HEAT AND MASS TRANSFER IN THE VERTICAL TWIN-PARALLEL-TUBE ABSORBER-EVAPORATOR UNIT

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ABSTRACT

A vertical twin-parallel-tube absorber-evaporator unit was built for the present study to investigate the overall heat transfer and mass absorption rates at various operation conditions. In order to establish the optimal surface wetting condition for the thin absorbent film during operation process, an annular-slot type solution distributor with declining surface was adopted for the present system. The main objective of the present study is to perform an experimental investigation of the controlling parameters associated with absorption by falling film, including the operation pressure, the inlet temperature of the cooling water, and the Reynolds number, temperature, and concentration of the inlet absorbent solution. An analogy between heat and mass transfer near the film surface is then applied to calculate the interfacial concentration and temperature of the absorbent film, and thus the heat and mass transfer coefficients of the absorbent film is determined. The present results show that, the higher the inlet Reynolds number of the absorbent solution is, the lower the total heat transfer rate is. Increasing operation pressure will increase the concentration difference of the absorbent solution. The concentration difference of absorbent solution film between inlet and outlet increases sharply as the Reynolds number of the inlet absorbent film decreases below 200. When the Reynolds number of the inlet absorbent film is around 200, optimal absorption-evaporation exists and will provide appropriate conditions for a high efficiency absorption air conditioning system.

I. INTRODUCTION

With features such as no chlorofluorocarbons released from the operating system, and capable of being driven by low-power energy sources like waste heat or solar energy, the absorption air conditioning system has regained attention recently. Compared with a vapor-compression system, an absorption

system may or may not be preferred depending on the efficiency of the absorber-evaporator unit.

A falling film absorber-evaporator unit has the distinct advantage of high transfer coefficients at low driving forces with absorption of saturated vapor taking place at the free interface on a falling liquid film, even under the condition of a negative temperature gradient (Brauner *et al.*, 1989). Consequently, a

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Fig. 1 Schematic description of the vertical twin-parallel-tube absorber-evaporator

newly developed vertical absorber-evaporator unit with a calcium chloride solution working fluid is proposed in the present study to show the merit of a falling film absorption mechanism, as depicted in Fig 1. A significant breakthrough of the system is the adoption of a vertical twin-parallel-tube absorber-evaporator, not the single vertical tube experimental unit previously used by Yang (1993) and Morioka (1993), as shown in Fig. 2.

An absorption unit constructed of a single vertical tube, as shown in Figs. 2(A), 2(B), and 2(C), introduces some problems: (1) The vapor stream spurting into the absorber with at high velocity causes the appearance of film rupture, owing to the great collision force of the vapor stream with the absorbent film and the increase of the critical flow rate for the absorbent film to avoid breakdown; (2) As the velocity of the vapor stream is much too high, it forms a great shear stress on the free surface of the absorbent film. Consequently, the unwanted wavy-film phenomenon may be induced easily; (3) The

Fig. 2 Schematic description of the comparison between single-tube and twin-tube absorption

absorption potential of absorbent film is strong at the section near the outlet of the vapor supply pipe, and is weak away from the outlet. Non-uniform absorption therefore results for the single vertical tube absorber.

On the contrary, the present absorption system, with a vertical twin-parallel-tube absorber-evaporator unit, as shown in Fig. 2(D), can avoid most of these problems. The advantages are: (1) The crossing area, through which the vapor passes through the absorber equals the inner diameter (0.23m) of the absorber multiplied by the whole wetting length (1.2m) of the vertical tube. With such a large crossing area, the vapor stream velocity is reduced to 0.15m/s, and the collision force on the absorbent film is very small; (2) The direction of the absorbate stream is perpendicular to the direction of the falling absorbent film, and the absorbate stream velocity is small. As a result, the effect of the shear stress driven by the absorbate flow on the vapor-liquid interface is small; (3) The absorbate vapor is uniformly

Fig. 3 The enthalpy-concentration chart for Lithium-Bromide/water solution

Fig. 4 The enthalpy-concentration chart for Calcium-Chloride/water solution

generated from the pure water film of the whole evaporating tube and is directly absorbed by the absorbing tube in the presented unit. It results in a totally uniform absorption; (4) The provided absorbate vapor is uniformly generated by an evaporating tube with pure water film falling on its outer surface, where the heat of evaporation is provided by heating the flow of chilled water inside this tube. The desired operating pressure is then attained by adjusting the flow rate of the pure water film, the temperature and the flow rate of the chilled water, and therefore, can be strictly and stably controlled.

