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Experimental Study on Heat Transfer Performance of Absorber with Variable Plate Types

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ABSTRACT

An experimental study of the absorption process of water vapor into a lithium bromide solution was performed. For the purpose of developing high performance absorption chiller/hater utilizing lithium bromide solution as working fluid, it is important to improve the performance of absorber with the larger heat transfer area of the four heat exchangers. The experimental apparatus was composed of a plate type absorber which could increase the heat exchange area per unit volume to investigate more detail characteristics instead of the conventional type, that is, horizontal tube bundle type. The size of plate absorbers were made for 0.4m× 0.6m and the design objective of a refrigeration capacity was 1RT. In this experiment, three kinds of plate absorbers namely flat plate, dimple plate and groove plate were used. The obtained results were less than the design objective values, that is, the refrigeration capacity was about 0.3~0.4RT and the overall heat transfer coefficient was 500~600 kcal/m^{2}h℃ at the standard conditions.

NOMENCLATURE

a	: Heat transfer area m^2
G	: Mass flow rate kg/s
h	: Enthalpy kcal/kg
K	: Heat transfer rate kW/(m^2 · ℃)
L	: Length of the plate m
Q	: Refrigeration capacity kcal/h
Re	: Renolds Number
Re_f	: Film Renolds Number
ΔT_{lm}	: Logarithmic mean temperature difference
μ	: Viscosity Pa · s
ξ	: Mass concentration wt%

Subscripts

A	: Absorber

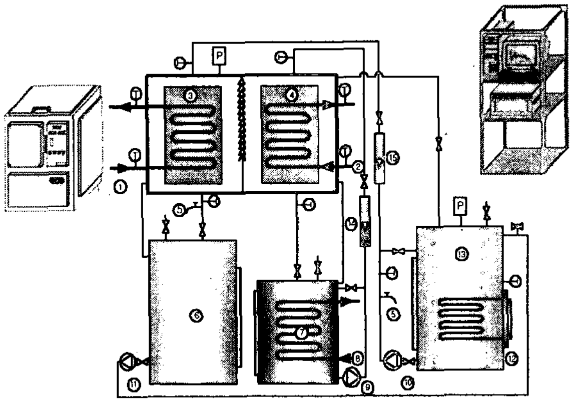
co : Cooling water
i : inlet
o : Outlet
s : Absorbent solution

1. INTRODUCTION

The manifold uses of absorption cooling and heating can efficiently resolve difficulties related to peak-hour-loads and greenhouse effects [1]. The absorption chiller/heater can operate using renewable energy like recycled heat, solar energy and heat from liquid or gas fuel using environment friendly working fluids such as water or ammonia [2]. System performance and the size of the absorption chiller/heater is the growing concern due to its lower COP and bigger size than the vapor compression type [3]. The heat and mass transfer coefficients are relatively low due to the poor mass transfer processes on the liquid side. In Korea, the absorption chiller/heater, which has a capacity of more than 30RT and uses H_2O/LiBr as a refrigerant/absorbent has been developed for commercial purposes[4-13]. In this study, flat type, dimple type and groove type plate heat exchangers are used to exchange heat between liquids which themselves utilize corrugated/bend surfaces and thus provide secondary flow and mixing to produce high heat transfer performance. Packaging for reducing the size and applying different types of plates for improving the heat transfer performance got equal importance in this experiment.

2. EXPERIMENTAL APPARATUS

Fig. 1 shows the experimental apparatus that is composed mainly of absorber, evaporator, strong solution tank, generator, weak solution tank, refrigerant tank, heater and tubes. One side of absorber is mounted with a large glass plate for clear viewing and the other side with the cooling water part. The heat transfer copper plates made in different geometric configurations are placed between the cooling water and LiBr solution. To make sure that the LiBr solution flows smoothly, a solution distributor is mounted in the absorber. A cooling tower is used to remove the heat from condensation and absorption process. A pump of 2HP/5kW has been installed to circulate the cooling water. Another pump of 3HP/8kW is used at the absorber. Fig. 2 shows the experimental apparatus used, Table 1 gives the experimental condition. Two constant temperature baths of capacities 3HP and 2HP are supporting the absorber and Evaporator respectively. Fig. 3 shows the plates used in our experiment. The size of the flat plate is 600mm in length and 400mm in width. The heater in the solution circulating pump has a heating capacity of 3kW.



