Journal of the Korean Society of Marine Engineers, Vol.28, No.2, pp.252~262, 2004(ISSN 1226-9549)

Numerical Analysis of Vertical Plate Absorber for Optimal Design

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(Manuscript : Received DEC 8, 2003 ; Revised JAN 5, 2004)

Abstract : A model of simultaneous heat and mass transfer process in absorption of refrigerant vapor into a lithium bromide solution of water-cooled vertical plate absorber, which was considered to the change of refrigerant vapor pressure along the plate width direction, was developed to evaluate the compactness of plate absorber and supply basis data for optimal design of plate absorber. The effects of plate interval as well as the effect of capacity for one piece of plate absorber on plate absorber size such as plate height, plate heating area and plate absorber volume have been investigated. It is confirmed that there is exist an optimal plate interval minimizing plate absorber volume. And the smaller capacity for one piece of plate absorber, the smaller plate absorber volume is obtained.

Key words : Absorption Heater/Chiller, Refrigeration, $LiBr/H_2O$, Numerical Analysis, Absorber

Nomenclature

C	: Concentration [wt%]
c_p	: Specific heat at constant pressure [kJ/kg · K]
D	: Diffusion coefficient [m^2/s]
dh	: Hydraulic diameter [m]
dz	: Strip width [m]
g	: Gravitational acceleration [m/s^2]
H	: Enthalpy [kJ/kg]
H_{abs}	: Heat of absorption [kJ/kg]
H_{latent}	: Latent heat of evaporation [kJ/kg]
$H_{dilution}$	: Heat of dilution [kJ/kg]
k	: Thermal conductivity [W/m · K]
$\dot{m}$	: Mass flux [kg/ $m^2 \cdot s$]
M	: Mass transfer rate [kg/s]
n	: Number of plate
P	: Pressure [kPa]
$\dot{q}$	: Heat flux [W/m^2]
Q	: Heat transfer rate [W]
Re	: Reynolds number
T	: Temperature [K]
u	: Velocity in x-direction [m/s]
v	: Velocity in y-direction [m/s]
w	: Vapor velocity [m/s]
x	: Coordinate in direction of flow [m]

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y : Coordinate in direction perpendicular to flow [m]

Greek letters

α : Thermal diffusivity [m^2/s]
δ : Liquid film thickness [m]
μ : Dynamic viscosity [Pa・s]
ρ : Density [kg/m^3]
Γ : Liquid mass flow rate per unit width [kg/ms]

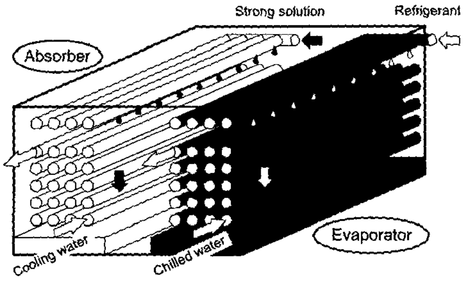
Subscripts

1 : Solid LiBr
2 : H_2O
abs : Absorption
c : Cooling water
f : Liquid film
in : Inlet
mean : Average value
out : Outlet
s : Absorption solution
surf : Liquid-vapor interface
wall : Plate wall

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 since the plate absorber can enlarge heat transfer per unit volume in comparison with the horizontal shell and tube absorber as shown in Fig. 1.

Most of previous theoretical works on plate absorber dealt with the modeling of simultaneous heat and mass transfer in a film falling down an inclined or vertical plate. In these models, the liquid film was assumed falling down one side of plate and the refrigerant vapor pressure is constant along the plate width direction. However, in a real plate absorber, the refrigerant vapor pressure is decreased along the plate width direction and the value of refrigerant vapor pressure drop depends on the interval between two adjacent plates. Grossman [1] analyzed absorption process of refrigerant vapor in a steady liquid film falling down over an incline plane on the assumption that uniform temperature and heat flux at wall, and the physical properties of liquid solution are constant and independent of temperature. Kawae et al. [2] performed research on effect of change of physical properties of LiBr solution and effect of operating condition on absorption mass transfer rate by numerical analysis on the assumption that wall temperature is uniform. Kim et al. [3] analyzed a falling film of LiBr solution of vertical plate wall cooled by air. Heat flux at the plate wall was determined from the cooling air temperature and heat transfer coefficient between the wall and the cooling air. Jeong et al. [4] proposed an unsteady quasi one-dimensional model of momentum, heat and mass transfer in a falling film of an air-cooled vertical plate absorber. The effect of operating condition on absorption mass flux was analyzed.



