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Heat transfer enhancement with a surfactant on horizontal bundle tubes of an absorber

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Abstract

This paper is concerned with the enhancement of heat transfer by surfactants added to aqueous lithium bromide (LiBr) solutions. Three different kinds of tubes in horizontally stagged arrangement are tested with and without an additive of normal octyl alcohol. The test tubes are a bare tube, a floral tube and a hydrophilic tube. The additive mass concentration is about 0.05–5.5 wt%. The heat transfer coefficient is measured as a function of the solution flow rate in the range 0.01–0.034 kg m^{-1} s^{-1}. The experimental results are compared with/without surfactants. Among three kinds of tubes, the hydrophilic tube shows the highest permeability. It has 4–73% higher wetted area than that of the bare tube, and 10–22% higher than that of the floral tube. Without surfactants, the hydrophilic tube is in the range 10–35% higher heat transfer coefficient than that of the bare tube, and 5–25% higher than that of the floral tube. With surfactants, the increase of the heat transfer coefficient is about 35–90% for the bare tube, 40–70% for the floral tube, and 30–50% for the hydrophilic tube.

1. Introduction

In recent years, the usage of an absorption chiller/heater is positively encouraged for the unused energy application, the preservation of earth environment, and the settlement of unbalance demand between electric power and city gas. Therefore, an absorption chiller/heater can reduce the demand for electric power by using unused city gas during the summer time. However, an absorption chiller/heater has a disadvantage. The size of an absorption chiller/heater is larger than that of a vapor compression type chiller/heater based on the same capacity. An absorption chiller/heater is composed of an absorber, an evaporator, a condenser, a generator, and a solution heat exchanger. Among these components the absorber has the largest volume. The absorber has about 33% of the total heat transfer area, and about 27% of the total volume [1]. Therefore, to have a high efficiency of an absorption chiller/heater, the absorber must be investigated carefully.

The absorber, which is the heat exchanger of the falling film type, is generally used because this type minimizes the performance decrease caused by the pressure loss. To improve the efficiency of the absorber the development of tubes having high efficiency and supply of proper surfactants is to be considered. Hoffmann [2] and Yoon et al. [3] reported that both the cases enhance the heat and mass transfer and improve the performance of the absorber. The development of the high efficiency tubes needs new investment and increases the production rate. Thus, many use the second approach of simply supplying the surfactants for improving the system performance. Many researchers reported that the surfactant additives dramatically affect the solvent [4]. The performance enhancement by supplying the surfactant is due to a decrease of the surface tension of the absorption solution and the surface disturbance by the Marangoni convection [5,6]. However, the mechanism for improving the performance is not clear. Cosenza and Vliet [7] investigated falling film absorption for smooth horizontal tubular surfaces. They obtained

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Nomenclature

A	heat transfer area (m^2)
C	concentration (wt%)
c_p	specific heat at constant pressure ($J\ kg^{-1}\ K^{-1}$)
d	diameter of the heat transfer tube (m)
G	mass flow rate ($kg\ s^{-1}$)
h	convection heat transfer coefficient ($kW\ m^{-2}\ K^{-1}$)
L	length of the heat transfer tube (m)
N	number of hill
Nu	Nusselt number ($= hL\lambda^{-1}$)
P	number of heat transfer tube
ppm	parts per million by mass
Pr	Prandtl number ($= c_p\mu\lambda^{-1}$)
Q	quantity of heat (kW)
Re	Reynolds number ($= V_{avg}Lv^{-1}$)
U	overall heat transfer coefficient ($kW\ m^{-2}\ K^{-1}$)
V_{avg}	average velocity in a tube ($m\ s^{-1}$)
T	temperature (°C)
ΔT_{lm}	logarithmic mean temperature difference

Greek letters

Γ	solution mass flow rate per unit length ($kg\ m^{-1}\ s^{-1}$)
v	kinematic viscosity ($m^2\ s^{-1}$)
λ	thermal conductivity ($kW\ m^{-1}\ K^{-1}$)
μ	dynamic viscosity ($N\ s\ m^{-2}$)

Subscript

A	absorber
co	cooling water
i	inlet (inside)
o	outlet (outside)
s	solution
T	total

experimental data for absorption of water vapor into an aqueous LiBr solution flowing over internally cooled horizontal tubes. Choudhurry et al. [8] analyzed the absorption phenomena of absorbent solution film flowing over a cooled horizontal tube. They obtained an optimum flow rate for a particular tube size on the basis of flow rate versus total mass flux relation.