① Cooling water ② Chilled water ③ Absorber ④ Evaporator ⑤ Sampling valve ⑥ Weak solution tank ⑦ Refrigerant tank ⑧ Chiller ⑨ Water pump ⑩ Solution pump ⑪ Solution circulation pump ⑫ Heater ⑬ Strong solution tank ⑭ Refrigerant flowmeter ⑮ Solution flowmeter Ⓟ Manometer Ⓣ Thermometer

Fig. 1 Schematic diagram of experimental apparatus

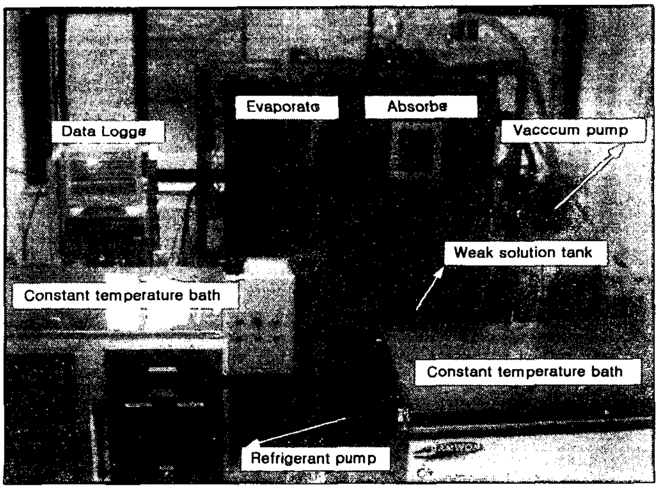
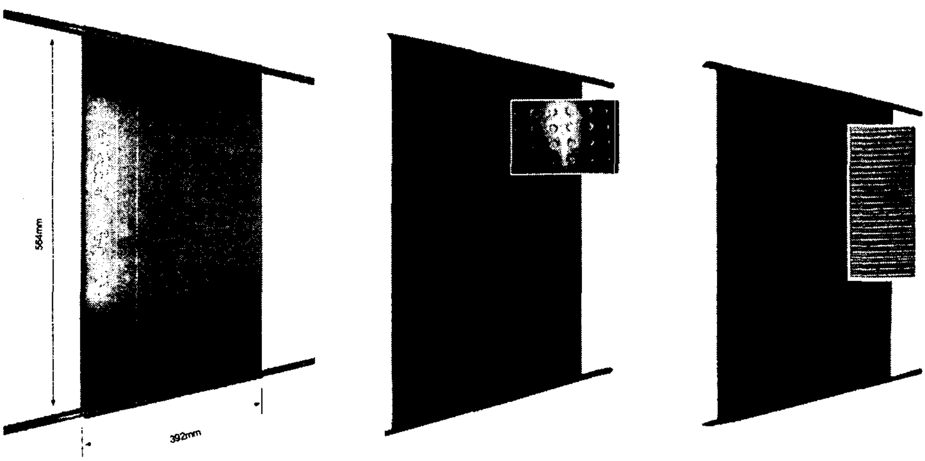


Fig. 2 Experimental Apparatus.