Shell and tube absorber

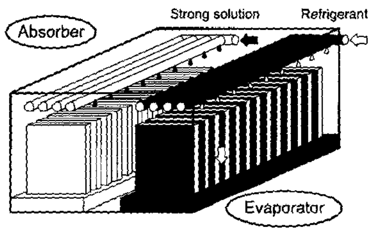


Plate absorber

Fig. 1 Comparison between shell and tube absorber and plate absorber

The purpose of the present study is to develop a model of simultaneous heat and mass transfer process in absorption of refrigerant vapor into a lithium bromide solution of water-cooled vertical plate absorber to evaluate the compactness of plate absorber and supply basis data for optimal design of plate absorber. Unlike most previous analysis models of a plate absorber, the change of refrigerant vapor pressure along the plate width direction was considered in this study. The analysis was carried out with three kinds of different capacity for one piece of plate absorber, which are 0.5RT, 1RT and 2RT, to investigate the effects of plate interval as well as the effect of capacity for one piece of plate absorber on plate absorber size such as plate height, plate heating area and plate absorber volume.

2. Analysis model of plate absorber

2.1 Model of entire plate absorber

The schematic diagram of entire plate absorber is shown in Fig. 2. Vertical plates are set in a row with the same interval. Strong solution is equally distributed and flowed down each plate in both of plate side. The refrigerant vapor from evaporator comes into absorber symmetrically at both of inlet side. Because of symmetrically at the inlet, the absorption process is similar in both of half plate. The refrigerant vapor is flowed along the plate width direction from the inlet to the center of plate and refrigerant vapor pressure is decreased following. Therefore the analysis was carried out on only half plate.

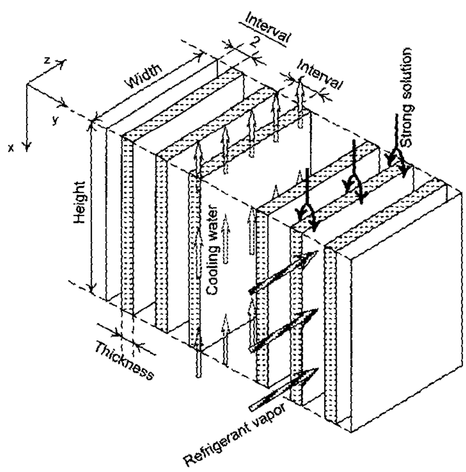


Fig. 2 Analysis model of plate absorber

2.2 Model of solution falling film for each plate

The numerical analysis of the absorption process in the present study is described schematically in Fig. 2. 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. At inlet solution ($x=0$), the liquid solution is at a uniform temperature and concentration corresponding to an equilibrium saturation pressure, which is different from refrigerant vapor pressure. As the result of this difference, a mass transfer process takes place at the liquid-vapor interface. The refrigerant vapor absorbed at the interface diffuses into the liquid film. The heat generated in the absorption results in a simultaneous heat transfer process [5]. The heat of absorption released at the interface is transported through the film, which is cooled by cooling water flowed inside the plate.

In formulating this model, the following assumptions have been made:

(1) The flow of the liquid film is laminar and fully developed throughout.
(2) The liquid solution is Newtonian and steady state.
(3) The mass of vapor absorbed per unit time is small compared to the mass flow rate of the liquid. Therefore it is assumed that the latter is constant and so are the film thickness and average flow velocity.
(4) No shear forces are exerted on the liquid by the vapor at the interface.
(5) There is no heat transfer in the vapor phase.
(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 liquid film and its temperature is changed linearly.

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

The film Reynolds number is defined as

$$Re_f = \frac{4\Gamma}{\mu} \tag{1}$$

For a laminar film flowing at a sufficient low Reynolds number ($Re_f < 20$) over a vertical plate, the film surface is smooth and its thickness is constant. The film thickness is determined by Nusselts analysis [6] as follows

$$\delta = \left(\frac{3\Gamma\mu}{\rho^2 g}\right)^{1/3} \tag{2}$$

Because of constant film thickness, the cross-stream velocity is neglected ($v=0$), the downstream velocity u profile shown in Fig. 3 is parabolic as a function of y and given by [5]

