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Experimental Investigation of Heat and Mass Transfer in Absorber with Enhanced Tubes

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In this paper, an experimental study of the absorption process of water vapor into lithium bromide solution is reported. For the purpose of developing high performance absorption chiller/heater utilizing lithium bromide solutions as working fluid, it brings the largest contribution to improve the performance of the absorber which normally requires the largest surface area among the four heat exchangers of the system. The performance of four types of absorber tubes; bare tube, bumping bare tube, floral tube and twisted floral tube, have been experimentally evaluated. The results show that the floral tube and the twisted floral tube show about 40% higher heat and mass transfer performance than the bare tube which is conventionally used in absorbers.

Key Words: Absorption Chiller/Heater, Absorber, Bare Tube, Floral Tube, Twisted Floral Tube, Bumping Bare Tube, Heat and Mass Transfer

Nomenclature

d : Tube diameter, m
G_R : Mass flow rate, kg/s
h : Heat transfer coefficient, kW/(m^2·K)
L : Tube length, m
Ng : Groove number
Q : Interchange heating value, kW
ΔT_{lm} : Log-mean temperature difference, ℃
U : Overall heat transfer coefficient, kW/(m^2·K)

Greeks

Γ : Film flow rate per length, kg/(m·s)
β : Mass transfer coefficient, m/h
ρ : Density, kg/m^3
ξ : Average concentration, wt%
ξ^* : Equilibrium concentration, wt%
$\Delta\xi_{lm}$: Log-mean concentration difference, wt%

Subscripts

A : Absorber
c : Cooling water
i : Inlet
m : Average
o : Outlet
s : Absorption solution

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1. Introduction

The absorption chiller/heater, which is operated by heat energy, does not use the CFC refrigerants, but has an advantage of using relatively cheap gas or waste heat. As the CFC refrigerants, accelerating global warming and ozone layer depletion, will be completely prohibited from using or producing in the near future, more attention has been drawn on the absorption heat pumps and extensive studies have been performed. Especially in the countries that do not have fossil fuel resources, it is important to develop the energy systems for efficient and balanced use of energy like the waste heat recovery and the reduction of a peak electricity demand in summer.

The major components of an absorption heat pump system are an absorber, an evaporator, a condenser and a generator. The absorber has in general the largest influence on the performance of the system. However, the enhancement of the system performance has been limited partly by the conventional tubes which had been chosen based only on heat transfer aspect. Therefore, it is an important subject to reveal an explicit characteristics of absorption phenomenon and an enhanced tube structure for an effective heat and mass transfer.

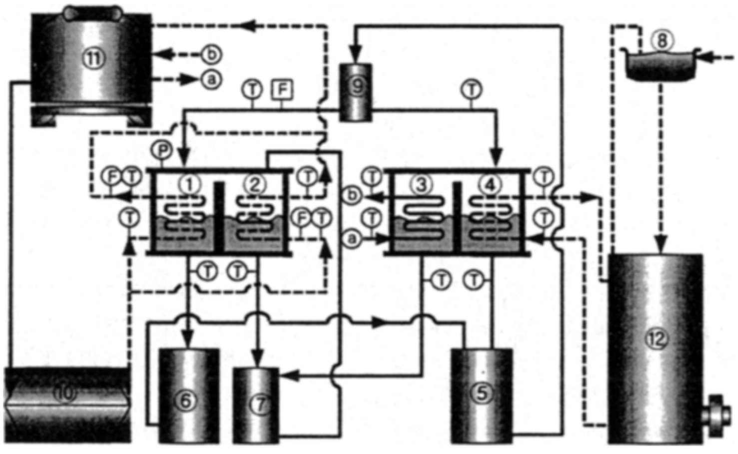
Although the falling-film type absorption is the major operating regime of the absorption system that has been used for several years, fundamental data and correlations for design purposes are seldom found in the open literature. Kawamata et al. (1985) performed an experimental study on the absorption of water vapor into a lithium bromide-water solution film falling outside a thermoexcel C tube and flute tube and analyzed the physical bases of a high performance tube. Furugawa et al. (1993) introduced two kinds of double fluted tubes for the absorber of absorption chiller/heater. They reported that the arm tube and floral tube have higher heat transfer performance than the plain tube conventionally used in a absorber. Isshiki et al. (1991) performed an experimental investigation of water vapor absorption into aqueous solution of LiBr flowing on a vertical CCS (Constant Curvature Surface) tube and compared the results to those of a vertical smooth tube. The degrading effects of non-absorb gas contents during an absorption process were studied for the films flowing over horizontal tubes (Cosenza et al., 1990), for the films inside the vertical tube (Kim et al., 1998). Yoon et al. (1995) conducted the absorption experiments in order to investigate the effects of inside tube diameter, tube length and tube shape for the film flowing inside the vertical tube. Kim et al. (1995) performed an experimental study on the absorption of water vapor into a lithium bromide-water solution film falling inside a vertical tube and analyzed the effects of system pressure and solution temperature. The cases of one to three pipes were studied by Kiyota et al. (1996) for three different surfaces in the range of the film Reynolds number between 5 and 40.