In this paper, enhancement of the heat transfer by using three kinds of tubes, a bare, a floral, and a hydrophilic, is tested experimentally. In the absorber horizontally stagged tubes are arranged to investigate the absorption enhancement. The experimental set-up has a commercial size. Thus, the error caused by the reduction of the test section can be minimized. Also, the effect of additives can be seen clearly. The fundamental experimental data for falling film absorption with surfactants are limited. In commercial equipment, additives are supplied by experience. This research is conducted to investigate the effects of three different tubes and surfactant additives on the absorption process.

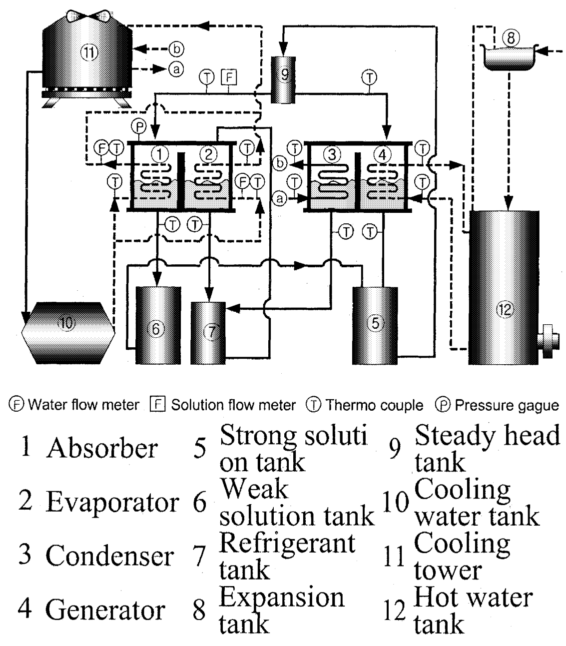


Fig. 1. Experimental apparatus.

2. Experimental apparatus and method

Fig. 1 shows a schematic diagram of the experimental apparatus. As shown in the figure, the apparatus consists of an absorber, an evaporator, a condenser, a generator, strong/weak solution tanks, and a refrigerant tank. These components are connected with pipes. Pipes and tanks are constructed of 321 stainless steel. The glass areas are installed to see the inner situation of these components. In the absorber 48 horizontal tubes of 400 mm length are installed. The tubes are arranged with six columns and eight rows (see Fig. 2 for the detailed construction of the absorber). The solution (water/LiBr) regenerated at a generator is stored at a strong solution tank, and the solution, which absorbs refrigerant (water vapor) at an absorber, at a weak solution tank. At the inlet of the absorber, the steady head tank is installed to prevent variations of the solution flow rate caused by pulsation phenomena of the solution pump. A vacuum pump (5×10^{-4} Torr) is then installed to keep the vacuum pressure constant in the apparatus. A circulation pump is used to circulate cooling water in the absorber

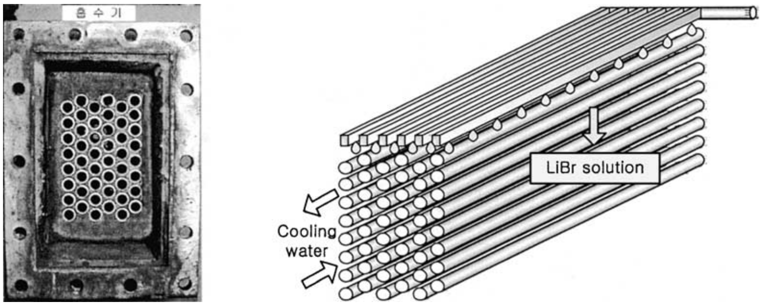


Fig. 2. Photograph and schematic drawing of the absorber structure.

and chilled water in the evaporator. At the inlet/outlet of the tubes of the absorber, the evaporator, the generator, the condenser, and the tanks, thermocouples (copper/constantan) are installed to measure temperature variations. Flowmeters are installed to measure the flow rate of cooling water, chilled water, and solutions. A detailed description of the experimental apparatus is given by Seol [9]. According to all instrument errors, uncertainity in the average heat transfer coefficients was 5.0–10.3% [10].