The heat and mass transfer efficiency of the present vertical twin-parallel-tube absorber-evaporator unit are greater than the single-vertical-tube absorbers. The data capturing condition is also more stable in the present system. The only experiment in this study that could not be carried out on the twin tube evaporator-condenser was the single absorption experiment.

Enhancement of vapor absorption on the liquid film can be achieved not only by the wavy-motion of a falling liquid film, but also by using a porous medium for the surface of the absorber wherein falling liquid film flows. The heat and mass transfer for the absorption process of a falling liquid film flow on a porous medium has been modeled by the

Brinkman-Forchheimer's extended Darcy equation. Their numerical solution shows that the transport rates are higher than the absorption process of falling liquid film on a smooth surface. Yang and Jou (1995) reported that the mass absorption rate increased with the increasing solution flow rate, the conductivity ratio, and the absorber pressure. The mass absorption rate is optimum with surface medium porosity of 0.9 and conductivity ratio of 10.

Thermal characteristics of the absorbent solution determine the amount of energy supplied to the generator and the efficiency of an absorption heat pump. In order to upgrade the coefficient of performance (C.O.P.) of the system, a better absorbent solution must always be considered. Since the integral heat of mixing of the calcium chloride solution is lower than that of the lithium bromide solution in the concentration range of 50%~60% normally used for commercial absorption system, as shown in Fig. 3 and Fig. 4, the amount of heat required to recover the concentration of the weak lithium bromide solution is greater than that of the calcium chloride solution. Hence the C.O.P. of the calcium chloride solution absorption system is greater than that of the lithium bromide for the same refrigerating capacity. Therefore, calcium chloride solution is used as the absorbent solution in the present study.

The heat and mass transfer characteristics for the non-isothermal vertical twin-parallel-tube absorber-evaporator are of major concern in evaluating the overall performance of an absorption air conditioning system. The difficulty in calculating the overall heat and mass transfer coefficients is to accurately determine the interfacial temperature and concentration of the absorbent liquid film.

In the work of Yih (1982), experiments on isothermal film absorption of different gases were conducted by using the viscosity-damped turbulence model. Their resultant correlations were

$$Sh/Sc^{0.5} = 2.995 \times 10^{-2} Re^{0.2134} \quad (1)$$

$$300 < Re \leq 1600$$

$$Sh/Sc^{0.5} = 9.777 \times 10^{-2} Re^{0.6804} \quad (2)$$

$$1600 < Re \leq 10500$$

Since uncertainties exist in determining the interfacial temperature and concentration of the absorbent film, descriptions of transfer phenomena near the film surface, especially in the turbulent flow region, are not yet conclusive in the literature (Brauner *et al.*, 1989; Grossman *et al.*, 1983; Lamourelle *et al.*, 1972; Mudawwar *et al.*, 1986; Yih *et al.*, 1982).

For the non-isothermal absorption process, Yuksel and Schlunder (1987) directly measured the interfacial temperature of absorbent film absorbing water vapor by means of an infrared pyrometer. It was shown that the assumption of un-damped turbulence near the film surface might be improper. Furthermore, they proposed a method to iteratively calculate the temperature and concentration of the liquid surface, based on the concept of analogy between heat and mass transfer near the interface. The heat and mass transfer coefficients obtained by these methods agreed well with those determined by direct measurement. Many theoretical models of heat and mass transfer for the absorber have been developed and many laboratory experiments have been performed, (Brauner *et al.*, 1991; Grigoreva *et al.*, 1977; Matsuda *et al.*, 1994; Nakoa *et al.*, 1987; William *et al.*, 1994), but the results and explanations are inconsistent and partially contradictory. More often than not, the heat transfer coefficient is obtained through personal assumption.

In order to investigate the performance characteristics of this new vertical twin-parallel-tube absorber-evaporator unit, a simulated experimental system was built to carry out experiments under various operating conditions. With precisely measured operating pressure, the temperatures and concentrations

of absorbent solution at inlet and outlet, the temperature of the cooling water, the inlet Reynolds number of absorbent film, and the determined overall heat and mass transfer coefficients of the present system are thereby evaluated to understand the mechanisms of heat and mass transfer in such a falling liquid film absorber-evaporator unit.

II. DATA REDUCTION

The key to understanding the characteristics of heat and mass transfer of the non-isothermal absorption process in the vertical tube is to precisely determine the interfacial temperature and concentration of the liquid-vapor interface.

