

On the pressure drop in Plate Heat Exchangers used as desorbers in absorption chillers

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A B S T R A C T

The influence of the pressure drop in Plate Heat Exchangers (PHE) in the boiling temperature of LiBr-H₂O and NH₃-H₂O solutions is studied. For the NH₃-H₂O solution, the pressure drop-temperature-saturation relationship states that high pressure drops can be allowed in the solution with negligible changes in the saturation temperature, and in the PHE performance. Besides, in the case of the LiBr-H₂O solution, as the working pressure is usually very low, the analysis of the pressure drop must be taken as a main limiting parameter for the use of Plate Heat Exchangers as vapour generators. In this case, the pressure drop may considerably change the boiling temperature of the solution entering the heat exchanger and therefore a higher heating fluid temperature may be required. A guideline to design these systems is proposed.

1. Introduction

Absorption chillers, compared to mechanical refrigeration systems, are generally larger in size and that limits to a great extent their use in low and medium power applications. For instance a typical value for the volume to refrigeration power ratio in single effect absorption chillers is in the order of 0.04 m³/kW (without considering the volume occupied by the cooling system) for refrigeration capacities between 10 and 30 kW, whereas the mechanical compressor systems can have a ratio equal to 0.02m³/kW for the same range of refrigeration capacities. This is a clear inconvenient for the generalization of the absorption technology use, limiting its benefits in the contribution to the reduction in CO₂ emissions, specially in the case of the LiBr-H₂O solution as this system can be fed with low heat temperature sources as solar panels [1,2].

In order to avoid this large volume concern, attempts have been made to scale down their size. One way of reducing the size of these systems is the use of compact heat exchangers. Plate Heat Exchangers (PHE) have been used not only for the solution heat recover [3] but also for the absorber [4,5] or the generator [6]. The precautions when using this last possibility relative to the solution

pressure drop and its influence on the boiling temperature are studied here.

In the present work, we are interested in the relationship between pressure drop and saturation temperature, and its effect on the temperature profile in the PHE generator of an absorption chiller. This new consideration must be taken into account as a relatively important pressure drop may occur in a PHE used as the generator of an absorption system. This was not the case until now as in pool boiling the pressure drop is not appreciable. Even if some authors have experimentally measured these high pressure drop values [6], to the authors knowledge, the effect on the heat exchanger design and performance have never been studied. Both aspects are treated here.

2. Description of the absorption cycle, without pressure drop in the generator

In most of the cases, absorption systems are theoretically studied considering a negligible pressure drop in the heat exchangers [7]. The resulting ideal thermodynamic cycle can be easily solved in this case and as a result, knowing the evaporation and the condensation temperatures, the inlet and outlet temperatures of the flowing fluids in the different heat exchangers can be calculated. In the present study, a LiBr-H₂O and a NH₃-H₂O ideal cycles have been taken as reference cases, with an evaporation temperature

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Nomenclature

ch	chilled water
CW	cold water
D_h	hydraulic diameter
DP	pressure drop
f	friction factor
G	fluid mass flux
Hf	hot flow
L	channel length
\dot{m}	mass flow rate
Pc	condensation pressure
T	temperature
V	mean velocity
x	mass fraction
y	vapour mass fraction

Greek symbols

ρ	density
μ	dynamic viscosity
v_m	specific volume of the vapour liquid mixture

Subscripts

eq	equivalent
g	vapour
i	inlet
l	liquid
m	mean
o	outlet
REF	refrigerant
DIS	solution

equal to 7 °C and a condensation temperature equal to 40 °C. The procedure for the calculation is similar to that presented in de Vega et al. [8], and the corresponding equipment is depicted in Fig. 1. In the case of the NH₃-H₂O, the vapour separator should also include a rectifier for the final distillation of NH₃.

The resulting calculated temperatures of the solution at the inlet (7: t_{INLET}) and at the outlet (8: t_{OUTLET}) of the desorber, and the corresponding initial boiling temperatures at the given inlet concentration of the solution ($t_{BOILING}$) are presented in Figs. 2 and 3 for different vapour mass fractions of the refrigerant separated during the desorption. In these figures, the mass fraction y is the ratio between the refrigerant mass flux entering the condenser and the solution mass flux entering the desorber. In terms of the inlet and outlet concentrations, it can be expressed as:

$$y = \frac{\dot{m}_{REF}}{\dot{m}_{DIS}} \frac{x_7}{x_1} \frac{x_8}{x_8} \quad (1)$$

$$y = \frac{\dot{m}_{REF}}{\dot{m}_{DIS}} \frac{x_8}{x_8} \frac{x_7}{x_7} \quad (2)$$

where x_7 and x_8 are the NH₃ (1) or the LiBr (2) mass fractions.