Although many experimental studies have been performed (Nagaoka et al., 1987; Kunugi et al., 1985; Seol et al., 1998), the data and physical interpretation of the phenomena are not fully consistent and partly contradictory. For these reasons, in the present study the absorption processes of heat and mass transfer are experimentally investigated for the water vapor absorption into LiBr-water solution flowing outside a group of horizontal, enhanced tubes. The present study is aimed at developing a new enhanced heat transfer tube by evaluating the performance of various tubes for improving the absorption performance on the basis of the operating conditions of the absorption chiller/heater unit which is currently used. It is also aimed to provide the useful design data for high performance and compact absorption systems.

2. Experimental Apparatus and Procedure

A schematic diagram of the experimental apparatus is shown in Fig. 1. It consists mainly of an absorber (1), an evaporator (2), a condenser (3), a generator (4), a strong and a weak solution tank (5, 6), a refrigerant tank (7), a cooling tower (11) and a hot water tank (12). Also, a vacuum pump with a gauge is installed to maintain a required vacuum in the system. The water circulation loops are constructed for the cooling water in the absorber and for the chilled water in the evaporator. To measure the temperatures of circulating water and solutions, copper-constantan thermocouples are attached on both the inlet and outlet of each component. Flow meters are located as shown in Fig. 1 to measure cooling water, chilled water, solution flow rates. A photographic view of the experimental apparatus is shown in Fig. 2.

The experiments were carried out in three phases of (1) experimental condition setting, (2) performance prediction and (3) solution generation. In the phase of experimental condition setting, the solution in the strong solution tank is first circulated by the solution pump after the system have been vacuumed. At this time the solution passes through the heat exchanger in which it exchanges heat with appropriate hot



1 Absorber
2 Evaporator
3 Condenser
4 Generator
5 Strong solution tank
6 Weak solution tank
7 Refrigerant tank
8 Expansion tank
9 Steady head tank
10 Cooling water tank
11 Cooling tower
12 Hot water tanks

Fig. 1 Schematic diagram of experimental apparatus.

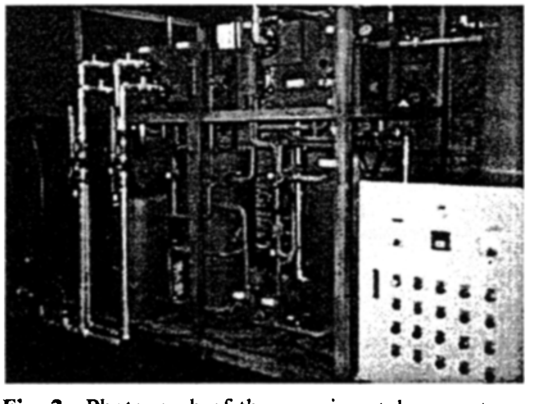


Fig. 2 Photograph of the experimental apparatus.

water and cooling water flow and returns back into the strong solution tank. Hot water temperature is controlled by a thermostat in the hot water heater and cooling water temperature is controlled by the cooling water tank and the cooling tower.

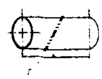
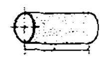
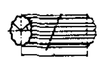
In the phase of performance prediction, the strong solution in the tank is passed through the absorber where it flows down on the test tubes in the form of liquid film. At this time, the solution flow rate at the inlet of a absorber, solution temperatures at the inlet and outlet of the absorber are checked. While the strong solution is passed through the absorber, a steady head tank makes the solution flow constant and extra absorbed solution is returned back to the strong solution tank through a bypass pipe. The refrigerant in the tank is passed through the evaporator. The evaporated refrigerant vapor flows into the absorber where it is absorbed into the solution films on the test tubes. The excess refrigerant liquid is returned back to the refrigerant tank. Here, the weak solution containing refrigerant is held in the weak solution tank and its concentration is checked by the solution being sampled out using a sampling device. Cooling water is pumped and circulated through the evaporator and the absorber. In the absorber, the cooling water and the solution flows form a cross-flow configuration. The cooling water heated in the absorber and the chilled water cooled in the evaporator are first mixed and further cooled down to a desired temperature in the cooling tower. Here, the inlet flow rates of the evaporator and the condenser are checked and also the inlet and outlet temperatures of the absorber are checked, including the cooling water temperature of each pass.

