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Numerical Simulation of Vertical Plate Absorber

Thanh-Tong Phan, Manirul Alam Sarker, Choon-Geun Moon, Jung-In Yoon*
Graduate School, Pukyong National University, Korea
Pukyong National University, Korea*

1. INTRODUCTION

Absorption cooling systems have been widely used for the cooling of large buildings since they can reduce electric peak load during summer time and do not use CFCs which is the main cause of ozone depletion of the stratosphere and global warming. Among major components of an absorption chiller/heater, the absorber, which has a direct effect on efficiency, size, manufacturing and operating cost of the system, is least understood[1]. To achieve a compact and efficient absorber, plate absorber is used to replace conventional shell and tube absorber.

Most of previous numerical studies on absorption process in plate absorber, such as the studies of G. Grossman et al.[1], Kawae et al.[2], Kim et al.[3], and so on, were performed on the assumption that either wall temperature was uniform or plate wall was cooled by air. The information of model of absorption process in plate absorber, which was cooled by cooling water, has been lacked.

The present study focuses on the numerical study on the combined heat and mass transfer process in absorption of water refrigerant vapor into a LiBr solution of vertical plate absorber, which was cooled by cooling water. The temperature and concentration profiles across the film thickness and along the vertical plate wall as well as the effect of Reynolds number on them were researched. Besides, the variations of the absorption heat and mass fluxes, the total heat and mass transfer rates and the heat and mass transfer coefficients along the vertical plate wall have been investigated. Especially, the selected operating condition was expected to be present in an actual absorption chiller/heater system.

2. ANALYSIS MODELS AND GOVERNING EQUATION

The numerical analysis for the absorption process in the present study is described schematically in figure 1. A film of LiBr solution composed of LiBr (absorbent) and H_2O (refrigerant) flows down over a vertical plate. The film is in contact with stagnant refrigerant vapor and cooled by cooling water flowed inside the plate.

The following assumption have been made:

(1) The flow of the liquid film is laminar, and fully developed throughout.
(2) The liquid is Newtonian and steady state.
(3) No shear forces are exerted on the liquid by the vapor at the interface.
(4) There is no heat transfer in the vapor phase.
(5) The system pressure is constant.
(6) Thermodynamic equilibrium exists at the interface.
(7) The thermal resistance of the plate wall has been neglected.
(8) The cooling water is flowed counter to the flow of LiBr solution and its temperature is changed linearly.

The simultaneous heat and mass transfer in the system described by the following

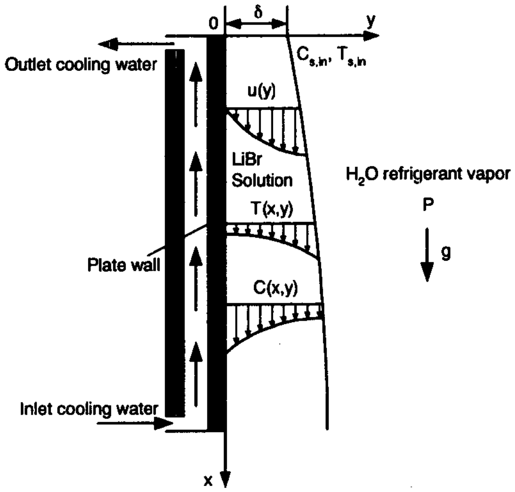

Fig. 1 Schematic diagram of absorption process in vertical plate absorber cooled by water

equations

Because of constant film thickness the cross-stream velocity is neglected ($v=0$), the downstream velocity u profile shown in figure 1 is parabolic as a function of y

$$u(y)=\frac{3}{2}\,u_{mean}\left[2\left(\frac{y}{\delta}\right)-\left(\frac{y}{\delta}\right)^2\right] \tag{1}$$

where $\quad u_{mean}=\frac{\Gamma}{\rho\delta} \qquad \delta=\left(\frac{3\Gamma\mu}{\rho^2 g}\right)^{1/3} \qquad Re_f=\frac{4\Gamma}{\mu}$

The energy equation written in the two-dimensional rectangular film coordinates is

$$u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}=\alpha\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right)+\left(\frac{\partial^2 C}{\partial y^2}T+\frac{\partial C}{\partial y}\cdot\frac{\partial T}{\partial y}\right)\frac{D(c_{p,LiBr}-c_{p,H_2O})}{c_p} \tag{2}$$

The diffusion equation is

$$u\frac{\partial C}{\partial x}+v\frac{\partial C}{\partial y}=D\left(\frac{\partial^2 C}{\partial x^2}+\frac{\partial^2 C}{\partial y^2}\right) \tag{3}$$