These temperatures in the ideal cycle increase as the solution and the generated vapour are circulating through the heat exchan

ger during the desorption. In fact, for a given operating pressure of the generator, the initial temperature for the boiling condition is the same (for given evaporator and condenser temperatures), independent of the vapour mass fraction of the refrigerant separated y as shown in Figs. 2 and 3. In these figures, as well as in the rest of figures of the paper, the temperature profiles are just estimated, as in Gabriellii and Vamling [9]: the interesting point is that in all the cases, for the solution, $t_{INLET} < t_{BOILING} < t_{OUTLET}$. This allows in principle the use of a counter current flow PHE configuration for the heating fluid.

Without considering pressure drop, the maximum temperature in the cycle is the one at the exit of the generator and therefore the temperature of the heating fluid has to be in accordance with this value. In the case of the NH₃-H₂O this temperature can be larger than 100 °C for a mass vapour fraction separated of 16.6% and it could be around 95 °C for the LiBr-H₂O case with a mass vapour fraction separated in the order of 7%, which allows the use of solar collectors as the heating source.

3. Boiling temperature and pressure drop in PHE

The assumption of a negligible pressure drop is consistent with the use of pool boiling for the vapour generation. Nevertheless,

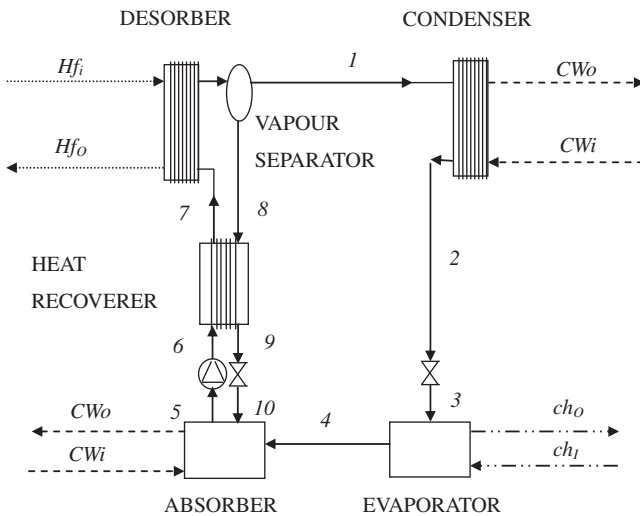


Fig. 1. Components of the absorption chiller.

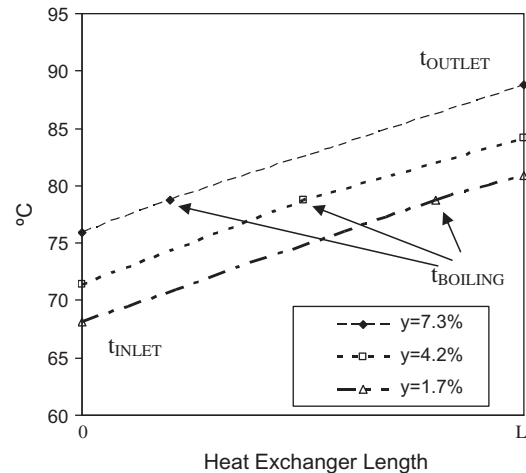


Fig. 2. Estimated temperature profiles along the desorber of a NH₃-H₂O cycle without pressure drop, for different vapour mass fractions of refrigerant: evaporation temperature = 7 °C; condensation temperature = 40 °C ($x_{i,NH_3} = 52.4\%$; $x_{out,NH_3} = 42.9\%$, 46.9% and 48.9%).

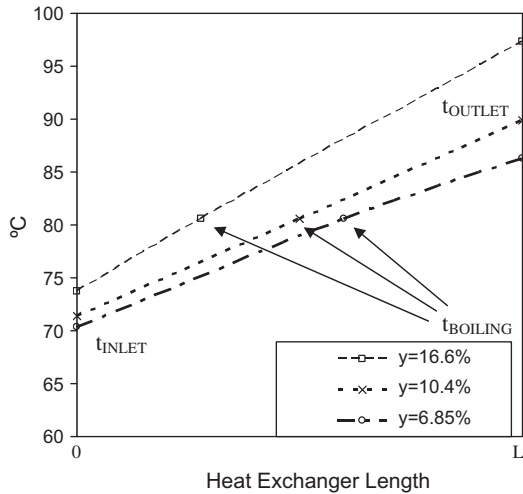


Fig. 3. Estimated temperature profiles along the desorber of a LiBr–H₂O cycle without pressure drop, for different vapour mass fractions of refrigerant: evaporation temperature = 7 °C; condensation temperature = 40 °C ($x_{inLiBr} = 56.9\%$; $x_{outLiBr} = 61.4\%$, 59.4% and 57.9%).

