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NUMERICAL EVALUATION FOR ABSORPTION CHILLER WORKING WITH WATER-AMMONIA BY COMPUTATIONAL FLUID DYNAMICS

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Abstract.*Absorption refrigeration systems have been gaining more space in companies and research groups because of their low power consumption, their main advantage over conventional systems. The present work has the objective of performing a numerical analysis in a flat plate type absorber with the flow of the absorbent solution and refrigerant vapor in direct contact and occurring from the top to the bottom surface of the absorber. The absorption capacity of the absorber will be evaluated by making use of the ammonia/water working pair. For this purpose, the CFX computational fluid dynamics software of the company ANSYS was used to solve the equations of mass, energy, momentum conservation, besides the correlations that describe the phenomena of heat and mass transfer.The parametric analysis was performed by varying the flow of refrigerant vapor. At the end of the study, the results showed that the increase of the refrigerant vapor flow by 10% led to an increase in the mass fraction of the solution by only 1%. The result also showed that a geometric optimization of the absorber is possible for the studied conditions.*

Keywords:*CFD, absorption chiller, ammonia-water*

1. INTRODUCTION

Due to climate changes increasingly proven by studies and observed by the general population, industries and researchers are seeking alternatives to reduce consumption and consequently emission of gases that cause damage to the planet. Among the alternatives, one of the most promising ones for reducing fossil fuel consumption is the use of cogeneration. Among the forms of cogeneration, it is worth mentioning the use of absorption refrigeration systems, which can be applied in the air-conditioning, at temperatures above 0ºC (using the pair of lithium bromide and water) or can be used at temperatures below 0ºC (making use of ammonia and water).

The absorption chiller differentiates from the compression chiller by replacing the compressor with a thermal unit compound of generator, absorber and solution pump. Due to the number of additional equipment for the same purpose, absorption chillers have larger dimensions and higher initial costs than conventional systems, in addition to having a lower thermal COP. These are some of the causes for the reduced use of the absorption chiller, representing approximately 1% of units sold (TRICHÉ et al., 2016).

As far as the difference between the systems is concerned, the absorption chillers operate with two or more working fluids and use thermal energy, unlike the steam compression systems that make use of electric energy as the source of the system's drive. It is precisely by operating with thermal energy the main advantage of absorption refrigeration systems. Them can operate not only with thermal rejects and solar energy but also by the burning of fossil fuels. Another advantage is due to the absence of moving parts, providing a lower maintenance cost (OCHOA, 2010).

To work around the equipment'ssize problem due to the size of the equipment, studies are necessary to understand and improve the performance of the equipment used. Since the heat pump and the heat exchanger are equipment applied to different types of systems, they are well developed and do not require further studies. Therefore, in order to achieve better performance, smaller equipment size and lower cost, studies aimed at the development of the generator and the absorber are necessary, which are crucial equipment for the system's operation. According to Sultana (2006), the performance of these systems is linked to the absorption capacity of the refrigerant.

This work aims to perform a global analysis of a flat plate absorber using ANSYS CFX, analyzing the impact of the flow of absorbent solution and refrigerant vapor on the absorption capacity of the absorber.

2. FORMULATION

Due to the importance of the flat plate absorber for reducing the size and cost of absorption refrigeration systems (CEREZO et al., 2009) and the lack of studies using this type of absorber (MICHEL; PIERRÈS; STUTZ, 2017), this work will use a flat plate absorber as this leads to a reduction in equipment dimensions and costs.

The analyzed absorber consists of a parallel flat plate absorber with the flow of absorbent solution and refrigerant vapor in direct contact and occurring from the top to the bottom of the absorber, while the cooling fluid flows counter-currently to the absorbent-refrigerant fluid and separate by a wall. The cooling fluid is inserted into the bubble system with a constant diameter, while the absorbent solution and the cooling fluid are in liquid form.Table 2 shows the geometric characteristics of the absorber and figure 1 shows de flow direction of the absorber.

Table 1. Geometric characteristics of the absorber.