Table 1 Experimental conditions

Absorber	**Pressure, P [kPa]**	**1.2**
LiBr solution	**Inlet temperature, T_{si} [℃]**	**49**
	Inlet concentration, ξ_{si} [wt%]	**60**
	Film Reynolds No., Re_f [-]	**6~35**
Cooling water	**Inlet temperature, T_{wi} [℃]**	**32**
	Flow rate, G_w [kg/s]	**0.17~0.3**



(a) Flat type (b) Dimple type (c) Groove type

Fig. 3 Heat exchanger of plate absorber

3. EXPERIMENTAL METHOD

To perform this batch type experiment in stable state, it was divided into processes of establishing experimental conditions, performance measuring and solution generating. The absorbent solution and refrigerant flow rates are set to specified values by adjusting control valves. The conduct of absorbent solution was observed w.r.t. temperatures, solution flow rates and cooling water flow rates. A data logger and a personal computer was used to process the data related to temperature, cooling water flow rate and solution flow rate.

3.1 Calculation method

An absorption model was established to evaluate heat transfer in an absorption process. The heat added to cooling water in the absorber can be written as

$$Q_{co} = G_{co} c_p (T_{coo} - T_{coi}) \tag{1}$$

The heat of absorption solution given by the difference between inlet and outlet at the absorber can be expressed as

$$Q_s = G_{si}\ h_{si} - G_{so}\ h_{so} \tag{2}$$

Here, solution enthalpy of inlet and outlet h_{si}, h_{so} can respectively be defined as follows

$$h_{si} = h(T_{si},\ \xi_{si}) \tag{3}$$

$$h_{so} = h(T_{so},\ \xi_{so}) \tag{4}$$

Refrigeration capacity Qr of this apparatus can be written as

$$Q_r = Q_{co} - Q_s \tag{5}$$

The overall heat transfer coefficient K and the logarithmic temperature difference respectively can be written as

$$\Delta T_{lm} = \frac{\{(T_{Asi} - T_{Acoo}) - (T_{Aso} - T_{Acoi})\}}{\ln\{(T_{Asi} - T_{Acoo})/(T_{Acoo} - T_{Acoi})\}} \tag{6}$$

$$K = \frac{Q_{co}}{\Delta T_{lm} \cdot a} \tag{7}$$

The film Reynolds number and the mass flow rate to solution flow rate per unit length is respectively determined as

$$Re_f = \frac{4\Gamma_s}{\mu_{si}} \tag{8}$$

$$\Gamma_s = \frac{G_{si}}{2L} \tag{9}$$

Here G gives the mass flow rate of absorbent.

4. EXPERIMENTAL RESULT AND DISCUSSION

To measure the characteristics namely heat flux, overall heat transfer coefficient and

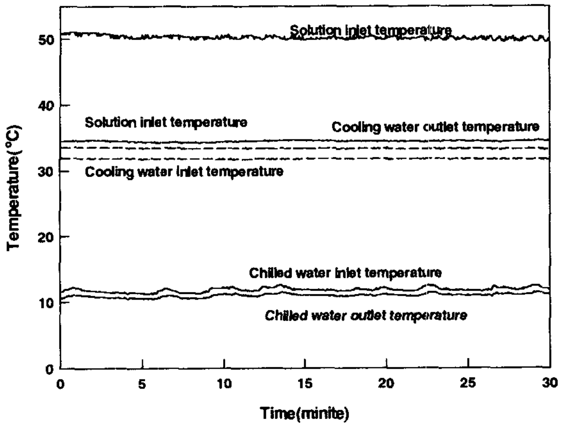


Fig. 4 Temperature of fluid inlet and outlet at steady state

refrigeration capacity with respect to chilled water inlet/outlet temperature, experiment was conducted under the experimental conditions given in table 1. The temperature of fluid inlet and outlet at steady state is shown in Fig. 4. The figure has been produced colleting data from data logger and these data explains the performance of the absorber. The data logger recorded the data every 5 seconds automatically. Fig. 5 gives the heat balance of experimental apparatus. From this figure, we see that, the heat balance is ± 15%. We choosen it considering the fact that heat has been lost through the boundary surface of the device followed by the pipes used.