when considering the possibility of using a PHE as desorber, an estimation of the expected pressure drop must be performed, and the possible changes in the temperature needed for the separation of the refrigerant should be calculated. This temperature is higher than the one in the ideal case, as it will correspond to the saturation temperature at the generator pressure (that is, the condenser operating pressure) plus the pressure drop. Figs. 4 and 5 show the initial boiling temperatures that are necessary to make the solution boil for different solution concentrations, as a function of different possible values of the pressure drop. The outlet operating pressures are the same as in the ideal case: 7.4 kPa for the LiBr H₂O and 1.6 MPa for the NH₃ H₂O solution.

The sensitivity of the boiling temperature on the generator pressure drop is plotted in Figs. 4 and 5, showing how it is more important in the LiBr H₂O case, due to the low working pressure

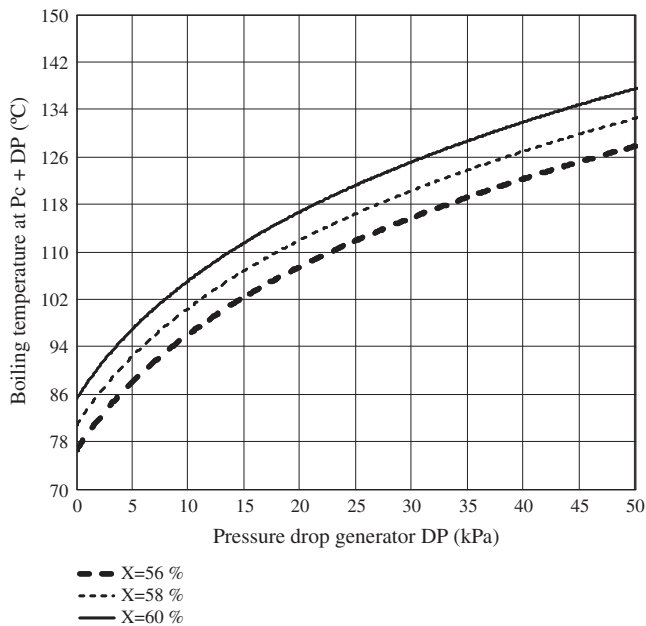


Fig. 4. Boiling temperature for a LiBr–H₂O solution for a generator operating at $P_c = 7.4$ kPa (condensation temperature of 40 °C) considering different values of the pressure drop DP and concentrations at the desorber inlet.

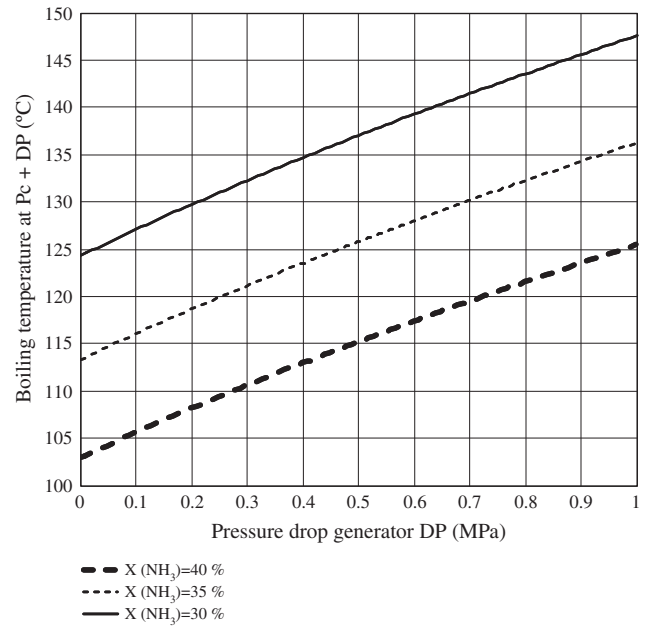


Fig. 5. Boiling temperature for a NH₃–H₂O solution for a generator operating at $P_c = 1.6$ MPa (Condensation temperature of 40 °C) considering different values of the pressure drop DP.

of the systems that use this solution. For pressure drops less than 10 kPa, an increase of 15 °C in the boiling temperature is expected. This is not the case of the NH₃ H₂O solution, where a 10% pressure drop implies a temperature increase of only 3 °C for the boiling process to take place.