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

where u_{mean} is the average flow velocity determined as follows

$$u_{mean} = \frac{\Gamma}{\rho\delta} \tag{4}$$

The energy equation written in the two-dimensional rectangular film coordinates is [2]

$$\rho c_p\left(u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}\right)=k\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right)+\frac{\partial}{\partial y}\left(\sum_{i=1}^{2}\rho D_i\frac{\partial C_i}{\partial y}H_i\right) \quad (4)$$

The relationship of concentration and mass diffusion coefficient between solid LiBr and H_2O as follows

$$C_1=1-C_2=C \quad D_1=D_2=D \quad (6)$$

For most liquid, the dependence of the enthalpy on temperature is almost linear. Thus, the enthalpy may be expressed as [7]

$$H_i=c_{pi}T \quad (7)$$

where c_p is the specific heat.

Substituting equation (6) and equation (7) into equation (5), the energy equation becomes

$$u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}=\alpha\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right)+\left(\frac{\partial^2 C}{\partial y^2}T+\frac{\partial C}{\partial y}\cdot\frac{\partial T}{\partial y}\right)\frac{D(c_{p1}-c_{p2})}{c_p} \quad (8)$$

The transport of the absorbate solute in a liquid absorbent film is governed by the diffusion equation [2]

$$u\frac{\partial C}{\partial x}+v\frac{\partial C}{\partial y}=D\left(\frac{\partial^2 C}{\partial x^2}+\frac{\partial^2 C}{\partial y^2}\right) \quad (9)$$

The following boundary conditions are applied

At solution inlet, the profiles of liquid film temperature and concentration can be assumed to be uniform

$$T=T_{s,in} \quad C=C_{s,in} \quad \text{at} \quad x=0 \quad (10)$$

At solution outlet, the gradients of liquid film temperature and concentration at the outlet can be assumed to equal to zero

$$\frac{\partial T}{\partial x}=0 \quad \frac{\partial C}{\partial x}=0 \quad \text{at} \quad x=L \quad (11)$$

At plate wall, the flow velocity u and the gradient of concentration of liquid film equal to zero. The liquid film temperature equals to wall temperature.

$$u=0 \quad T=T_{wall} \quad \frac{\partial C}{\partial y}=0 \quad at\ y=0 \quad (12a)$$

The plate wall was cooled by cooling water and under the assumptions (7) and (8), the wall temperature at a distance of x from the inlet can be determined as follows [8]

$$T_{wall}=T_{c,in}+\left(\frac{L-x}{L}\right)\left(T_{c,out}-T_{c,in}\right) \quad (12b)$$

where $T_{c,in}$ and $T_{c,out}$ are the inlet and outlet cooling water temperature respectively.

At liquid-vapor interface, under assumption (6), the solution concentration at the liquid-vapor interface C_{surf} is determined from the relation between the solution temperature and concentration in equilibrium with refrigerant vapor pressure at the interface was given in [9]

$$C_{surf}=C_{surf}\,(T_{surf},P) \quad \text{at} \quad y=\delta \quad (13a)$$

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

$$\dot{m}_{surf}=-\rho D\left(\frac{\partial C}{\partial y}\right)_{y=\delta} \quad (13b)$$

$$M_{surf}=\int_0^L \dot{m}_{surf}\,(x)dx \quad (13c)$$

The absorption heat flux generated at the interface is

$$\dot{q}_{surf} = k\left(\frac{\partial T}{\partial y}\right)_{y=\delta}(c_{p2}-c_{p1})T\rho D\left(\frac{\partial C}{\partial y}\right)_{y=\delta} = H_{abs}\,\dot{m}_{surf} \tag{13d}$$

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

$$H_{abs} = H_{latent} + H_{dilution} \tag{13e}$$

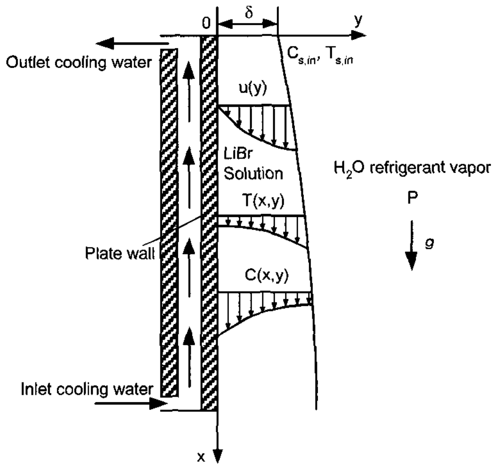


Fig. 3 Schematic diagram of absorption process in water-cooled vertical plate absorber