From the solution pump, the strong solution in the strong solution tank flows down over the outer surface of horizontal tubes of the absorber through a tray installed at the upper part of the absorber. The refrigerant stored at the refrigerant tank flows over the outer surface of the tubes of the evaporator heat exchangers. And at the inner surface of the tubes of the evaporator, the cooling load is given by flowing cooling water. The refrigerant vapor evaporated from the evaporator is injected to the absorber through the eliminator located between the evaporator and the absorber. The rest of the refrigerant returns to the refrigerant tank. The absorber is kept at a vacuum pressure of 7 mm Hg. The solution temperature is adjusted depending on the saturated temperature with changes of the solution concentration. The strong solution absorbs the refrigerant vapor evaporated from the evaporator falling down the horizontal tubes. The steady head tank is installed at the inlet of the absorber to have a constant flow rate. The rest of the strong solution returns to the strong solution tank through a bypass tube. The weak solution weakened by absorbing the refrigerant is stored at the weak solution tank. The cooling water in the cooling water tank is supplied to the evaporator and the absorber. The cooling water supplied at the absorber obtains heat flowing though the heat transfer tubes, and the chilled water deprives of heat at the evaporator. The working fluid cools to the appropriate temperature at the cooling tower. As an additive the *n*-octanol is used in six different concentrations of 500, 1500, 2500, 3500, 4500, and 5500 ppm (parts per million by mass).

Table 1
Experimental conditions

Items	Parameters	Conditions
Refrigerant	Evaporating temperature (°C)	6
LiBr solution	Inlet concentration (wt%)	60
	Inlet temperature (°C)	45
	Mass flow rate (kg m^{-1} s^{-1})	0.01–0.034
Cooling water	Inlet temperature (°C)	32
	Velocity (m s^{-1})	1.0
Surfactant	*n*-Octanol (ppm)	500–5500

In this study, three different types of tubes are used such as a bare tube, a hydrophilic tube, and a floral tube. The bare tube has an outer diameter of 15.88 mm and an inner diameter of 14.05 mm. The hydrophilic tube was plasma-treated to induce the best hydrophilic property on the tube surface. The best proportion of reaction gases was found to be the ratio of acetylene to nitrogen 7:3 [11]. The hydrophilic tube has the same diameters of the bare tube. The floral tube has an outer diameter of 15.88 mm and an inner diameter of 13.88 mm. The hydrophilic tube has an enhanced surface to improve the permeability, and has the same geometry with the bare tube. Experimental conditions are shown in Table 1. Figs. 3 and 4 show the specification and the photograph of test tubes, respectively.

2.1. Calculation of the heat transfer coefficient

The log mean temperature difference for a bank of tubes is defined as follows:

$$\Delta T_{\rm lm} = \frac{(T_{\rm Asi} - T_{\rm Acoo}) - (T_{\rm Aso} - T_{\rm Acoi})}{\ln\{(T_{\rm Asi} - T_{\rm Acoo})/(T_{\rm Aso} - T_{\rm Acoi})\}}, \qquad (1)$$

where $T_{\rm Acoo}$ and $T_{\rm Acoi}$ are the average cooling water temperatures for the inlet and outlet tubes in the bank, respectively, and $T_{\rm Asi}$ and $T_{\rm Aso}$ are the equilibrium tem-

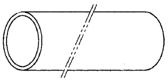
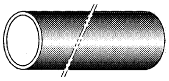
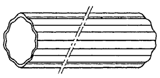
Type	Appearance	Dimensions
Bare tube		d_o=15.88 mm d_i=14.05 mm L=400 mm
Hydrophilic tube		d_o=15.88 mm d_i=14.05 mm L=400 mm
Floral tube		d_o=15.88 mm d_i=13.88 mm N=11 L=400 mm

Fig. 3. Specification of the test tubes.