We will try in the following to calculate what could be a reasonable value of the pressure drop in a conventional PHE used as generator in order to predict the real initial boiling temperature.

3.1. Lithium bromide water generator

We have considered an application of a chiller with a 5 kW of refrigeration capacity. We propose a design for the desorber using a PHE. We will focus first in the LiBr H₂O case. Initially, the design will be based on the conventional use of the inlet and outlet temperatures of the solution of Fig. 3 as design temperatures. For the present case, we have selected oil as the heating fluid as in Marcos et al. [6]. Therefore the design working conditions are: inlet solution temperature, 75 °C, outlet solution temperature 88 °C which correspond to a mass vapour fraction of refrigerant separated equal to 6.6%. According to Hewitt [10] the selected PHE for these design conditions is an Alfa Laval AC30 model, with 15 plates. The design aims at having a small sized heat exchanger (according to a small refrigeration capacity 5 kW) with the minimum number of plates for the desorber capacity required (6.4 kW). The size of the heat exchanger (325 mm (high), 93 mm (width) and 28.5 mm (depth)) is smaller than the size required for a similar application using a pool boiling configuration.

To the authors knowledge there is no correlation available in the literature for the calculation of the pressure drop in channels or tubes for the desorption of the refrigerant in LiBr H₂O or NH₃ H₂O solutions. Accordingly, a literature review has been done, relating to the process of boiling of different refrigerant mixtures, in order to estimate the order of magnitude of the pressure drop presumably encountered in a PHE when used in the desorption process in an absorber chiller [11–14].

The result of the literature review allows us to estimate this pressure drop considering mainly the influence of the frictional factor as:

$$DP = \frac{1}{2} f \frac{L}{v_m D_h} V^2 \quad (3)$$

where L is the channel length, D_h is the hydraulic diameter (which is twice the channel gap of the PHE considered), v_m is the specific volume of the vapour liquid mixture, V the mean velocity and f is the friction factor. According to Hsieh and Lin [14], the following correlation can be used for the phase change in mixtures:

$$f = 15250 \text{Re}_{eq}^{-1.25} \quad (4)$$

where Re_{eq} is the equivalent Reynolds number, defined as:

$$\text{Re}_{eq} = \frac{G_{eq} D_h}{\mu_l} \quad (5)$$

and

$$G_{eq} = G \left[1 + y_m + y_m \left(\frac{\rho_l}{\rho_g} \right)^{1/2} \right] \quad (6)$$

G_{eq} is an equivalent mass flux which is a function of the fluid mass flux (G), mean quality y_m and density at the saturated conditions.

It is worth noting that the employed correlation in [14] has been obtained for R 410A, and not for a LiBr–H₂O solution. Nevertheless, when the results are compared to the experimental values of Marcos et al. [6], a reasonable agreement is found and a difference lower than 15% is obtained.

The designed heat exchanger will work with a pressure drop in the solution size of 40.9 kPa (according to Eq. (3)), with a heating fluid inlet and outlet temperatures of 132 °C and 119 °C. It is interesting to note that the use of a small sized PHE may have a negative consequence on the possible use of a low temperature heating source, as the heat transfer surface lowers.

3.1.1. Temperature along the PHE

The desorber will now be studied taking into account the pressure drop and its influence on the temperature profile along the PHE. For the present case, a pressure drop of 40.9 kPa will require a higher boiling temperature, i.e. 122 °C according to Fig. 4.

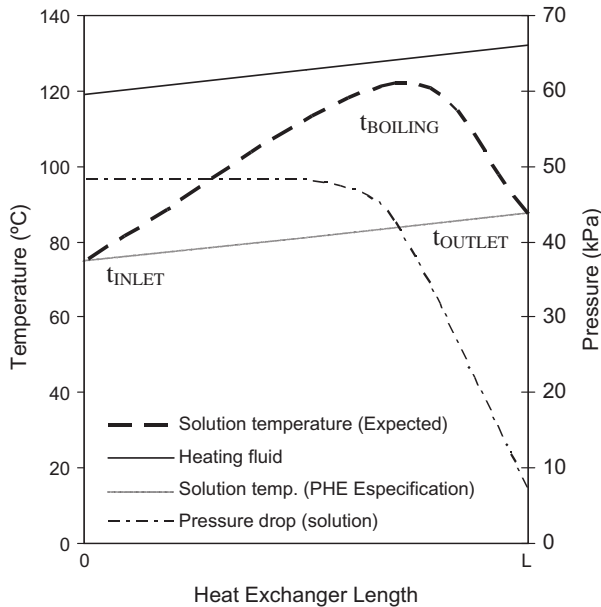


Fig. 6. Expected temperature profiles and pressure along the desorber of a LiBr–H₂O cycle with pressure drop: evaporation temperature = 7 °C; condensation temperature = 40 °C ($x_{inLiBr} = 56.9\%$; $x_{outLiBr} = 60.9\%$).