Length of absorber	300 mm
Width of the mixing channel	4.0 mm
Width of the cooling channel	1.5 mm

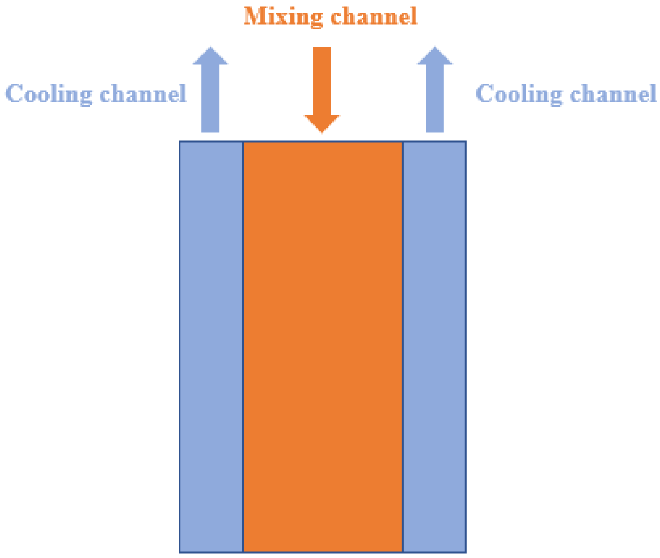

Figure 1. Flow direction of the absorber.

Aiming at reducing computational stresses, the thickness of the wall that separates the refrigerant-absorbent mixture from the cooling fluid was omitted. This simplification was justified by the wall thickness of less than 1mm, offering no resistance to conduction heat transfer. The injector was replaced by a flow distributed along the inlet of the absorber.

2.1 MATHEMATICAL MODELING

The equations of conservation of mass, energy and momentum were applied to analyze the phenomena of heat and, mass transfer. For this, some simplifying hypotheses were assumed.

- Two-dimensional analysis;
- Steam flow distributed at the inlet of the absorber;
- Uniformly distributed refrigerant vapor and absorbent solution;
- Steady state;
- Zero resistance to a transfer of heat and mass to the refrigerant fluid.

2.2 MASS CONSERVATIONEQUATION

The equation of mass conservation, when applied the imposed simplifications, is presented as the following form:

$$\nabla(\rho\vec{v}) = S_m \tag{1}$$

Where ρ is the density, v is the velocity and S_m is the source term, which represents the amount of mass transferred between the phases and is obtained by Eq.(2)

$$S_m = \dot{m}_{p,q} = k_{p,q}A_i\left(\rho_{q,e}^i - \rho_q^i\right) \tag{2}$$

Where A_i is the interface area between the phases, $\rho_{q,e}^i$ is the equilibrium mass balance of ammonia in the liquid phase, obtained by Eq.(4), ρ_q^i is the mass concentration of ammonia in the liquid phase and$k_{p,q}$ is the mass transfer coefficient between the phases given by Eq.(3).

$$k_{p,q} = \frac{Sh_q D_q}{d_p} \tag{3}$$

D_p is the diffusivity of the continuous phase, d_p is the bubble diameter of the dispersed phase andSh_q is the Sherwood number of the continuous phase obtained by Eq. (7)

$$\rho_{q,e}^i = K_{q,p}^{\rho}\rho_{q,e}^j \tag{4}$$

Where$\rho_{q,e}^j$ is the equilibrium of the water mass concentration in the liquid phase and$K_{q,p}^{\rho}$ can be written as shown in Eq.(5):

$$K_{q,p}^{\rho} = \frac{\rho_q}{\rho_p}K_{q,p}^j \tag{5}$$

Where ρ_q and ρ_q are the concentrations of the species of interest in the vapor and liquid phases respectively and $K_{q,p}^y$ is obtained by Eq. (6), its production process being described in Lima et al. (2019).

$$K_{q,p}^y = -4.814\cdot 10^{-7}(-1.095(T_L)^2 + 744.91(T_L) - 126307)^2 + \\ + 3.827\cdot 10^{-5}(-1.095(T_L)^2 + 744.91(T_L) - 126307) + 0.452 \tag{6}$$

$$Sh = \begin{cases} 2 + 0.6Re_p^{1/2}Pr^{1/3},\ 0 \le Re_p < 776.06 & 0 \le Pr < 250 \\ 2 + 0.27Re_p^{0.62}Pr^{1/3},\ 776.06 \le Re_p & 0 \le Pr < 250 \end{cases} \tag{7}$$

2.3 EQUATION OF THE CONSERVATION OF THE AMOUNT OF MOVEMENT

Applying the hypotheses, the equation of conservation of the momentum takes the following form:

$$\nabla(\rho\vec{v}\vec{v}) = -\nabla p + \rho\vec{g} + \vec{F} \tag{8}$$

Were the first term in the left side of the equality represents the variation of the amount movement, p is the static pressure, $\rho\vec{g}$ is the gravitational force and F are the external forces acting on the body.