When vapor is absorbed by the absorbent solution, it releases latent heat and is condensed to liquid at the interface. The condensed water will dilute the absorbent solution, and the released heat increase its temperature, especially at the liquid-vapor interface. Both effects will increase the equilibrium vapor pressure of absorbent solution and reduce the vapor solubility. Besides, boundary layers, of concentration and temperature, will be formed in the absorbent solution film due to the condensed water. That means heat and mass transfer are simultaneously happening in the absorbing process.

Chen and Sun (1997) proposed methods of evaluating the temperature and concentration at absorbent solution interfaces. As shown schematically in Fig. 5, the absorbent film on the surface outside the vertical tube is taken as control volume for analyzing the heat and mass transfer effect. The absorbent solution of mass flow rate, M_i , concentration, C_i , and temperature, T_i , enters the solutions distributor, and the saturated absorbent pressure corresponding to C_i and T_i is P_i . As for the outlet absorbent solution of mass flow rate, M_o , concentration, C_o , and temperature, T_o , the saturated absorbent pressure corresponding to C_o and T_o is P_o .

The vertical twin-parallel-tube absorber-evaporator is filled with vapor of the pressure P_v . The saturated temperature corresponding to P_v is T_v , which is lower than the inlet absorbent film temperature T_i . However, the relatively cold vapor is absorbed and condensed by the absorbent film, provided P_v is larger than the absorbent vapor pressure P_i (and P_o), that is, $P_v \geq P_o > P_i$.

Consider the annulus region of thickness dz , as shown in Fig. 5. As vapor is absorbed, the heat of absorption, Q_a , is released at the film surface. Therefore, there are mass flow rate, M_a , and heat flux, Q_a , of the absorbed water transporting from the film surface to the surface of the vertical tube simultaneously. Temperature and concentration boundary layers based on the heat and mass transfer near the

Fig. 5 Physical diagram of the model of falling film

interface are formed, then.

The bulk temperature, T_B , and bulk concentration, C_B , of the absorbent film are defined by

$$T_B = \frac{\int_0^\delta u T dy}{\int_0^\delta u dy}; \quad C_B = \frac{\int_0^\delta u C dy}{\int_0^\delta u dy} \quad (3)$$

and $u=u(r)$ is the velocity distribution function of the absorbent film.

Near the outer-face, the heat transfer coefficient η_f is defined by

$$\eta_f = \frac{q_a}{(T_f - T_B)} \times \frac{1}{E_T} \quad (4)$$

where q_a and Achermann correction factor E_T , (Bird et al., 1965), are given by

$$q_a = m_a(h_v - h_w) \quad (5)$$

$$E_T = \frac{(h_v - h_w)}{c_{pw} \times (T_f - T_B)} \times \ln \left\{ 1 + \frac{c_{pw} \times (T_f - T_B)}{(h_v - h_w)} \right\} \quad (6)$$

Here, h_v is the enthalpy of water vapor at its temperature, T_v , and pressure, P_v , h_w is that of

condensed water at the interfacial temperature, T_f , and C_{pw} is the specific heat of liquid water.

The mass transfer coefficient β_f is defined by regarding the uni-direction mass transfer as follows

$$\beta_f = \frac{m_a}{\rho(C_B - C_f)} \quad (7)$$

where ρ is the density, and C_f is the interfacial concentration of the absorbent solution. At the thermodynamic equilibrium condition, $C_f = C_f(T_f, P_v)$.

In the experimental work of Yüksel and Schlunder (1987), T_f was found to be larger than T_b . They also concluded that turbulence was damped in the vicinity of the interface due to surface tension or viscosity. Based on this finding, an analogy between heat and mass transfer can be applied to this region:

$$\frac{\eta_f}{\beta_f} = \rho c_p Le^n \quad (8)$$

where c_p is the specific heat of the absorbent solution, and exponent n has a value between 0.5 and 0.6 depending on the experimental results.

The following equation can be derived from Eqs. (4) to (8):

$$(C_B - C_f) = \frac{c_p}{c_{pw}} Le^n \ln \left[1 + \frac{c_{pw}}{(h_v - h_w)} (T_f - T_B) \right] \quad (9)$$

By using Eq. (9) and $C_f = C_f(T_f, P_v)$, T_f and C_f can be calculated iteratively with the known bulk temperature of the absorbent, T_b , the bulk concentration of the absorbent, C_b , and the corresponding physical properties. This yields an easier way to determine T_f and C_f when there is difficulty measuring them directly.