2.3 Model of refrigerant flow for each strip

The half plate is divided into z similar discrete strips. The refrigerant vapor is assumed flowing in a rectangular duct with the area of cross-section area equals [height×interval] and the length is [width/2]. The governing equations of refrigerant vapor flow at k^{th} strip are expressed as follows:

The absorbed refrigerant mass flow rate

$$M_{abs} = M_{surf}\cdot dz\cdot height \tag{14a}$$

$$\text{where} \quad dz = \frac{width}{2z} \tag{14b}$$

The inlet refrigerant vapor mass flow rate

$$M_{H_2O(k)} = M_{H_2O(k-1)} - 2M_{abs(k-1)} \tag{15}$$

The refrigerant vapor pressure drop is determined by using the equation of pressure drop in rectangular duct [10]

$$\Delta P_{H_2O(k)} = \frac{96}{Re_{H_2O(k)}}\cdot\frac{dz}{dh}\cdot\rho_{H_2O}\cdot\frac{w^2_{H_2O(k)}}{2} \tag{16a}$$

$$\text{where} \quad Re_{H_2O(k)} = \frac{w_{H_2O(k)}dh}{\nu} \tag{16b}$$

$$dh = \frac{4\cdot height\cdot interval}{2(height + interval)} \tag{16c}$$

Therefore, the refrigerant vapor pressure is determined as follows

$$P_{H_2O(k)} = P_{H_2O(k-1)} - \Delta P_{H_2O(k-1)} \tag{17}$$

Bound ($k=1$)ary condition at inlet refrigerant vapor

$$P_{H_2O(1)} = P_{H_2O,inlet} \quad M_{H_2O(1)} = M_{H_2O,inlet} \tag{18}$$

3. Method of numerical analysis

Three kinds of capacity for one piece of plate absorber, which are 0.5RT, 1RT and 2RT, have been investigated to evaluate the compactness of plate absorber. The operating conditions were chosen similar to an actual operating condition of absorption chiller/heater. Table 1 shows the operating conditions for 1RT plate absorber unit.

Table 1 Typical operating conditions for 1RT plate absorber unit

Fluids	Conditions	Values
Refrigerant	Flow rate [kg/h]	5.4
	Pressure [kPa]	1
Solution	Flow rate [kg/h]	62.5
	Inlet temperature [℃]	46
	Inlet concentration [wt%]	60.2
Cooling water	Inlet temperature [℃]	32
	Outlet temperature [℃]	36

In calculation for each strip, the integral forms of conservation equations were solved by the finite volume method proposed by Patankar [11]. The power-law scheme was used to treat the convection-diffusion term and get the discretization equations. The TDMA method was applied line by line to solve the system of discretization equations.

In calculation for half plate, the plate height is controlled to increase until the plate capacity obtained by calculation satisfies the value of plate capacity, which is set at input data.

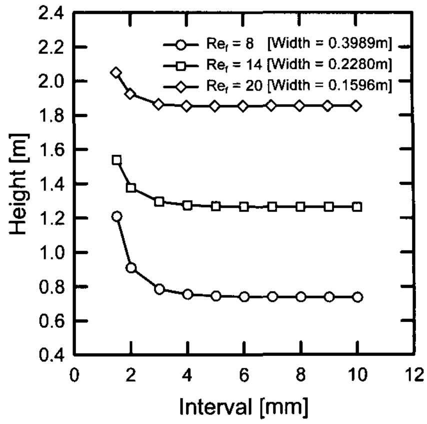
The plate absorber volume is calculated as shown in Fig. 2

$$Volume = n \times (interval + thickness) \times height \times width \quad (19)$$