2.4 ENERGY CONSERVATION EQUATION

Applying the simplifications, the general form of the energy conservation equation is given:

$$\nabla\left(\vec{v}(\rho E + p)\right) = \nabla\left(k\nabla T - \textstyle\sum_j h_j J_j\right) + S_h \tag{9}$$

Where the left-hand term represents convection heat transfer, E is the fluid energy given by equation 10 and h is the sensible enthalpy given by Eq.(11). The first term on the right-hand side is heat diffusion, and the second term is the interdiffusion.

$$E = h + \frac{p}{\rho}\frac{V^2}{2} \tag{10}$$

$$h = \sum_j Y_j h_j - \frac{p}{\rho} \tag{11}$$

Sh represents the heat absorbed by the liquid phase at the interface, obtained by Eq.(12)

$$S_h = Q_{pq} = \bar{h}_{pq} A (T_V - T_L) \tag{12}$$

Where the coefficient of heat transfer can be obtained through Eq.(13)

$$\bar{h}_{pq} = \frac{k_p Nu_p}{d_p} \tag{13}$$

The Nusselt number obtained by equation 14

$$Nu_p = \begin{cases} 2 + 0.6 Re_p^{1/2} Pr^{1/3}, \ 0 \leq Re_p < 776.06 & 0 \leq Pr < 250 \\ 2 + 0.27 Re_p^{0.62} Pr^{1/3}, \ 776.06 \leq Re_p & 0 \leq Pr < 250 \end{cases} \tag{14}$$

3. RESULTS

For the following analyzes, fixed values were adopted at the inlet of the absorber for the temperature of the absorbent solution, refrigerant vapor, it is also fixed the flow and temperature of coolant fluid. The values adopted for these variables are presented in table 2. The other data as they are varied during the analysis are presented during the discussion.

Table 2. Input data of the absorber.

Refrigerant vapor temperature	301.75 K
Absorbent solution temperature	338.95 K
Coolant fluidtemperature	319.95 K
Coolant fluid flow	1140 mg/s
Absorbent solution flow	87.5 mg/s

Figure 2 shows the behavior of the ammonia mass fraction in the absorbent solution when the flow of refrigerant varies. The mixture enters the left at the point marked by 300 mm. Refrigerant vapor flows of 23.75 mg/s, 26.25 mg/s, and 28.75 mg/s were used, while the flow rate of absorbent solution was maintained at 87.5 mg/s. It can be observed that with increasing refrigerant vapor flow there is an increase in ammonia concentration in the absorbent solution, as expected, due to the increase in ammonia concentration in the fluid, shifting the equilibrium curve to higher values.

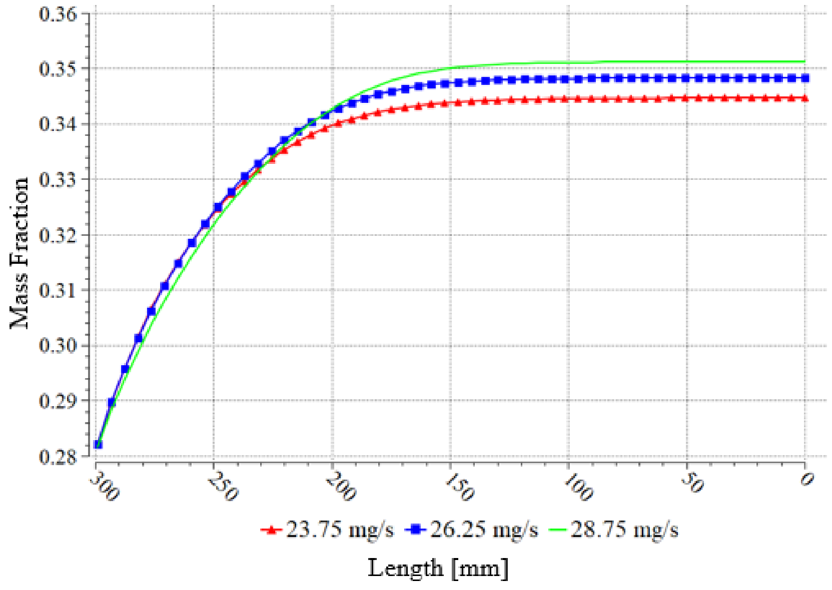

Figure 2. Impact of the refrigerant flow on the mass fraction of ammonia in the absorbent solution