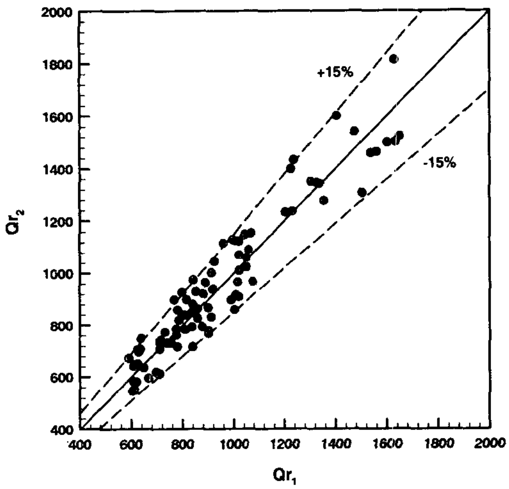


Fig. 5 Heat balance of experimental apparatus

4.1 Characteristics by solution flow rate

Fig. 6 shows the influence of solution flow rate on heat flux. We can probably claim from Fig. 8 that heat flux of every plate is almost directly proportional to the film Reynolds number. At a flow rate of 18 l/min, the heat flux for flat plate was lowe than both dimple and groove plates but for the groove plate, it didn't differ that much from dimple one. At the maximum solution flow rate, the surface contact/wet ratio was found to be less than 60% through the side glass.

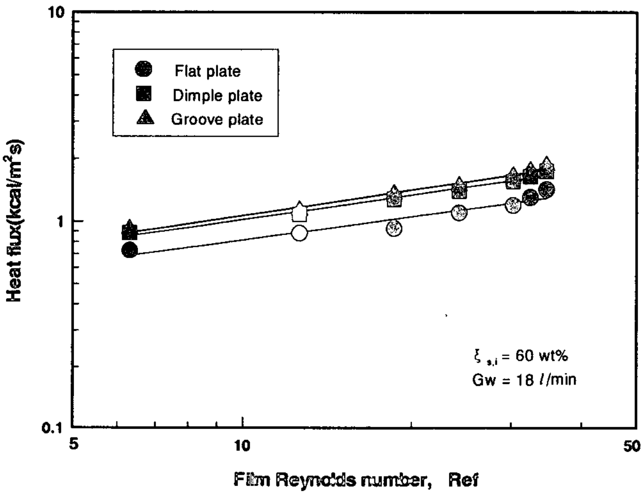


Fig. 6 Influence on heat flux by solution flow rate

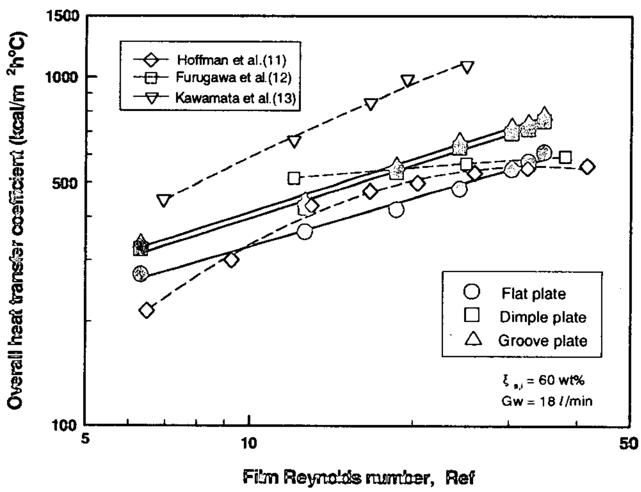


Fig. 7 Overall heat transfer coefficient of plate absorber on solution flow rate

Fig 7 gives the comparative overall heat transfer coefficient of plate absorber by solution flow rate from our obtained data and those performed by other researchers. The results are slightly different but not the difference is not that significant. The difference occurred mainly due to the difference in the experimental condition, the shape/geometrical structure of the absorber followed by the arrangement of the device.

For flat plate absorber, the overall heat transfer coefficient increases with the increase of the solution flow rate. In the standard experimental condition of Re_f=18, the overall heat transfer coefficient for flat plate was around 500~600kcal/m^{2}h℃. The results for dimple and groove are around 25% and 30% higher than that of the flat plate respectively.