In the phase of solution generation, the weak solution gathered in the tank is circulated to the strong solution tank by the weak solution pump. When all weak solution is circulated to the strong solution tank, it is heated by hot water in the heat

Table 1 Experimental conditions.

Items	Parameters	Conditions
Refrigerant	Evaporation temperature (℃)	9±1
LiBr-solution	Inlet concentration (wt%)	58±0.5
	Inlet temperature (℃)	40±0.5
	Mass flow rate (kg/min)	1.7~8.3
Cooling water	Inlet temperature (℃)	28±0.5
	Mass flow rate (kg/min)	80~310

Table 2 Specification of test tubes.

Type	Appearance	Dimensions
Bare tube		$d_o=15.88$ $d_i=13.88$ $L=400$
Bumping bare tube		$d_o=15.88$ $d_i=13.88$ $L=400$
Floral tube		$d_o=15.88$ $d_i=13.88$ $N_g=11$ $L=400$
Twisted floral tube		$d_o=15.88$ $d_i=13.88$ $N_g=11$ $L=400$ $D=5°$

exchanger after circulation. The heated weak solution becomes the strong solution owing to the separated refrigerant. Then, it is stored at the refrigerant tank after cooling and condensing by cooling water. The strong solution sampled out by the sampling valve is heated in the heat exchanger until it has an appropriate concentration. Cooling water is circulated to the condenser and cooled in the cooling tower. Hot water temperature is controlled by using a heater and a thermostat in the same way as in the setting phase of the experimental condition.

The experimental conditions are summarized in Table 1 and the specification of four types of the absorber tubes are given in Table 2. The four tubes tested in this study are the bare tube, bumping tube, floral tube, and twisted floral tube. The floral tube has 11 threads. The twisted floral tube, tilted 5 degrees, is designed to enhance the solution flow and thus to improve the heat transfer performance. The bumping bare tube is similar to the bare tube but its surface is made rough.

In overall, the experimental apparatus is similar to a single-stage absorption unit and designed to deliver 2 USRT (7kW) refrigeration capacity. The overall size of the absorber is 400 mm in length. The absorber tubes are 360 mm in the effective length, 15.88 mm in outside diameter, and 13.88 mm in inner diameter. The horizontal tube bank of 6 rows and 8 tubes in a row has staggered arrangement. Six tray tubes with 3 mm hole at every 40 mm along the tube are located at the top of the absorber. These tray tubes have the typical dimensions that are found in commercial systems. Through the tray tube holes the solution flows down the tube bank by gravity. In the absorber, the cooling water flows in four parallel circuits, each circuit connecting 12 tubes of two rows. Thus in the absorber the cooling water and solution flows form a cross-flow configuration.

3. Calculation of Heat and Mass Transfer Coefficients

3.1 Heat transfer coefficient

To evaluate the heat transfer characteristics in the absorption process, an energy balance in the absorber is formulated. A logarithmic mean temperature difference (LMTD) of a heat exchanger is defined as Eq. (1) and the overall heat transfer coefficient, U, can be written as Eq. (2).

$$\Delta T_{lm}=\frac{\{(T_{ASi}-T_{ACoo})-(T_{ASo}-T_{ACoi})\}}{\ln\{(T_{ASi}-T_{ACoo})/(T_{ASo}-T_{ACoi})\}} \tag{1}$$

$$U=\frac{Q}{\{\Delta T_{lm}\cdot(\pi\cdot d_o\cdot L)\}} \tag{2}$$

The heat transfer coefficient h_o of the solution flowing outside the tube can be obtained from the heat balance of Eq. (3). In this relation the thermal resistance of the tube wall is neglected.

$$\frac{1}{h_o}=\frac{1}{U}-\frac{d_o}{d_i}\cdot\frac{1}{h_i} \tag{3}$$

The film flow rate per unit length, Γ, is defined as

$$\Gamma = G_s/(2L) \tag{4}$$

where G_s is the mass flow rate of the absorption solution. Using the measured temperatures and flow rates of solution and cooling water, the heat transfer coefficient h_o of the solution flowing outside the tube can be calculated from Eq. (1) to (3).