Based on the mass balance, as shown in Fig. 5, the vapor absorption rate M_T can be calculated by

$$M_T = M_o - M_i \quad (10)$$

The flow rate of CaCl_2 carried by the absorbing film is constant. That is

$$M_i C_i = M_o C_o \quad (11)$$

$$M_o = M_i C_i / C_o \quad (12)$$

From Eqs. (10) to (12), the total vapor absorption rate can be calculated with inlet flow rate, inlet concentration, and outlet concentration; therefore,

$$M_T = M_i (C_i / C_o - 1) \quad (13)$$

The total heat transfer rate, Q_T , transferred through the tube wall into the cooling water can be

calculated by energy balance,

$$M_i h_i + M_T h_v = M_o h_o + Q_T \quad (14)$$

where h_i and h_o are the enthalpies corresponding to the inlet and outlet absorbent solution respectively. By using Eqs. (13) and (14), Q_T can be expressed as:

$$Q_T = M_i [(h_i - h_v) - C_i / C_o (h_o - h_v)] \quad (15)$$

On the other hand, total heat transfer rate, Q_T , must equal the heat taken by the cooling water,

$$Q_T = M_c (T_{co} - T_{ci}) C_{pw} \quad (16)$$

Overall heat transfer coefficient, U , of the absorber unit, based on the logarithmic mean temperature difference, ΔT_{lm} , between the inlet and outlet conditions is defined as,

$$U = \frac{Q_T}{(\pi d L) \Delta T_{lm}} \quad (17)$$

with

$$\Delta T_{lm} = \frac{(T_{ei} - T_{ci}) - (T_{eo} - T_{co})}{\ln \left(\frac{T_{ei} - T_{ci}}{T_{eo} - T_{co}} \right)} \quad (18)$$

The mass-transfer driving potential of the absorber unit is defined in terms of the logarithmic mean concentration difference ΔC_{lm} . Then, the overall mass-transfer coefficient, N , is defined as,

$$N = \frac{M_T}{\rho (\pi d L) \Delta C_{lm}} \quad (19)$$

with

$$\Delta C_{lm} = \frac{(C_i - C_{fi}) - (C_o - C_{fo})}{\ln \left(\frac{C_i - C_{fi}}{C_o - C_{fo}} \right)} \quad (20)$$

where C_i and C_o are the bulk absorbent concentration at the inlet and the outlet, and C_{fi} and C_{fo} are the interfacial absorbent concentrations correspondingly.

To sum up, M_i , M_o , T_i , T_{wi} , and T_{wo} are determined from measurements in the experiments; h_i , h_o and h_v can be obtained from a property table; C_i and C_o are functions of the density and temperature of the absorbent solution at inlet and outlet; and C_{fi} and C_{fo} can be iteratively calculated from the equilibrium condition at the vapor-liquid interface using $C_f = C_f(T_f, P_v)$ and Eq. (9). Then the overall heat transfer rate, vapor absorption rate, heat transfer coefficient, and mass transfer coefficient can all be calculated by Eqs. (13), (15), (17) and (19).

III. EXPERIMENTAL METHODOLOGY

1. Experimental Apparatus

The primary components of the experimental system include the combined absorber-evaporator unit, the cooling water system, the vapor generation system, the supply tank for strong solution, the return tank for weak solution, the water storage tank, the vacuum-generating system, the pumping system, and the measuring devices. The system operates in a batch mode. The schematic diagram of the experimental system is shown in Fig. 6.

The absorber-evaporator unit consists of a cylinder with two parallel tubes inside. Two stainless steel end-plates are mounted on both ends of the cylinder by six treated steel rods, and O-ring seals are placed between the end-plates and the cylinder. The two stainless steel tubes passed through the end plates are held vacuum-tight by compression type O-ring sealed fittings welded on the end plates. Outside diameter of the inner stainless steel tube is 19mm. The total length of the tube is 1.7m, and the net test section length of the tube is 1.2 m. The outer cylinder is made of a 1.2m long Pyrex glass tube with outside diameter 250mm and thickness 10mm. The Pyrex tube is for maintaining the vacuum pressure in order to facilitate flow absorption. The falling films are established on the outer surfaces of both stainless steel tubes, while cooling water flows upwards inside the absorbing tube to remove the heat of absorption and chilled water flows upwards inside the evaporating tube to supply the heat of vapor generation.

