the depth of the grooves was set to 1 mm).

Because of the contact angle between the solution and the stainless steel surface close to 90° , the liquid does not flood the grooves when the grooves' width is narrower than the capillary length. The 0.5-, 1- and 2-mm-wide grooves are therefore unable to ensure a reasonable wetting rate. The liquid on the 8-mm-wide groove never wet the entire base of the groove for a solution mass flow rate below $5 \text{ kg.h}^{-1}.\text{cm}^{-1}$. The 4-mm-wide groove is the only design ensuring complete wetting of the surface for a solution mass flow rate above $2 \text{ kg.h}^{-1}.\text{cm}^{-1}$.

$^1.\text{cm}^{-1}$. The performance of this new exchanger design is being characterized and will be implemented on the interseasonal heat storage device.

CONCLUSIONS

An interseasonal heat storage system for dwelling heating applications and an associated experimental prototype is analyzed in this article. The article focuses on the falling films heat exchangers with vertical tubes, which exhibit very low performances compared to the desired ones. Operating conditions are characterized by very low falling film flow-rates per unit width compared to conventional absorption machines in order to guarantee high thermal efficiency. A numerical model was developed and validated using a dedicated experiment to describe the absorption/condensation and desorption/evaporation coupled processes in the system reactor. Simulations reproduce the evolutions of the different characteristic variables of the system (concentration and temperatures) and estimates the order of magnitude of the heat and mass transfer (i.e. the vapour mass flow, the solution cooling (heating), the solution dilution (concentration) and the heat transfer fluid heating (cooling) in absorption (desorption) operation modes, when considering a partial wetting rate of the falling films at the evaporator, the absorber and the desorber. The best agreement in charging and discharging modes are obtained with a wetting rate equal to 12 and 20%, respectively.

Different wettability tests were made on vertical metallic plates of brass and stainless steel using the LiBr solution and distilled water to confirm the wetting rate estimated with the model. The results indicate that, in all cases, the wetting rate estimated with the experiment agrees with the wetting rate deduced from modeling/experimentation comparisons for identical flow rates per unit width.

This study therefore highlights the critical influence of the falling films' wetting rate on the heat transfer using falling film exchangers. A new exchanger design involving grooved plates is proposed to ensure that the exchangers have a high wetting rate.

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NOMENCLATURE

D_i	Internal diameter, m
D	Mass diffusivity, $\text{m}^2.\text{s}^{-1}$
e	Plate thickness, m
g	Gravitational acceleration, m.s^{-2}
h	Enthalpy, J.kg^{-1}
h_p	Partial enthalpy, J.kg^{-1}
k	current segment
k_m	Convection mass transfer coefficient, m.s^{-1}
k_T	Heat transfer coefficient, $\text{W.m}^{-2}.\text{K}^{-1}$
L	Plate width wetted by the liquid film, m
L_{cap}	Capillary length, m
$\dot{m}$	Mass flow rate, kg.s^{-1}
$\dot{m}''$	Mass flux per unit surface, $\text{kg.s}^{-1}.\text{m}^{-2}$
n	Number of segments
Nu	Nusselt number
P	Pressure, Pa
Pr	Prandtl number
$\dot{q}''$	Heat flux, W.m^{-2}

Re	Reynolds number
S	Wetting rate
Sh	Sherwood number
T	Temperature, K
u	Velocity, m.s ⁻¹
x	Mass fraction, kg.kg ⁻¹
y	Distance to the plate, m
z	Distance along the plate, m

Greek symbols

δ	Film thickness, m
Δz	Segment height, m
λ	Thermal conductivity, W.m ⁻¹ .K ⁻¹
μ	dynamic viscosity, Pa.s
ρ	density, kg.m ⁻³
ξ	Non dimensional distance along the plate
σ	Surface tension, N.m ⁻¹

Subscripts

abs/des	absorbed or desorbed water
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<i>abs</i>	absorbed
<i>avg</i>	average
<i>eva/con</i>	Evaporated or condensed water
<i>exp</i>	experimental
<i>fp</i>	flute plate
<i>htf</i>	Heat transfer fluid
<i>H₂O</i>	Water
<i>i</i>	Inlet
<i>int</i>	Film interface
<i>k</i>	Segment k
<i>LiBr</i>	Lithium bromide
<i>o</i>	Outlet
<i>sat</i>	Saturated conditions
<i>sim</i>	simulation
<i>st</i>	LiBr solution
<i>vap</i>	Water vapor in the reactor
<i>w</i>	Metallic plate wall

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Table 1: Experimental inlet conditions considered for the LiBr falling film and heat exchangers on the desorber and condenser (charge mode)

Desorber	$\dot{m}_{st,i}$ [kg h ⁻¹]	$T_{st,i}$ [°C]	$x_{LiBr,i}$ [mLiBr/m _{st}]	$Re_{st,i}$	$\dot{m}_{htf}$ [kg h ⁻¹]	$T_{htf,i}$ [°C]
	[35 – 40]	[10 – 30]	[0.54 – 0.56]	[9-20]	720	90
Condenser	$\dot{m}_{H_2O,i}$ [kg h ⁻¹]				$\dot{m}_{htf}$ [kg h ⁻¹]	$T_{htf,i}$ [°C]
	0				360	20

Table 2: Experimental inlet conditions considered for the LiBr falling film, water film and heat exchangers on the absorber and evaporator (discharge mode)

Absorber	$\dot{m}_{st,i}$	$T_{st,i}$	$x_{LiBr,i}$	$Re_{st,i}$	$\dot{m}_{h_{tf}}$	$T_{h_{tf},i}$
	[kg h ⁻¹]	[°C]	[m _{LiBr} /m _{st}]		[kg h ⁻¹]	[°C]
	70	[24 – 26]	[0.55 – 0.54]	[28-30]	360	26
Evaporator	$\dot{m}_{H_2O,i}$	$T_{H_2O,i}$			$\dot{m}_{h_{tf}}$	$T_{h_{tf},i}$
	[kg h ⁻¹]	[°C]			[kg h ⁻¹]	[°C]
	20	15		42	720	20

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Fig. 1: Interseasonal absorption storage system principle. Charging mode (top) and discharging mode (bottom)

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$$T_{st,i} = 26^{\circ}\text{C} ; 78 < \text{Re} < 84; 0.54 < x_{\text{LiBr},i} < 0.59 / \text{HTF inlet conditions: } T_{htf,i} = 25^{\circ}\text{C} ;$$

$$\dot{m}_{htf} = 300 \text{ kg.h}^{-1} \text{ vapor pressure inside the reactor } P_{vap} = 13.5 \text{ mbar}$$

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Fig. 1: Interseasonal absorption storage system principle. Charging mode (top) and discharging mode (bottom).

Fig. 2: Scheme of the interseasonal absorption storage system

Fig. 3: Scheme of the falling film exchanger

Fig. 4: Evolution of the local Sherwood and Nusselt number along an isothermal plate with ξ

the non-dimensional distance along the plate, and $\text{Pr}_{st} = \frac{\mu_{st} C_{p_{st}}}{\lambda_{st}}$ the Prandtl Number of the

solution

$$x_{H_2O,i}$$

Fig. 5. Transfer modes considered across the metallic surface for partial wetting cases. View from the top. a) Optimistic case (1F); b) Pessimistic case (2F)

Fig. 6: Boundary conditions used for mass and energy balance within the control volume