Therefore, for the design conditions (i.e., outlet temperature of the solution and calculated pressure drop), the new expected temperature profile is shown in Fig. 6. In this figure, the expected pressure drop is also shown: we have considered that during the single phase heat exchange, the pressure drop is considerably lower than in the two phase flow case [11,15,16].

The temperature profile in the solution side along the desorber changes drastically when compared to the ideal case, and $t_{INLET} < t_{OUTLET} < t_{BOILING}$. Therefore, if the PHE has to be able to transfer the designed 6.5 kW and to separate the required mass of refrigerant, a heating temperature higher than the one initially considered will be needed.

One way to avoid this performance problem is to design the PHE in order to limit the allowable pressure drop that will make $t_{BOILING} = t_{OUTLET}$. Fig. 7 shows the limiting pressure drop allowable in the generator for different operating conditions as a function of the mass vapour fraction separated. If the expected pressure drop is higher than the one represented in the Fig. 7, the temperature profile along the PHE will give a maximum not at the exit, but somewhere inside it, and therefore a design procedure for the counter cross flow configuration must be considered with care (this is explained in the next section). In addition, the temperature of the external heating fluid should be higher than in the ideal case.

3.2. Ammonia water generator

A similar procedure employed in the NH₃–H₂O case will provide a temperature profile in the desorber as shown in Fig. 8, for a pressure drop of 16 kPa.

In the case of the NH₃–H₂O system, the temperature profile in the PHE is similar to the ideal case, without pressure drop, and the sequence $t_{INLET} < t_{BOILING} < t_{OUTLET}$ still remains.

For the NH₃–H₂O solution, the pressure drop temperature saturation relationship (Fig. 5) established that high pressure drops can be allowed in the solution with negligible changes in the temperature profile (Fig. 8), i.e. this parameter is not a limiting factor for the use of PHE in the desorber.

Besides, in the case of the LiBr–H₂O solution (Fig. 4), as the working pressure is very low, the analysis of the pressure drop must be taken as a main limiting parameter for the use of PHE as an adequate heat exchanger in the vapour generator. In this case, the pressure drop may considerably change the boiling temperature of the solution entering the heat exchanger (Fig. 6).

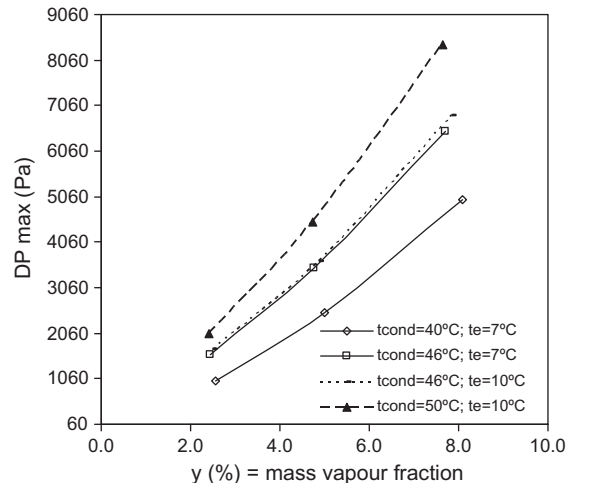


Fig. 7. Pressure drop that makes the initial boiling temperature equal to the outlet temperature in the desorber for LiBr–H₂O.