The solution distributor mounted on the upper end of the vertical stainless tube is constructed of Plexiglas with an annular slot distributor head accurately machined to have a radial clearance between it and the stainless tubes as shown in Fig. 7. In order to utilize the elasticity of Plexiglas to compress the distributor on the outer surface of the stainless tube, the inner diameter of the distributor is designed slightly smaller than the outer diameter of the stainless tube. Thus, it is easy to assemble and fix the distributor on the upper end of the absorbing tube or evaporating tube to avoid slipping. The purpose of the annular region I, as shown in Fig. 7, is to interrupt the heat transfer between the solution and the vertical tube and to maintain the temperature of the solution as stable as possible before the falling film forms. The absorbent solution or evaporating water is introduced through the solution inlet A, as shown in Fig. 7, so as to eliminate possible disturbances. The solution does not flow directly to the distributor head but through the annular cavities of region B, D, F, and H until they have been filled. Eventually, the solution flows through the annular slot of the distributor head where

Fig. 6 Schematic diagram of experimental system

it forms a uniform film flowing smoothly along the outer surface of the stainless tube.

The circulating system of the absorbent solution consists of a strong solution tank, a weak solution tank, a mass flow-meter, and a circulation pump. The strong solution tank is sealed so that it can be de-aerated by the vacuum pump. The strong solution storage tank and the water storage tank are equipped with cooling coils of 2 refrigeration tons capacity. In addition, the strong solution storage tank is also equipped with a 9kW-heating element to regenerate the weak solution after all tests are done. Temperature of the inlet absorbent solution is controlled by

the heating coil. The strong absorbent solution and the evaporating water are driven to their individual film distributor by a magnetic-driven pump. After absorbing the water vapor, the weak absorbent solution is collected at the bottom of the absorber-evaporator unit by a funnel, and then drained into the weak solution storage tank by gravity. The storage tanks and the piping system are made of 304 stainless steel to prevent corrosion from the absorbent solution.

The water vapor supplied for absorption is generated by the evaporation of the water film falling on the outer surface of the evaporating tube. The heat of evaporation is supplied by the chilled water inside

Fig. 7 Schematic description of the absorbent solution film distributor

this tube. Deionized water serving as the absorbate is de-aerated by heating the water in the storage tank and then evacuated by the vacuum pump.

The cooling water in the absorbing tube and the chilled water in the evaporating tube are supplied by circulating pumps from the isothermal tanks, and the isothermal tanks control the temperatures of the cooling water and the chilled water.

A Type RA0100B543 BUSH Company vacuum pump is used as the vacuum system. Three Type MD-32R-220N magnetic-driven Iwaki pumps are applied in the piping system to introduce strong absorbent solution and pure water for evaporation, and circulating weak absorbent solution. Two rotary pumps are applied in the piping system for cooling water and chilled water. Rotameters are used to monitor the

liquid flow rates.

The density and mass flow rate of the strong absorbent and weak absorbent solution are measured by a digital density sensor and a mass flow-meter from Micro Motion Company (Type DS040H124SU). The meter is based on the mechanical oscillator technique with a built-in digital thermometer for temperature measurement. The error of the digital density meter is within 10^{-4} . From the relationship among density, temperature, and concentration of calcium chloride solution, the absorbent concentration can be inferred by the measured density and temperature (Wimby, 1994).

The temperatures of the system at various locations are measured by K-type thermocouples. Data are recorded by the HR 1300 data logger of the YOKOGAWA Company. The error is within $\pm 0.5\%$ of reading value. Two MSK Company absolute pressure gauges of variable-capacitance type are installed in the absorber-evaporator unit to measure the absolute vacuum pressure, with operation range from 0 to 100 torr and accurate to within $\pm 0.5\%$ of the reading value.

2. Experimental Process

The experimental system is evacuated by the vacuum pump before testing. Meanwhile, the absorbent solution in the strong solution tank is heated to regenerate the solution and to reduce the solubility of the non-absorbable gas in it. When the desired concentration is obtained, the strong solution tank is evacuated by the vacuum pump again for de-aeration and then cooled by the cooling coils until the desired absorbent inlet temperature is obtained. The water storage tank is also evacuated by the vacuum pump for de-aeration and is cooled by the cooling coils until the desired temperature is reached.