3.2 Mass transfer coefficient

Mass transfer resistance of the refrigerant vapor in the absorption process is considered to exist only between the liquid-vapor interface and absorption solution, assuming the resistance between the liquid-vapor interface and the bulk vapor is negligibly small. Also, assuming the vapor pressure at the liquid-vapor interface is equal to the vapor pressure in the falling film, the LMCD, $\Delta\xi_{lm}$, is defined as Eq. (5) in terms of equilibrium concentration ξ^* and concentration ξ of falling film. On the basis of these assumptions, the mass transfer coefficient, β, is derived as Eq. (6).

$$\Delta\xi_{lm} = \frac{\{(\xi^*_{Asi} - \xi_{Asi}) - (\xi^*_{Aso} - \xi_{Aso})\}}{\ln\{(\xi^*_{Asi} - \xi_{Asi})/(\xi^*_{Aso} - \xi_{Aso})\}} \tag{5}$$

$$\beta = G_R/\{\rho_m \cdot \Delta\xi_{lm}(\pi \cdot d_o \cdot L)\} \tag{6}$$

Here, G_R is the refrigerant vapor absorption rate and the equilibrium solution density, ρ_m, is defined as

$$\rho_m = (\rho_{Asi} + \rho_{Aso})/2 \tag{7}$$

where, ρ_{Asi} is the solution concentration of the absorber inlet, and ρ_{Aso} is the solution concentration of the absorber outlet.

4. Experimental Results and Discussion

The wetting characteristics of the four absorber tubes were first evaluated by flowing water colored with dye over the tubes. Figures 3 and 4 show the results of the wetted rate and wetted area as the flow rate varies. If the wetted area is only considered, low-fin tube is the best although it has a little variation as the flow rate changes. However, the floral tube and twisted floral tube, which have a good wetting behavior and provide thin film flow, can be the most efficient tubes when considering absorption phenomenon, solution hold-up at the tube surface and fabrication cost.

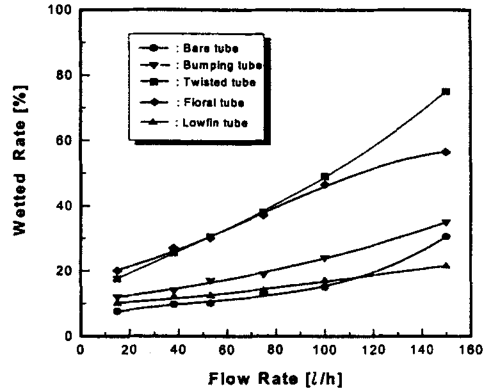


Fig. 3 Wetted rate of each tube vs. solution flow rate.

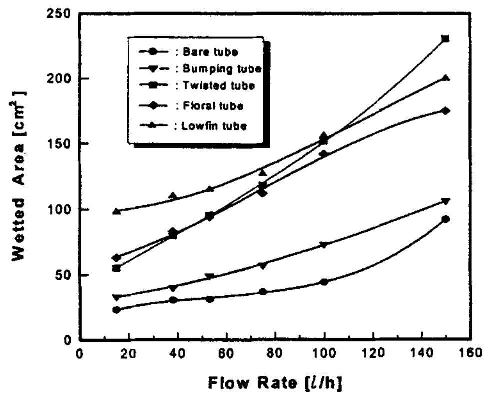


Fig. 4 Wetted area of each tube vs. solution flow rate.

Figure 5 shows the variation of heat transfer coefficients of the four tubes as the absorption solution flow rate changes. For the bare tube, the present data are compared with the data of Furukawa et al. (1993). The agreement in the comparison as shown in Fig. 5 appears good, indicating that the present experimental data are adequately acquired. The heat transfer coefficients tends to show large increase with the absorption solution flow rate, but the slope of increase is gradually reduced in the higher flow rate.

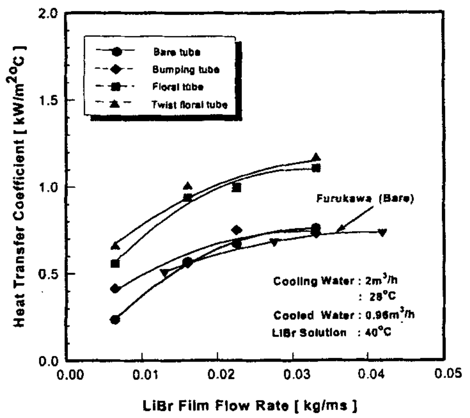


Fig. 5 Heat transfer coefficients vs. LiBr film flow rate.