This increase observed in the quantitative form occurs in such a way that, for an increase of 23.75 mg/s to 26.25 mg/s (an increase of 10%) in the flow of refrigerant vapor it is observed an increase of than 1.16% in the percentage of ammonia in the absorbent solution. While the increase from 26.25mg/s to 28.75mg/s (an increase of 9.5%) in the flow of refrigerant leads to an increase of 0.85%. It is also possible to observe that the mass transfer phenomenon does not require the total length of the absorber to get into equilibrium, being necessary for these cases only 250mm.

Figure 3 shows the behavior of the ammonia mass fraction when the flow of refrigerant vapor at the inlet of the absorber varied. In this case, a mass fraction of ammonia present in the solution equal to 0.308 (10% more than in the previous case presented in Fig. 1) was adopted. The other data remained constant, with an absorbent solution flow equal to 87.5 mg/s and it was adopted refrigerant vapor flows equal to 23.75 mg/s, 26.25 mg/s, and 28.75 mg/s. As in the previous case and as was also expected, there was a gain in the mass fraction of ammonia in the absorbent solution due to the increase of the vapor flow, for the same reason previously discussed.However, the increase in ammonia present in the solution when the flow rate was changedfrom 23.75 mg/s to 26.25 mg/s (increasement of 10%) was 1.10%. When the flow rate was passedfrom 26.25 mg/s to 28.75 mg/s (increasement of 9.5%), the gain obtained in the mass fraction of ammonia present in the solution was 0.80%. In comparison with the data obtained by Fig. 2, it is possible to contact that there is a reduction in the increase of the mass fraction of ammonia in the solution of approximately 0.05% when gainin 10% (from 23.75 mg/s to 26.25 mg/s) the fraction mass of ammonia present in the absorbent solution, as well as when the increase of 9.5% (26.25mg/s to 28.75mg/s) is performed. From this observation, it can be concluded that the riseof the mass fraction of ammonia in the absorbent solution does not affect the percentage gainof ammonia due to the increase in the flow of refrigerant vapor.

Comparing Fig. 2 and Fig. 3 it is also possible to observe that the increase of the mass fraction of the absorbent solution leads to a reduction of the absorption capacity of the absorber when compared to the same flow rate. Thus, for the 10% increase in the mass fraction of ammonia in the absorbent solution inlet there is a reduction of approximately 20% of the absorption capacity when using the refrigerant flow rateof 23.75 mg/s. Thus, although the increase of the mass fraction leads to small differences in the absorber behavior (the percentage increase of ammonia remains approximately constant with the increase of the flow), its impact on the efficiency of the system must be considered, since the efficiency of the refrigeration system by absorption is associated with its absorption capacity.

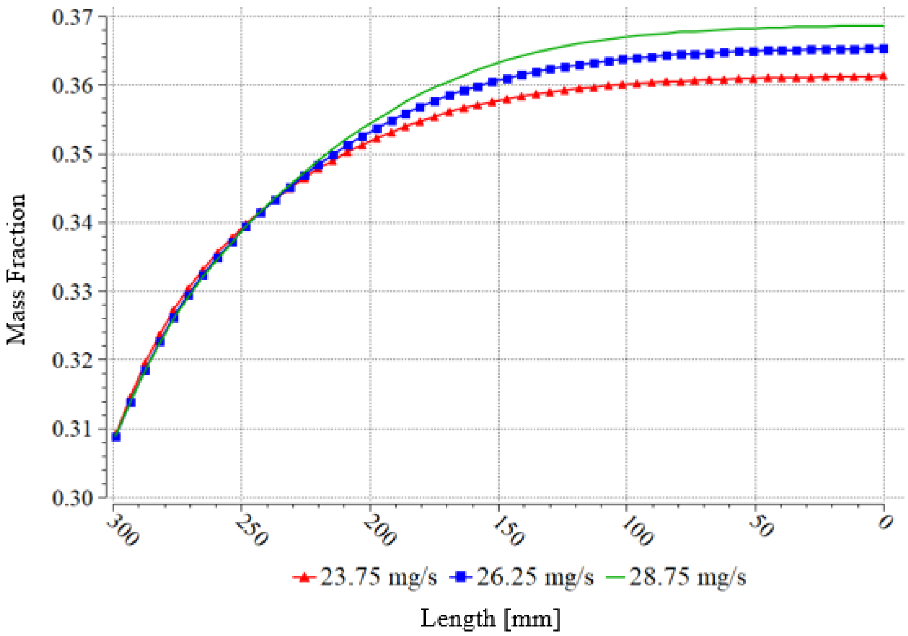

Figure 3. Impact of the refrigerant flow on the mass fraction of ammonia in the absorbent solution with mass fraction of 0.308