Fig. 8 gives the refrigeration capacity of the absorber with respect to solution flow rate for all three plates. The trend is seen to be increasing with the increasing flow rate. For the flat plate at Re_f=18, the refrigeration capacity is found to be around 800~1,200kcal/h which is about 30~40% of our expected designed objective. The seemingly decrepit result is probably due to poor contact/wet ratio. The refrigeration capacity for dimple and groove plates were 50% and 60% higher than that of the flat plate respectively. The reason for this remarkable increase in cases of dimple and groove is that the heat transfer area have increased for both these plates and the mixing was better as well.

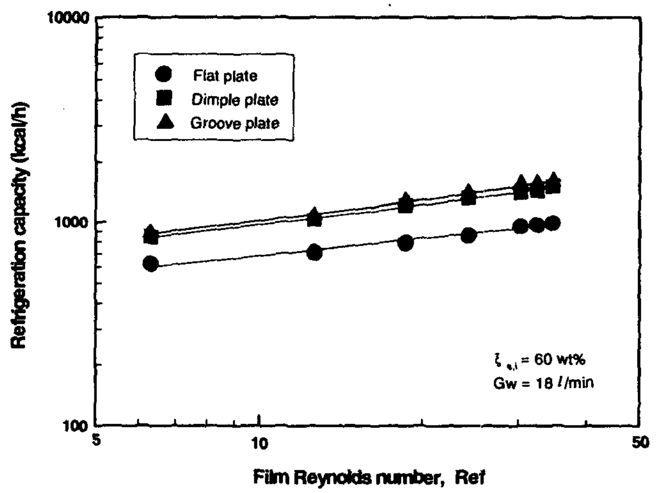


Fig. 8 Refrigeration capacity of plate absorber on solution flow rate

4.2 Characteristics by cooling water flow rate

Comparative heat flux with respect to cooling water flow rate has been shown in Fig. 9 at the standard experimental condition. The heat flux increases with the increase of the cooling water flow rate as can be clearly seen. This could be due to the fact that the increase of the cooling water induces an increase of the heat transfer coefficient in the cooling water part of the absorber. When the cooling water flow rate is changed like 10, 15, 18 l/min, then the heat flux increased around 30~50% for all the plates. The heat fluxes for the dimple and groove plates is found to be around 50% higher than that of the flat plate. The groove type plate's performance was similar to that of the dimple one but not exactly identical. The reason is once again the increase of the heat transferring area and better mixing/wetting for the heat transferring surfaces for both dimple and grove plates. Since we used the same dimensional area for all the plates, so the results obtained for the dimple and flat plates could be a bit over estimated. To measure the extended surface areas of dimple and groove is a bit troublesome, so we used the basic dimension for all the plates. It is assumed as default that (5-10)% of the area could be increased for the case of dimple and groove plates.

Fig 10 gives the refrigeration capacity with respect to cooling water flow rate. It shows that the cooling capacity increases with the increase of the cooling water flow rate. The performance of the groove plate was the best.

Fig 11 shows the overall heat transfer coefficient with a vailable cooling water flow rate. It can be seen that the overall heat transfer coefficient increases almost linearly with the increase of the cooling water flow rate. The overall heat transfer coefficient of the groove plate is found to be the most in this case as well.