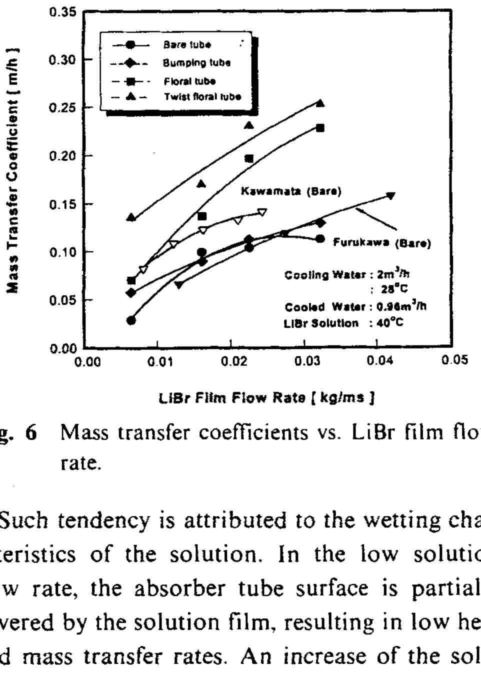


Fig. 6 Mass transfer coefficients vs. LiBr film flow rate.

Such tendency is attributed to the wetting characteristics of the solution. In the low solution flow rate, the absorber tube surface is partially covered by the solution film, resulting in low heat and mass transfer rates. An increase of the solution flow rate causes the wetting area to increase, thus heat and mass transfer rates increase. If the flow rate is further increased, the mass transfer rate is still increased in the same trend. The heat transfer, however, can be retarded by the increased thermal resistance across the thickened solution film. When such adverse effect of increased thermal resistance counteracts the effect of enlarged wetting area at a certain film flow rate, the heat transfer rate gets to be reduced. This observation has been also reported by Kiyota et al. (1996). Therefore, it implies that in designing an absorber it is important to determine an optimum solution flow rate. In the present study, the optimum solution flow rate per unit tube length appears to be 0.03 kg/ms, considering the tendency of slow rise of heat and mass transfer coefficients against the increased film flow rate as well as the pumping power of the solution flow. In the tests with the optimum flow rate of 0.03 kg/ms, it was visually observed through the view port of the test section that the solution delivered through the tray holes was well distributed along the absorber tubes and all the tubes were completely covered with the solution film flow as it flew down the tube banks by gravity.

Four different types of absorber tubes were tested to compare the performance of heat and mass transfer. The twisted floral tube shows the best performance among the four types of tubes. But, the difference in performance between the twisted floral tube and the floral tube was small. The floral tube showed about 40% higher heat and mass transfer performance than the bare tube. The reason for this appears to be that it has a larger surface area and the grooved-like surface can hold the solution and release repeatedly, enhancing the heat transfer and forming relatively larger wetted area. Especially, the twisted floral tube showed a little better performance than the floral tube due to the twisting effect by which the solution flows faster while still contacting on the tube surface. In case of the bumping tube, the heat transfer performance was enhanced little compared to the bare tube. Bumping effect of the rough surface of the tube appears to cause an enhancement in low solution flow rate, but as the film flow rate increases, the effect is soon diminished.

The variation of the mass transfer coefficients as the LiBr film flow rate changes is shown in Fig. 6. As seen in the figure, the present data for bare tube also show a reasonable agreement with the experimental data for bare tube reported by Kawamata et al. (1985) and Furukawa et al. (1993). The floral tube shows the better mass transfer performance than the bare tube. Especially, the twisted floral tube shows the best performance due to its twisted surface which makes the absorption solution fall down easily.

Compared to the bare tube, however, the bumping bare tube shows little improvement in performance.

5. Conclusion

For the purpose of developing high efficiency heat transfer tubes for a absorber in the absorption chiller/heater systems, bare tube, floral tube, twisted floral tube and bumping bare tube were tested experimentally to compare the heat and mass transfer performance. The floral tube showed a 40% higher performance than the bare tube. The bumping bare tube, which surface was purposedly roughened to improve heat and mass transfer, did not show an appreciable enhancement. It is also observed that as the film flow rate increases, the heat and mass transfer coefficients increase markedly in the low range of film flow rates, but in the high range of film flow rate, the slope of increase is reduced. This indicates that an optimum film flow rate of solution exists. Considering the tendency of slow rise of heat and mass transfer coefficients against increased film flow rate as well as the pumping power of the solution flow, it is proposed that an optimum film flow rate per unit tube length is 0.03 kg/ms. Based on the present experimental study, it is also proposed that the floral tube and the twisted floral tube bring an improved heat and mass transfer performance when it is used for absorber tubes in the absorption chiller/heater.

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