The following boundary conditions are applied

At solution inlet : $x=0 \quad T=T_{s,in} \quad C=C_{s,in}$ (4)

At solution outlet : $x=L \quad \frac{\partial T}{\partial x}=0 \quad \frac{\partial C}{\partial x}=0$ (5)

At plate wall : $y=0 \quad T=T_{wall} \quad \frac{\partial C}{\partial y}=0$ (6)

At liquid-vapor interface : $y=\delta \quad C_{surf}=C_{surf}(T_{surf},P)$ (7)

The refrigerant mass flux and the total mass transfer rate absorbed to the liquid film at the interface are respectively determined as follows

$$\dot{m}_{surf}=-\rho D\left(\frac{\partial C}{\partial y}\right)_{y=\delta} \qquad M_{surf}=\int_0^L \dot{m}_{surf}(x)dx \tag{8}$$

The absorption heat flux generated at the interface is

Table 1 Normal operating condition of plate absorber

Parameters	Values	
	For comparison [2,3]	Present study
Plate height L [m]	10	1
System pressure P [kPa]	1	1
Inlet solution temperature $T_{s,in}$ [°C]	46.5	46
Inlet solution concentration $C_{s,in}$ [wt%]	60	60.2
Wall temperature T_{wall} [°C]	35	--
Inlet cooling water temperature $T_{c,in}$ [°C]	--	32
Outlet cooling water temperature $T_{c,out}$ [°C]	--	36
Film thickness δ [mm]	0.2	0.2197
Film Reynolds number Re_f [--]	10.6	14
Film mass flow rate Γ [kg/m · s]	0.0142	0.0187

$$\dot{q}_{surf} = k\left(\frac{\partial T}{\partial y}\right)_{y=\delta} - (c_{p,H_2O} - c_{p,LiBr}) T \rho D \left(\frac{\partial C}{\partial y}\right)_{y=\delta} = H_{abs} \cdot \dot{m}_{surf} \tag{9}$$

where H_{abs} is the heat of absorption per unit mass of absorption solution

$$H_{abs} = H_{latent} + H_{dilution} \tag{10}$$

3. METHOD OF NUMERICAL ANALYSIS

In the present analysis, the calculation area was established only in liquid film of absorption solution. The integral forms of conservation equations were solved by the finite volume method proposed by Patankar [5]. The power-law scheme was used to treat the convection-diffusion term and get the discretization equations. The TDMA method (Tri-Diagonal Matrix Algorithm) was applied line by line to solve the system of discretization equations. In the arrangement of the control volume, the uniform grid coordinate of 302 pieces by x-direction and 32 pieces by y-direction was performed.

4. VALIDITY OF THE MODEL

In figure 2, the interface concentration profiles along the plate wall were compared with those of Kawae et al[2] and Kim et al.[3] under the same operating condition shown in table 1 such as the wall temperature is constant and equal to 35°C to test the validity of the present model. It can be seen that the interface temperature and concentration of the present model agree very well with the above previous models.

5. RESULTS AND DISCUSSIONS

The present study was performed under similar to operating condition of an actual absorption chiller/heater with considering the LiBr solution is in a thermodynamic equilibrium state at the inlet was shown in table 1.

5-1. Temperature and concentration distributions

Figure 3 presents the temperature distribution along the plate wall by the change of film Reynolds number. The interface temperature decreases rapidly at the inlet range but

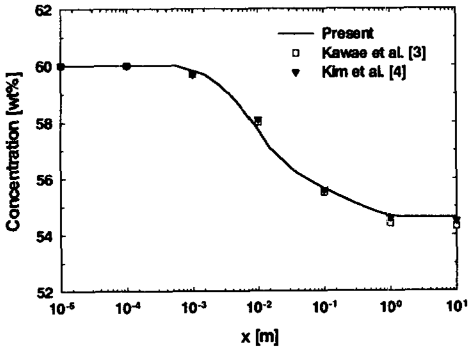

Fig. 2 Comparison of interface concentration of present study with previous investigations