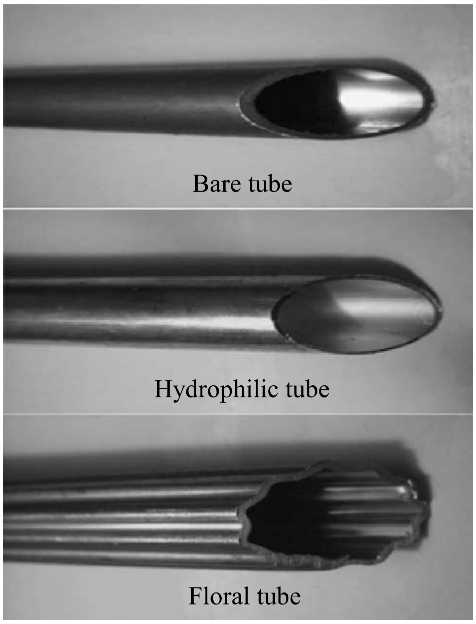


Fig. 4. Photograph of the test tubes.

peratures at the absorber for the inlet and outlet solutions, respectively.

The heat quantity transferred to the cooling water is

$$Q = G_{co}c_{pco}(T_{coo} - T_{coi}) = UA_T\Delta T_{lm}, \tag{2}$$

where $A_T = \pi d_o \cdot L_T$.

An experimental correlation for the convection heat transfer coefficient of the cooling waterside of the inner tube surface, h_i, given by Dittus–Boelter is

$$Nu = 0.023Re^{0.8}Pr^{0.4} = \frac{h_i L}{\lambda}. \tag{3}$$

The heat transfer coefficient of the outer tube surface of the absorption solution, h_o, can be obtained from Eq. (4) and the thermal resistance of the tube wall is assumed to be negligible.

$$h_o = \frac{1}{1/U - d_o/(d_i h_i)}. \tag{4}$$

Finally, the mass flow of the solution per tube length, given by Eq. (5), is

$$\Gamma_s = \frac{G_s}{(2LP)}. \tag{5}$$

3. Results and discussion

Fig. 5 shows photographs of falling film at the atmospheric pressure for 0.015 kg m^{-1} s^{-1} solution flow rate. Experimental results for the variation of the wetted areas with the mass flow rate for the previous figure are plotted in Fig. 6. Comparing the bare tube and the hydrophilic tube, the wetted area of the hydrophilic tube is about 110% higher than that of the bare tube when the flow rate is 0.0075 kg m^{-1} s^{-1}. As the mass flow rate increases, the rate of increase of the wetted area decreases. At the flow rate of 0.025 kg m^{-1} s^{-1}, the wetted area of the hydrophilic tube is 30% higher than that of the bare tube. The floral tube shows 20–70% higher wetted area than that of the bare tube as the flow rate decreases. It is concluded that the hydrophilic tube shows better permeability than the other tubes. It is because that the surface of the hydrophilic tube is treated to increase the permeability.

Fig. 7 compares the heat transfer coefficients for three different types of tubes in terms of solution flow rates with/without surfactants. For the experiment the temperature of the solution supplied at the absorber is set to have equilibrium with pressure. The concentration of surfactant is 3500 ppm by mass. Wasekar and Manglik [12] and Hetsroni el al. [13] investigated the critical concentration of the surfactant for the saturated

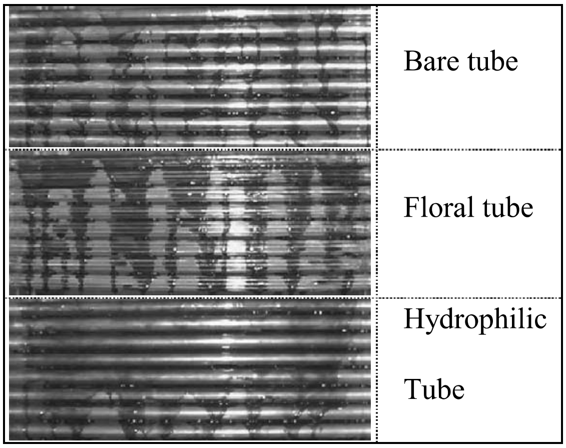


Fig. 5. Photograph of flow pattern.