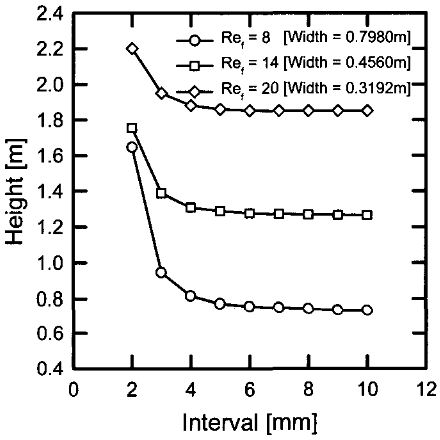
4. Results and discussions

4.1 Effect of plate interval on plate height

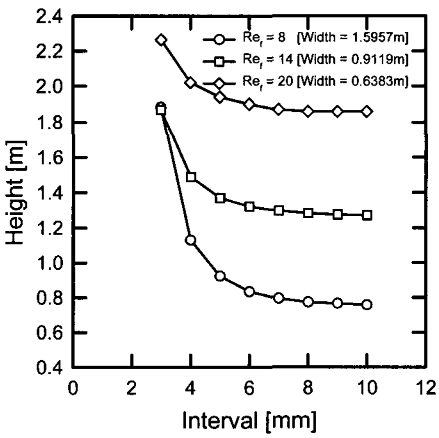
The effect of plate interval on plate height for 0.5RT, 1RT and 2RT are shown in Fig. 4. The plate height decreases rapidly at the small value of the plate interval as plate interval increases but slowly at the high value of plate interval.



(a) 0.5 [RT/piece]



(b) 1 [RT/piece]



(c) 2 [RT/piece]

Fig. 4 Effect of plate interval on plate height

It can be explained that as plate interval is narrow, refrigerant vapor pressure drop changes rapidly so it is more effective to absorption mass flux therefore plate height also change rapidly but as plate interval is wider, the refrigerant vapor pressure drop almost constant.

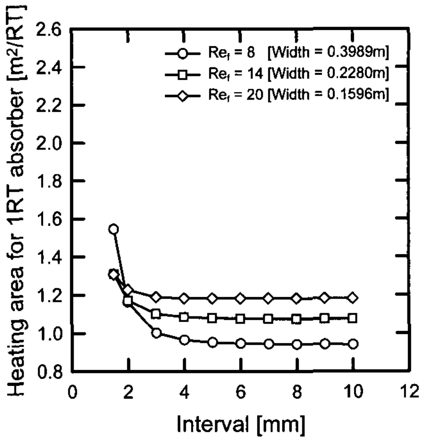
In the case of smaller Reynolds number with the same plate interval, the plate width is wider with the result that the plate height is also lower following. In the case of bigger capacity for one piece of plate absorber with the same plate interval, the plate height is higher since as capacity for one piece bigger, the plate height and the plate width have to be increase to get a bigger heating area, which is enough to absorb refrigerant vapor.

4.2 Effect of plate interval on plate heating area

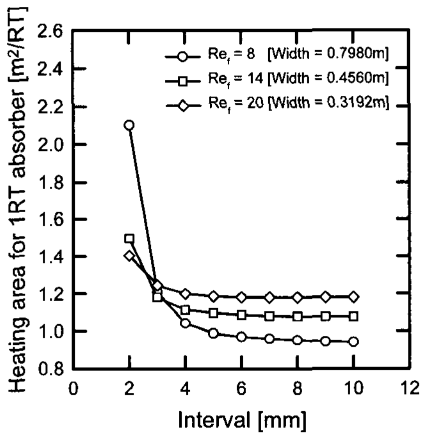
Fig. 5 show the effect of plate interval on plate heating area for 1RT absorber in the case of 0.5RT, 1RT and 2RT respectively. The change of heating area due to plate interval is similar to the change of plate height except in the case of Reynolds number. As Reynolds number increases, the plate heating area decreases at the small value of the plate interval but increases at the high value of plate interval.

4.3 Effect of plate interval on plate absorber volume

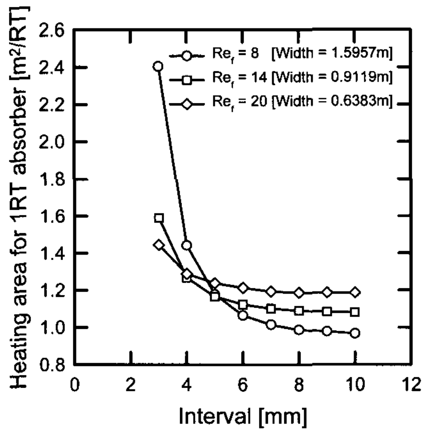
The effect of plate interval on plate absorber volume for 20RT plate absorber unit in the case of 0.5RT, 1RT and 2RT



(a) 0.5 [RT/piece]