Figure 4 shows the behavior of the ammonia mass fraction when the average bubble diameter of the refrigerant vapor at the inlet of the absorber is varied. For this, three different bubble diameters were analyzed, 0.5 mm, 1.0 mm and 2 mm. Refrigerant vapor flow rate was adopted as 25.0 mg/s, while the flow rate of the absorbent solution was adopted as 87.5 mg/s, the system mass fraction was 0.28. It is possible to observe that the increase of the bubble diameter leads to a delay in the mass transfer process of the system, since bubbles with smaller diameters lead to larger areas of heat and mass exchange, increasing the mass transfer rate. Quantitatively, when a bubble diameter increases from 1 mm to 2 mm, at the absorber inlet, it’s required an absorber 200 mm longer for the bubble diameter of 2 mm compared to the 1 mm bubble. When increased from 0.5 mm to 1 mm the increase in the absorber's length required for the complete transfer of mass becomes 75 mm.

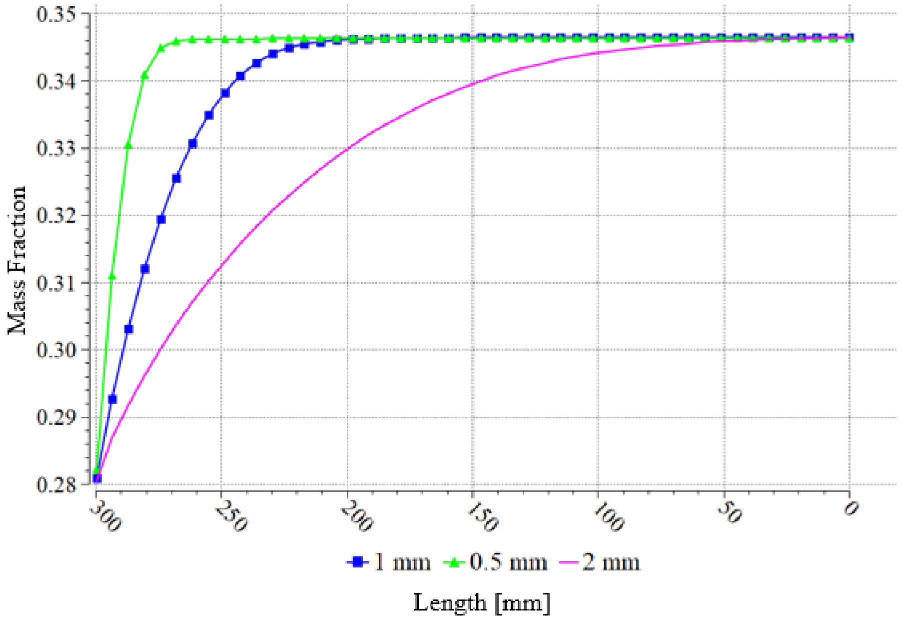

Figure 4. Impactof the bubble diameter on the mass fraction of ammonia in the absorbent solution

4. CONCLUSION

- It can be concluded from the results presented that the increase of the refrigerant vapor flow leads to a non-linear increase of the ammonia present in the absorbent solution, being obtained a 1% increase in the mass fraction for a 10% increase in the flow of refrigerant vapor.
- It is possible to conclude that the length of the absorber is overestimated and can be optimized.
- It is also possible to conclude that for the mass fraction values discussed in this paper, a 10% increase in its value leads only to a 0.05% difference in the increase in the mass fraction due to the increase in the refrigerant vapor flow.
- However, increasing the mass fraction leads to a reduction in the absorption capacity of the absorber. When the lowest refrigerant vapor flow rate is used for the two mass fractions, a loss of 20% in the absorption capacity is obtained.
- Finally, it is possible to conclude that the increase of the bubble diameter in the flow leads to a delay of the heat and mass transfer process, and larger equipment is necessary to complete the whole process.

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