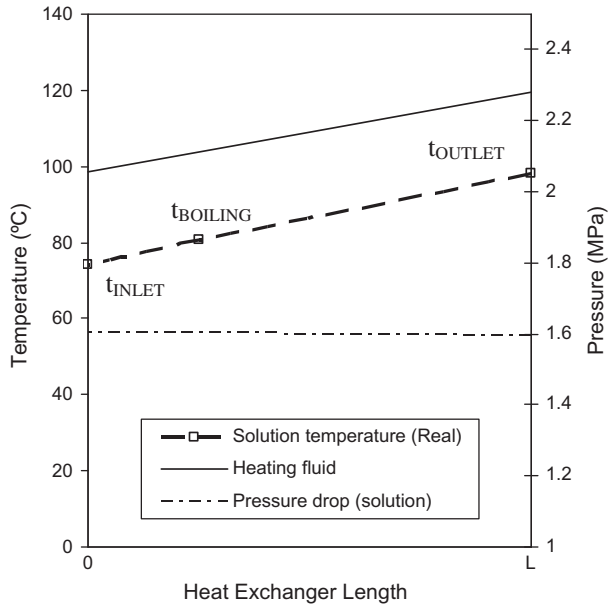


Fig. 8. Expected temperature profiles and pressure along the desorber of a $\text{NH}_3\text{-H}_2\text{O}$ cycle with pressure drop: evaporation temperature = 7 °C; condensation temperature = 40 °C ($x_{\text{inNH}_3} = 52.4\%$; $x_{\text{outNH}_3} = 42.9\%$).

3.3. Experimental considerations

As already stated, to the authors' knowledge, there is not enough literature information. In Marcos et al. [6], a double effect LiBr H_2O absorption chiller has been tested with an indirectly heated PHE used as high temperature desorber. In this application, the operating pressure of the PHE is 130 kPa. As in a previous section, we can calculate the change in boiling temperature, for different pressure drops (Fig. 9).

Comparing Fig. 9 with Fig. 4, it can be seen that, as the PHE is working at a higher pressure, the change in boiling temperature

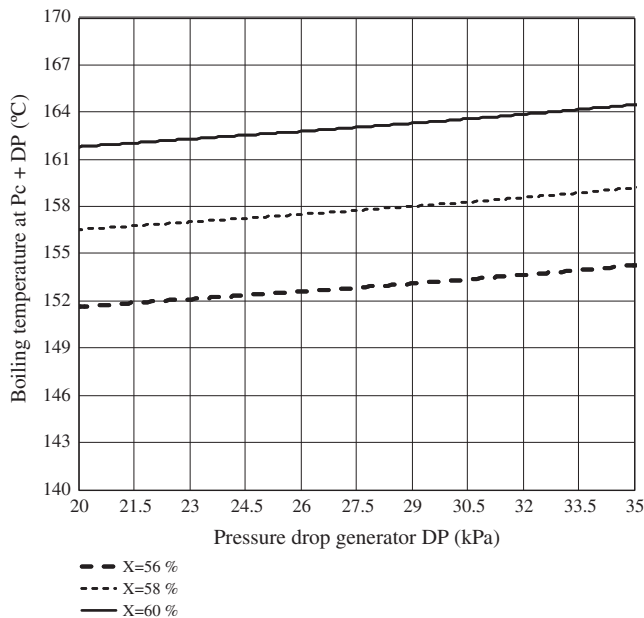


Fig. 9. Boiling temperature for a $\text{LiBr-H}_2\text{O}$ solution for a PHE generator operating at 130 kPa considering different values of the pressure drop DP and concentrations at the desorber inlet.

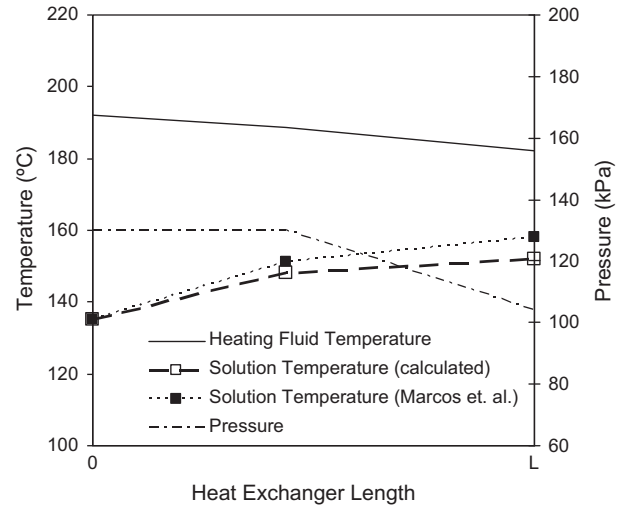


Fig. 10. Expected and measured temperature profiles and pressure along the desorber of a $\text{LiBr-H}_2\text{O}$ PHE with pressure drop: pressure inlet: 130 kPa; ($x_{\text{inLiBr}} = 56\%$; $x_{\text{outLiBr}} = 57.5\%$).

is around 5 °C for a pressure drop of 25 kPa. This is in agreement with the experimental results of Marcos et al. [6]. According to these authors, the employed PHE has 30 chevron type plates, 112 mm width and 250 mm length with a plate thickness of 0.5 mm and a distance between plates of 2.6 mm. The inlet and outlet temperatures of the solution are measured. The inlet and outlet temperatures of the heating fluid (oil) are also measured. Following a similar procedure as the one presented in Section 3.1., we have compared these measurements with the expected values of the solution along the PHE, taking into account the pressure drop. The results are presented in Fig. 10.