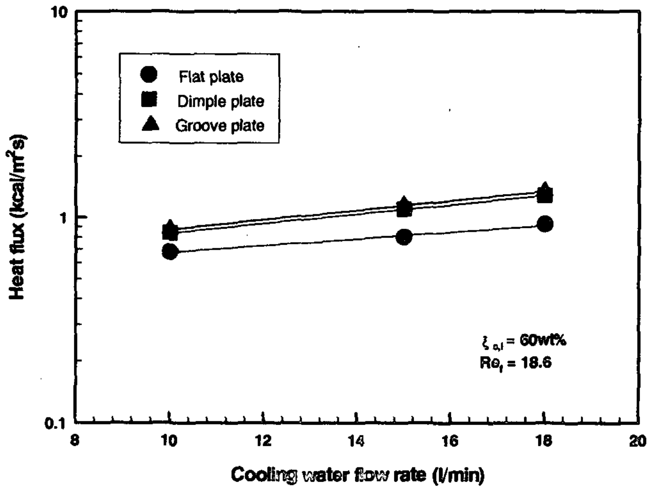


Fig. 9 Heat flux of Plate Absorber on Cooling water flow rate

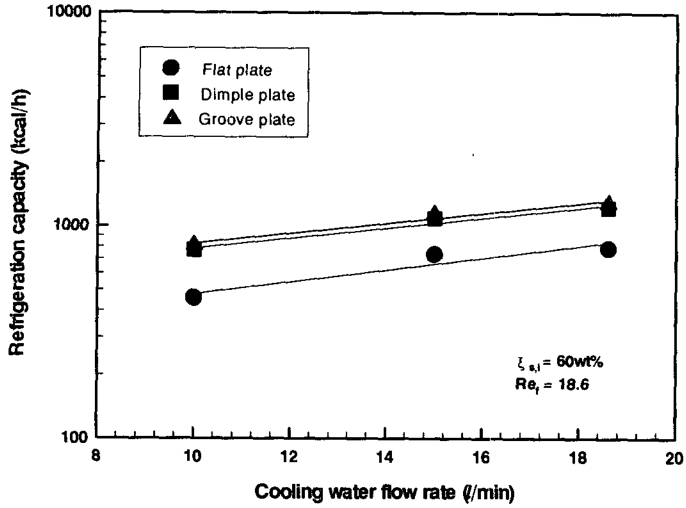


Fig. 10 Refrigeration Capacity of Plate Absorber on Cooling water flow rate

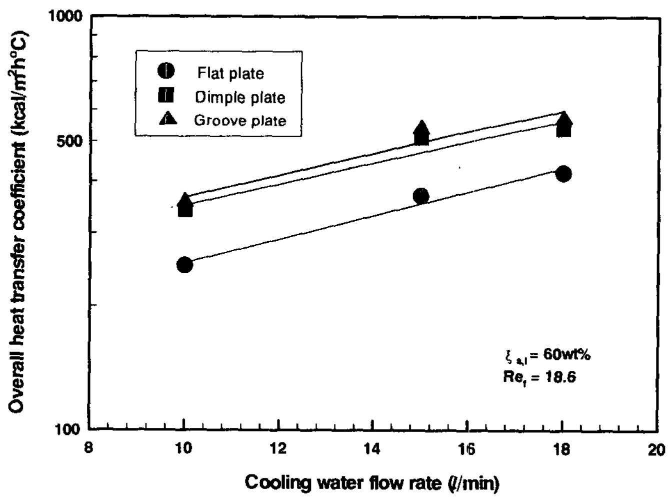


Fig. 11 Overall heat transfer coefficient of Plate Absorber on Cooling water flow rate

5. CONCLUSION

Following conclusions can be drawn based on results of batch type absorption experiment of different types plate absorber using standard design capacity of 1RT.

1. In a standard condition of our experiment, the heat transfer coefficient for flat plate is found to be about 500~600kcal/m^{2}h℃ and the dimple and groove plates heat transfer coefficients are 25~30% higher than that of flat plate with respect to total

heat transfer coefficient.

2. In case of a flat plate absorber having dimension of 0.6 m by 0.4 m, the cooling capacity is about 0.3~0.4RT and that the shows an increasing trend with the increase of solution flow rate.
3. Although the refrigeration capacity of the dimple and groove plates are about 50% more than a flat plate, but yet they failed to reach the expected objective of the experiment and the method of changing the shape of the plate has a limit as well in improving the cooling capacity. Therefore we are required to use even better and active methods like one including additional aquaas solution for improving heat transfer.

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