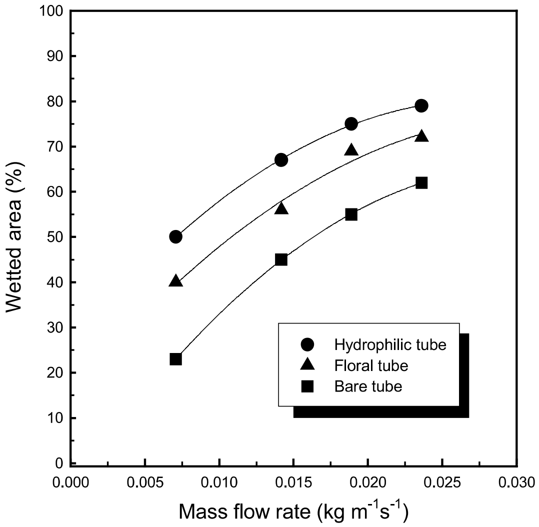


Fig. 6. The variation of wetted area ratio on flow rate.

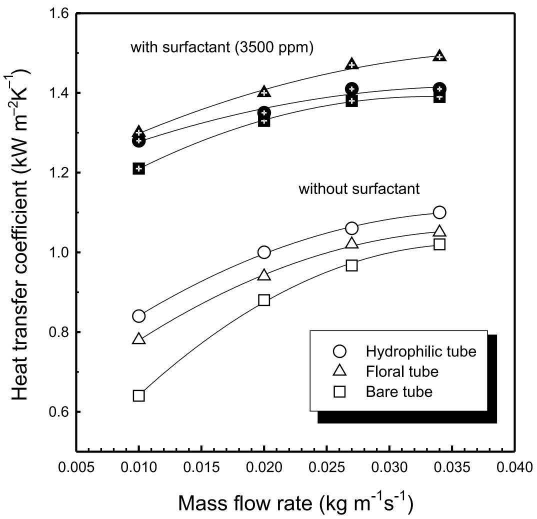


Fig. 7. Comparison of heat transfer coefficients for three types of tubes.

pool boiling heat transfer performance of aqueous surfactant solutions. They found the maximum effect of the surfactant in a closed non-circulation system. The present results, given at the experiment of a closed circulation system are difficult to be related to the observation made for the saturated pool boiling heat transfer.

Within the experimental ranges the heat transfer coefficients of all three types of tubes increase with the solution flow rate. Without the additive, the heat transfer coefficients of the hydrophilic tube show about 35% better performance than those of the bare tube at the solution flow rate of 0.01 kg m^{-1} s^{-1} while showing 10% difference at 0.034 kg m^{-1} s^{-1}. In the case of the floral tube, the heat transfer coefficients increase from 5% to 25% although the rate of increase decreases when the flow solution rate increases. With the insertion of a surfactant the heat transfer coefficients increase by about 35–90% in the case of the bare tube, about 40–70% in the case of the floral tube and about 30–50% in the case of the hydrophilic tube. The rate of increase is higher with the small range of solution flow rates than with the large range of solution flow rates. Without a surfactant, the hydrophilic tube shows higher heat transfer coefficients than the floral tube does. With a surfactant, it is opposite. Heat transfer coefficients in the case of the floral tube are higher than in the case of the hydrophilic tube.

Fig. 8 shows the heat transfer coefficients for three different tubes with varying surfactant concentrations and a solution flow rate of 0.027 kg m^{-1} s^{-1}. The experiments are conducted with the additive concentrations of 500, 1500, 2500, 3500, 4500 and 5500 ppm. The absorption heater/chiller usually has a concentration of 2000–3000 ppm. The heat transfer coefficients increase until the surfactant concentration reaches 3500 ppm while it shows no noticeable difference above 3500 ppm for all the three types of tubes. With the surfactant concentration below 1500 ppm, the hydrophilic tube shows a higher heat transfer coefficient than the floral tube does. However, with the surfactant concentration above 1500 ppm, the floral tube shows a higher heat transfer coefficient than that of the hydrophilic tube. With the increase of the surfactant concentration, the hydrophilic tube shows almost same or slightly higher heat transfer coefficient than the bare tube does when the additive concentration is above 3500 ppm.