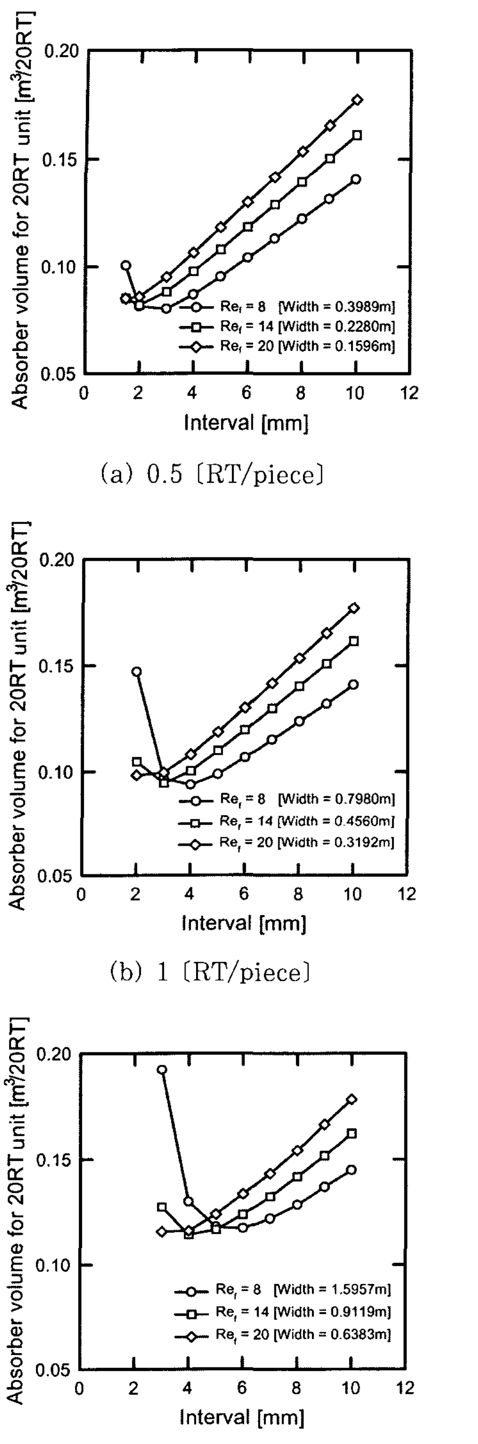


(b) 1 [RT/piece]



(c) 2 [RT/piece]

Fig. 5 Effect of plate interval on plate heating area



(a) 0.5 [RT/piece]

(b) 1 [RT/piece]

(c) 2 [RT/piece]

Fig. 6 Effect of plate interval on plate absorber volume

are respectively shown in Fig. 6. The plate absorber volume decreases at the small value of the plate interval as plate interval increases but increases at the high value of plate interval. It can be explained that as plate interval is wider, the volume increase but the plate height decrease following so the volume also decrease, therefore, it is exist an optimal plate interval, which minimizes the plate absorber volume.

In the case of 0.5RT and Reynolds number is 8, the optimal plate interval is 3mm. In the case of 1RT and Reynolds number is 8, the optimal plate interval is 4mm. Finally, in the case of 2RT and Reynolds number is 14 and optimal plate interval is 4mm. In the case of smaller capacity for one piece of plate absorber, the plate absorber volume is smaller. As shown in figures, in the case of 0.5RT, Reynolds number is 8 and optimal plate interval is 3mm, the plate absorber volume is smallest. In the case of 2RT, Reynolds number is 14 and optimal plate interval is 4mm, the plate absorber volume is biggest.

5. Conclusions

The compactness of water-cooled vertical plate absorber is evaluated to supply basis data for optimal design of plate absorber. The result obtained can be summarized as follows:

The plate height decreases rapidly at the small value of the plate interval as plate interval increases but slowly at the high value of plate interval. The plate height increases as Reynolds number increases and capacity for one piece of plate absorber increases with the same plate interval. As Reynolds number increases, the plate heating area decreases at the small value of the plate interval but increases at the high value of plate interval. It is confirmed that there is exist an optimal plate interval minimizing plate absorber volume. It is equal to 3mm in the case of 0.5RT and Reynolds number is 8. In the case of smaller capacity for one piece of plate absorber, the plate absorber volume is smaller. Among three kind of capacity for one piece of plate absorber, in the case of 0.5RT, Reynolds number is 8 and optimal plate interval is 3mm, the plate absorber volume is smallest. The present model was not considered to effect of film disturbance by surfactant. Actually, the plate absorber can be obtained smaller volume by adding surfactant into absorption solution.

Acknowledgement

This study was supported financially by the Korea Science and Engineering Foundation through the Center For Advanced Environmentally Friendly Energy Systems, Pukyong National University, Korea (Project number : R12-2003-001-00002-0).

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