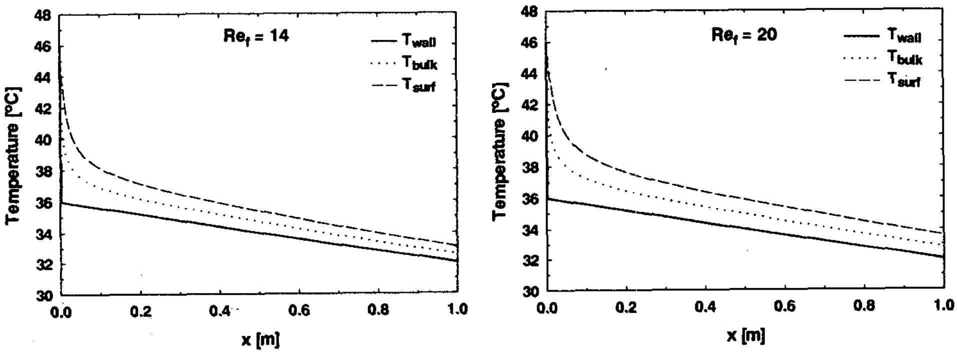

Fig. 3 Temperature profiles along the plate wall

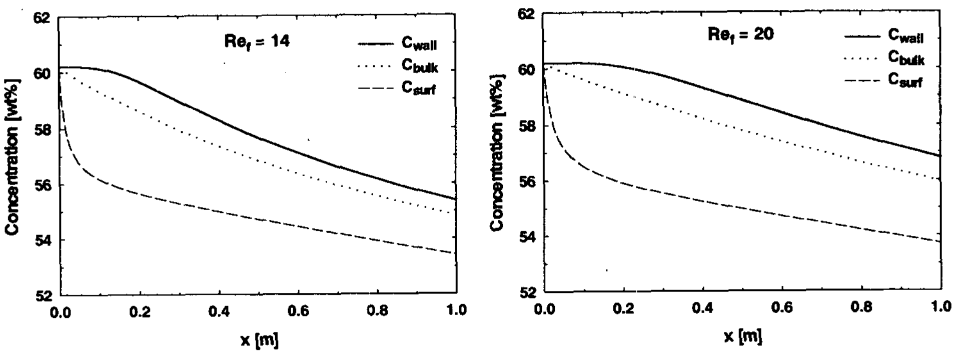

Fig. 4 Concentration profiles along the plate wall

slowly as x increases since the temperature difference between the film and the wall is decreased as x increases.

In the case of small film Reynolds number, the liquid film thickness is thinner. Consequently, the thermal resistance of the liquid film is smaller hence the interface temperature and the temperature difference between the interface and the wall are

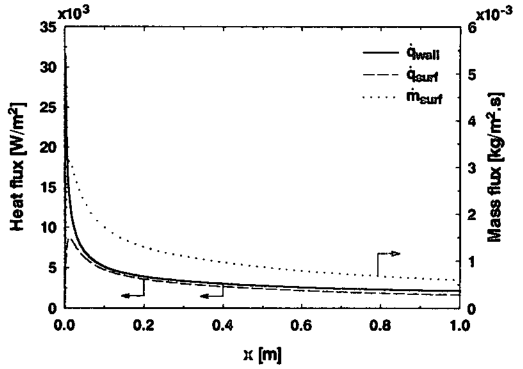

Fig. 5 Variation of local heat and mass fluxes along the plate wall

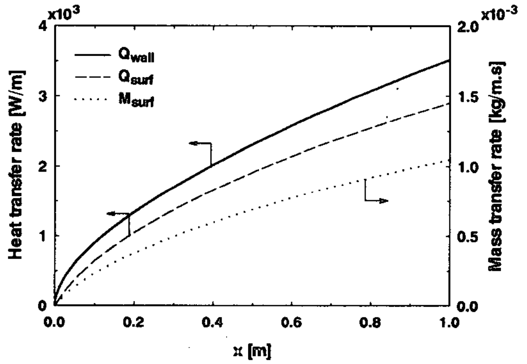

Fig. 6 Variation of total heat and mass transfer rates along the plate wall

decreased.

Figure 4 presents the concentration distribution along the plate wall by the change of film Reynolds number. The interface concentration is rapidly decreased at the inlet range but slowly as x increases.

In the case of small film Reynolds number, solution mass flow rate is decreased therefore absorption capacity is decreased following. Consequently, the interface concentration and the concentration difference between the interface and the wall are decreased. However, at the inlet range, the refrigerant vapor has not reached to the wall yet, with the result that the wall concentration remains unchanged whereas the interface concentration is decreased. Hence the concentration difference between the interface and the wall is increased at the inlet range as Reynolds number decreases.