Figs. 9–11 show the heat transfer coefficients for three types of tubes as a function of the surfactant concen-

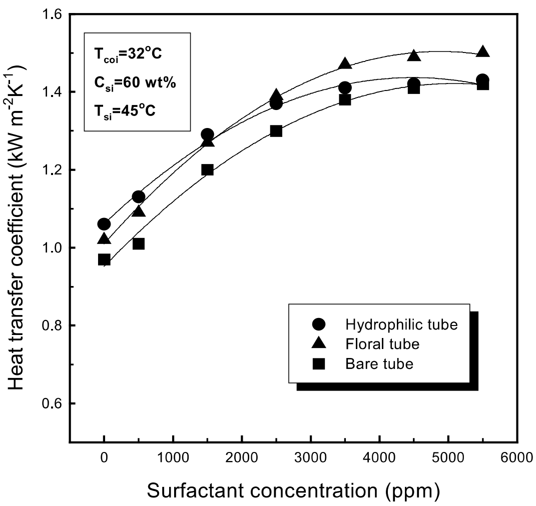


Fig. 8. Effect of surfactant concentration on heat transfer coefficient.

trations for different solution flow rates. With the solution flow rates of 0.01, 0.02, 0.027, and 0.034 kg m^{-1} s^{-1} at $T_{coi} = 35°C$, $C_{si} = 60$ wt%, and $T_{si} = 45°C$, the increase rates of heat transfer coefficients at 3500 ppm decrease 7%, 5%, 1% in the case of the bare tube (see Fig. 9); 8%, 6%, 2% in the case of the floral tube (see Fig. 10); 5%, 4%, 1% in the case of the hydrophilic tube (see Fig. 11). This result leads to a conclusion that when the solution flow rate is low the additive effect of surfactant is high. Among three kinds of tubes the floral tube shows the best improvement of the heat transfer coefficient. The enhancement of the floral tube is obvious. For a low flow rate, the surface geometry of the floral tube increases the heat transfer rates. However, when the solution flow rate increases, the effect of the outer surface diminishes.

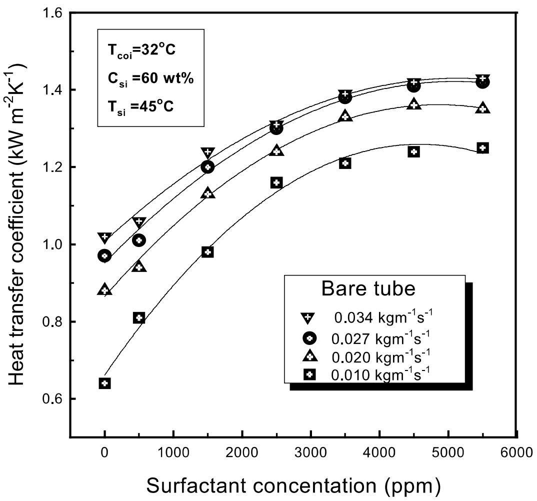


Fig. 9. Effect of surfactant concentration for the bare tube.

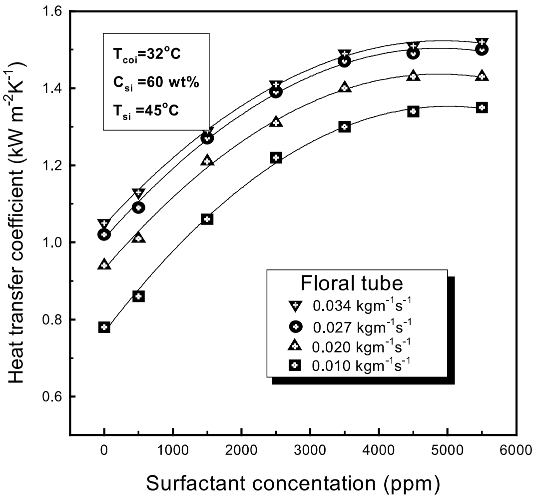


Fig. 10. Effect of surfactant concentration for the floral tube.