In this case, as the operating pressure in the PHE is the one corresponding to the high pressure desorber of a double effect cycle (130 kPa) it is considerably higher than in the generator of a single effect chiller (7.5 kPa). Therefore, the pressure drop influence is not a limiting parameter for the operation.

4. Plate Heat Exchangers used as generators

It seems that the reduction in size with the use of PHE as desorber may have negative consequences. Nevertheless, we suggest that the use of this kind of heat exchangers may be useful provided some care is taken in the design.

In order to reduce the pressure drop, one solution may be the use of a higher number of plates until the solution mass flow rate (and its velocity) will be low enough to reduce the pressure drop as required in Fig. 7. However, this option will lead simultaneously to a lower mass flow rate per channel in the hot side: while the high boiling heat transfer coefficient of the solution side will not considerably change at a lower mass flow rate, in the hot side, on the contrary, the lower the mass flow rate, the lower heat transfer coefficient and the higher heat transfer area (owing to a high number of plates) will not compensate the overall heat transfer coefficient decrease. Therefore with this configuration, in order to transfer the heat required between the hot and the solution side, the hot temperature maybe too high (which will limit the use of low temperature solar collectors, as heating sources). Therefore, a better option will be to increase the number of plates as required by the pressure drop limit, but minimizing the decrease in the heat transfer coefficient in the hot fluid side. The proposed design is then a multipass PHE.

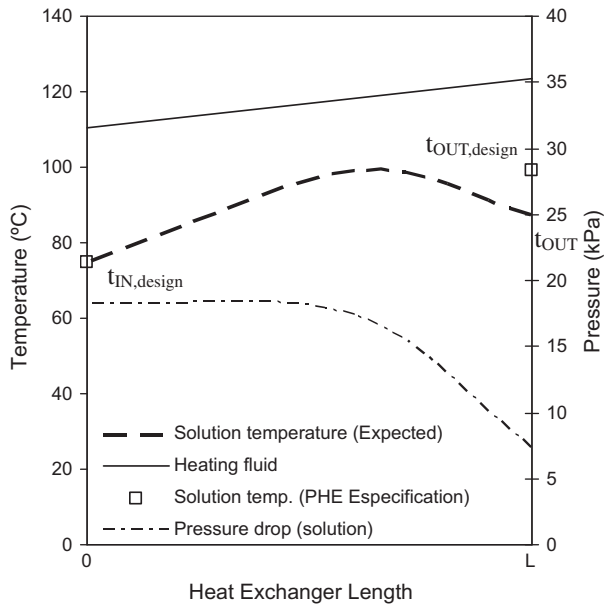


Fig. 11. Temperatures along the desorber of a LiBr–H₂O cycle with pressure drop: evaporation temperature = 7 °C; condensation temperature = 40 °C ($x_{in, LiBr} = 56.9\%$; $x_{out, LiBr} = 60.9\%$).

If the limit in the pressure drop in Fig. 7 is too restrictive, an other possibility for the design is to take the initial boiling temperature that results from the pressure drop in the generator as design temperature for the solution. In this case the temperature lift needed will be higher but the refrigerant separation will be guaranteed.

For our case, taking into account the first consideration, the suggested heat exchanger will be a multipass PHE, with two passes for the heating fluid and 1 pass for the solution [17]. With the same model (AC30) and this 2 1 configuration a 53 plate PHE will provide both, the heat transfer needed and the required quantity of refrigerant, and the final size will still be small (325 × 93 × 85.5 mm and 5.6 kg), According with the second condition design (the design temperature is not the outlet solution temperature but the real boiling temperature), the heat transfer and the refrigerant required in the cycle will be guaranteed.

The results in Fig. 11 show that the increase in the heat transfer surface compensates for the decrease in the heat transfer coefficient and therefore, which is also important and beneficial, the inlet hot temperature is lower. As a conclusion, the product of the overall heat transfer coefficient and the heat exchanger surface, UA is higher than in a 1 1 pass PHE and therefore the logarithmic mean temperature difference can be lower, for a given heating capacity. For our case, with the multipass configuration and 53 plates to reach a solution temperature of 99.3 °C (which corresponds to the boiling temperature) and a refrigerant mass fraction of 6.5%, the heating fluid will be at 123.4 °C less than the initial 132 °C of the one pass configuration. With this configuration, the pressure drop is lower than 11 kPa.