5-2. Variation of absorption heat and mass transfer rates

Figure 5 shows the variation of local heat and mass fluxes along the plate wall. The local heat flux at plate wall is determined as follows

$$\dot{q}_{wall} = k\left(\frac{\partial T}{\partial y}\right)_{y=0} \tag{11}$$

The heat and mass fluxes at the interface increase rapidly at the inlet range and reach

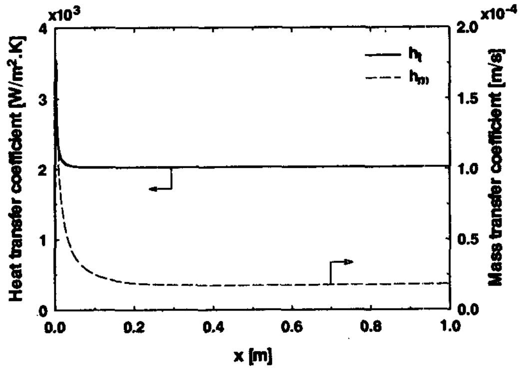

Fig. 7 Variation of local heat and mass transfer coefficients along the plate wall

their maximum values $\dot{q}_{surf,\max} = 8.7336 \times 10^{3}\ W/m^{2}$ and $\dot{m}_{surf,\max} = 3.1499 \times 10^{-3}\ kg/m^{2} \cdot s$ respectively at the distance of $x = 11.7 \times 10^{-3} m$. Then they decrease rapidly at near their maximum values but slowly as x increases.

The heat and mass fluxes are monotonically decreased toward to their critical values as x increases. Those are $\dot{q}_{surf,\infty} = \dot{m}_{surf,\infty} = 0$ as x approaches infinite $x = \infty$.

Figure 6 presents the variation of total heat and mass transfer rates along the plate wall. The total heat and mass transfer rates are rapidly increased at the inlet range but slowly as x increases.

5-3. Variation of heat and mass transfer coefficients

Figure 7 shows the variation of local heat and mass transfer coefficients. The local heat and mass transfer coefficients are determined as follows

$$h_t = \frac{k\left(\frac{\partial T}{\partial y}\right)_{y=0}}{T(\delta) - T(0)} = \frac{\dot{q}_{wall}}{T_{surf} - T_{wall}} \tag{12}$$

$$h_m = \frac{-D\left(\frac{\partial C}{\partial y}\right)_{y=\delta}}{C(0) - C(\delta)} = \frac{\dot{m}_{surf}}{\rho(C_{wall} - C_{surf})} \tag{13}$$

where h_t and h_m are the local heat transfer coefficient from the liquid to the wall and the local mass transfer coefficient from the interface to the liquid film respectively.

The heat and mass transfer coefficient are zero at the inlet $(x = 0)$ and get the highest value as x begins to increase from the inlet. Those are $h_{t,\max} = 3.5386 \times 10^{3}\ W/m^{2} \cdot K$ and $h_{m,\max} = 1.7527 \times 10^{-4} m/s$ respectively at the distance of $x = 1.67 \times 10^{-3} m$. Then they are rapidly decreased at the inlet range but slowly and become steady as x increases since the mass flux and the concentration difference between the interface and the wall are rapidly changed at the inlet range but slowly as x increases.

5. Acknowledgement

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6. CONCLUSIONS

The results obtained can be summarized as follows:

The interface temperature and concentration are rapidly decreased at the inlet range but slowly as x increases. In the case of small film Reynolds number, the interface temperature and concentration decrease.

The absorption heat and mass fluxes are rapidly increased at the inlet range and reached their maximum values $\dot{q}_{surf,\max}=8.7336\times 10^{3}\ W/m^{2}$ and $\dot{m}_{surf,\max}=3.1499\times 10^{-3}\ kg/m^{2}\cdot s$ respectively at the distance of $x=11.7\times 10^{-3}m$ then rapidly decreased but tend to decrease slowly to zero as x increases. Similarly, the total heat and mass transfer rates are rapidly increased at the inlet range but slowly as x increases.

The heat and mass transfer coefficients get the highest values at adjacent to the inlet. Those are $h_{t,\max}=3.5386\times 10^{3}\ W/m^{2}\cdot K$ and $h_{m,\max}=1.7527\times 10^{-4}m/s$ respectively at the distance of $x=1.67\times 10^{-3}m$ then rapidly decreased but tend to steady as x increases.

REFERENCES

1. G. Grossman, Simultaneous heat and mass transfer in film absorption under laminar flow, International Journal of Heat and Mass Transfer 26 (3) (1983) 357-371.
2. N. Kawae, T. Shigechi, K. Kanemaru, T. Yamada, Water vapor absorption into laminar film flow of a lithium bromide-water solution (Influence of variable properties and inlet film thickness on the absorption mass transfer rate, Scripta Technica Inc., 1989, pp. 58-70.
3. S.C. Kim, M.D. Oh, J.H. Lee, A numerical study on heat and mass transfer in a falling film of vertical plate absorber cooled by air, Transaction of the KSME 19 (4) (1995) 1071-1082.
4. L.A. McNeely, Thermodynamic properties of aqueous solution of lithium bromide, ASHRAE Transaction 85 (1) (1979) 413-434.
5. S.V. Patankar, Numerical Heat Transfer and Fluid Flow, Hemisphere Publishing Corporation, 1980.