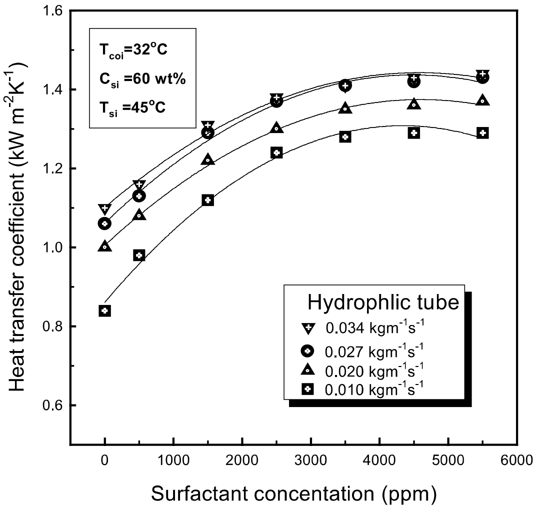


Fig. 11. Effect of surfactant concentration for the hydrophilic tube.

4. Conclusions

The wetted area for three kinds of tubes such as the bare tube, the floral tube, and the hydrophilic tube is visualized and the heat transfer properties are investigated according to the different surfactant concentrations. The conclusions are shown as follows.

1. Among three kinds of tubes, the hydrophilic tube shows the highest permeability. It shows 4–73% higher wetted area than that of the bare tube, and 10–22% higher than that of the floral tube.
2. Without surfactants, the hydrophilic tube has 10–35% higher heat transfer coefficient than that of the bare tube, and 5–25% higher than that of the floral tube.
3. Irrespective of the tubes, the addition of a surfactant more than 3500 ppm does not show an improvement of the heat transfer coefficients. It is an important factor for the decision on a surfactant addition. With surfactants, the floral tube shows the highest heat transfer coefficient, and the heat transfer coefficient increases by about 35–90% for the bare tube, about 40–70% for the floral tube and about 30–50% for the hydrophilic tube.

References

[1] N.U. Inoue, Practical studies on absorbers in Japan, Refrigeration Engineering Division EBARA Corporation, 1988, pp. 18–19.

[2] L. Hoffmann, Experimental investigation of heat transfer in horizontal tube falling film absorber with aqueous solutions of LiBr solution without surfactants, Int. J. Refrig. 19 (1996) 331–341.
[3] J.I. Yoon et al., The effect of surfactant in the absorptive and generative processes, KSME Int. J. 13 (1999) 264–272.
[4] A. Beutler, I. Greiter, A. Wagner, L. Hoffmann, S. Schreier, G. Alefeld, Surfactants and fluid properties, Int. J. Refrig. 19 (1996) 342–346.
[5] T. Kashiwagi, The activity of surfactant in high-performance absorber and absorption enhancement, Int. J. Refrig. 60 (1985) 72–79.
[6] K.J. Kim, N.S. Berman, B.D. Wood, Experimental investigation of enhanced heat and mass transfer mechanisms using additives for vertical falling film absorber, Int. Absorpt. Heat Pump Conf. ASME (1993) 41–47.
[7] F. Cosenza, G.C. Vliet, Absorption in flowing water/LiBr films on horizontal tubes, ASHRAE Trans. 96 (1990) 693–701.
[8] S.K. Choudhurry et al., Absorption of vapors into liquid films flowing over cooled horizontal tubes, ASHRAE Trans. 99 (1993) 81–89.
[9] W.S. Seol, Study on improvement of absorption chiller/heater performance using surfactant, Ph.D. dissertation, Pukyeong National University, Pusan, South Korea, 2000.
[10] R.J. Moffat, Describing the uncertainties in experimental results, Exp. Thermal Fluid Sci. 1 (1988) 3–17.
[11] B.H. Kang, H. Kim, N.S. Park, K.D. Hwang, Effects of hydrophilic surface treatment on evaporation heat transfer at the outside wall of enhanced tubes, in: Proceedings of the 4th JSME–KSME Thermal Engineering Conference, October, Kobe, Japan, 2000.
[12] V.M. Wasekar, R.M. Manglik, Pool boiling heat transfer in aqueous solutions of an anionic surfactant, J. Heat Transfer 122 (2000) 708–715.
[13] G. Hetsroni, J.L. Zakin, Z. Lin, A. Mosyak, E.A. Pancallo, R. Rozenblit, The effect of surfactants on bubble growth wall thermal patterns and heat transfer in pool boiling, Int. J. Heat Mass Transfer 44 (2000) 485–497.