5. Conclusions

The influence of the pressure drop in the desorption of the refrigerant in NH₃ H₂O and LiBr H₂O solutions has been analyzed.

For the NH₃ case, as the working pressure is relatively high, the differences found in the desorption temperature without and with pressure drop during the desorption, are almost negligible. This is not the case for the LiBr H₂O solution as changes in desorption temperatures as high as 30 °C for pressure drops as low as 20 kPa can occur. Therefore, the pressure drop in the desorber is the key factor in the design of LiBr H₂O absorption chillers operating with PHE. The design temperature, which defines the thermal level of the hot fluid, may not be the outlet temperature of the solution, but the initial boiling temperature which is a direct function of the pressure drop. A limiting value of this pressure drop that gives the possibility of a counter current flow with the maximum temperature localized at the exit of the heat exchanger is also proposed as an alternative design guideline.

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References

- [1] Kim DS, Infante Ferreira CA. Air-cooled LiBr–water absorption chillers for solar air conditioning in extremely hot weathers. *Energy Convers Manage* 2009;50:1018–25.
- [2] Mateus T, Oliveira AC. Energy and economic analysis of an integrated solar absorption cooling and heating system in different building types and climates. *Appl Energy* 2009;86:949–57.
- [3] Jeong JY, Hong HK, Kim SK, Kang YT. Impact of plate design on the performance of welded type plate heat exchangers for sorption cycles. *Int J Refrig* 2009;32:705–11.
- [4] Vallès M, Bourouis M, Boer D, Coronas A. Absorption of organic fluid mixtures in plate heat exchangers. *Int J Therm Sci* 2003;42:85–94.
- [5] Cerezo J, Bourouis M, Vallès M, Coronas A, Best R. Experimental study of an ammonia–water bubble absorber using a plate heat exchanger for absorption refrigeration machines. *Appl Therm Eng* 2009;29:1005–11.
- [6] Marcos JD, Izquierdo M, Lizarte R, Palacios E, Infante Ferreira CA. Experimental boiling heat transfer coefficients in the high temperature generator of a double effect absorption machine for the lithium bromide/water mixture. *Int J Refrig* 2009;32:627–37.
- [7] Kang YT, Kunugi Y, Kashiwagi T. Review of advanced absorption cycles: performance improvement and temperature lift enhancement. *Int J Refrig* 2000;23:388–401.
- [8] de Vega M, Almendros-Ibanez JA, Ruiz G. Performance of a LiBr–water absorption chiller operating with plate heat exchangers. *Energy Convers Manage* 2006;47:3393–407.
- [9] Gabrieli C, Vamling L. Changes in optimal distribution of heat exchanger area between the evaporator and suction gas heat exchanger when replacing R22 with R407C. *Int J Refrig* 1998;21:440–51.
- [10] Hewitt GF, editor. *Thermal and hydraulic design of heat exchangers. Heat exchanger design handbook, Part 3*. New York: Begel House Inc.; 1998.
- [11] Wellsandt S, Vamling L. Heat transfer and pressure drop in a plate-type evaporator. *Int J Refrig* 2003;26:180–8.
- [12] Kreissig G, Müller-Steinhagen HM. Frictional pressure drop for gas/liquid two-phase flow in plate heat exchangers. *Heat Transfer Engineering* 1992;13:42–52.
- [13] Djordjevic E, Kabelac S. Flow boiling of R134a and ammonia in a plate heat exchanger. *Int J Heat Mass Transfer* 2008;51:6235–42.
- [14] Hsieh YY, Lin TF. Saturated flow boiling heat transfer and pressure drop of refrigerant R-410A in a vertical plate heat exchanger. *Int J Heat Mass Transfer* 2002;45:1033–44.
- [15] Manglik RM. *Plate heat exchangers for process industry applications: enhanced thermal-hydraulic characteristics of chevron plates, process, enhanced and multiphase heat transfer*. Begell House: New York; 1996 [p. 267–76].
- [16] Morales-Ruiz S, Rigola J, Pérez-Segarra CD, García-Valladares O. Numerical analysis of two-phase flow in condensers and evaporators with special emphasis on single-phase/two-phase transition zones. *Appl Therm Eng* 2009;29:1032–42.
- [17] Miura RY, Galeazzo FCC, Tadini CC, Gut JAW. The effect of flow arrangement on the pressure drop of plate heat exchangers. *Chem Eng Sci* 2008;63:5386–93.