The actual experiment commenced with the establishment of the film flows. The absorber-evaporator unit is evacuated down to the desired pressure by the vacuum pump and the absorbate water vapor generated inside this unit is initiated. The absorbent solution flow is then introduced into the distributor at a high flow rate in order to flush the vertical tube surface. When the surface is thoroughly wet, the throttle valve and the valve of the by-pass circuit are adjusted to obtain the desired solution flow rate. The desired operation pressure is attained by heat added to the chilled water flowing inside the evaporating tube. The generated water vapor is absorbed by the falling film of the absorbent solution and the heat of absorption is transmitted to the cooling water flowing inside the absorbing tube.

With the absorbent film established at the proper film Reynolds number and the total pressure of the

Fig. 8 Effect of operating pressure level on the absorbent solution concentration difference

Fig. 9 Effect of operating pressure level on the overall absorption rate

absorber-evaporator unit constant, the other controlling variables are monitored and adjusted accordingly. When all of the control variables reach the desired values and a steady state, data collection commences.

IV. RESULTS AND DISCUSSIONS

1. Effect of Operating Pressure on the Absorber-evaporator Unit

The difference between the operating pressure, P_v , and the equilibrium saturated vapor pressure P_f of the absorbent solution at the liquid-gas interface is the actual driving potential of the present absorption process. P_f is assumed to be equal to the inlet equilibrium saturated vapor pressure P_i (T_i , C_i) corresponding to the inlet absorbent bulk temperature and bulk concentration. The cold vapor generated from the evaporating tube is absorbed and condensed by the warmer absorbent film which is provided by the positive driving potential, ΔP ($\Delta P = P_v - P_i$). ΔP will decrease as the absorption process commences.

The effect of different operating pressures, P_v , on the concentration difference and the heat and mass transfer rates are presented in Fig. 8 to Fig. 13. Bulk temperature and concentration of the inlet absorbent remain at $T_i = 40^\circ\text{C}$ and $C_i = 40\%$, and the corresponding equilibrium saturated vapor pressure of the absorbent solution, P_v , equals 13 torr. Six values of operating pressure of the absorber-evaporator unit P_v are tested, 13, 15, 20, 30, 40, and 50 torr. The corresponding driving potentials, ΔP , are 0, 2, 7, 17, 27, and 37 torr. The inlet cooling water temperature

Fig. 10 Effect of inlet absorbent film Reynolds number on the overall absorption rate

and flow rate remained at $T_{ci} = 25^\circ\text{C}$ and $M_c = 40$ lpm during the tests.

The difference between the absorbent inlet bulk concentration, C_i , and outlet bulk concentration, C_o , of the absorbent falling film is assigned as ΔC . The effect of operating pressure, P_v , and inlet absorbent Reynolds number, Re_i , on the concentration difference, ΔC , is shown in Fig. 8. As shown in this figure, a larger driving potential, ΔP , due to increased operation pressure would result in an increase of ΔC . Thus, the driving potential of the absorbent falling

Fig. 11 Effect of operating pressure level on the overall mass transfer coefficient

Fig. 13 Effect of operating pressure level on the overall heat transfer coefficient

Fig. 12 Effect of operating pressure level on the overall heat transfer rate

film for absorbing vapor increases as ΔP increases.

As shown in Fig. 8, ΔC for operating pressure $P_v=40$ torr ($\Delta P=27$ torr) shows a slow linear increase as Re_i decreases from 800 to 300, and a gradually increasing trend as Re_i decreases from 300 to 200. The ring-shaped waves are always observed on the absorbent film. The surface renewal effect on the absorbent film is the main reason for high mass and heat transfer in the vertical falling film. However, the driving potential of the transport process rises due to

decrease of film thickness as Re_i is below 200, and in consequence, ΔC increases sharply. The film rupture phenomenon appears at the bottom of the absorbing tube near the collection funnel as Re_i reduces to 90. The corresponding mass flow rate of the absorbent solution is considered as the critical flow rate for film breakdown.

Similar trends have been developed for different operating pressures except for $P_v=13$ torr. At $P_v=13$ torr, driving potential of the inlet absorbent $\Delta P=0$ torr, and no vapor is likely to be absorbed. As the absorbent flows down, the bulk temperature of absorbent film is reduced by the cooling water flowing through the absorbing tube. The equilibrium saturated vapor pressure on the absorbent film interface will be lower than the operation pressure, and the driving potential for absorption is generated consequently.

Accordingly, in the falling film absorber-evaporator operation, the ideal operating flow rate of absorbent falls when there is sharp increase in ΔC . The concentration difference of absorbent solution film between inlet and outlet increases sharply as the Reynolds number of the inlet absorbent film decreases below 200. The irreversibility of the operating procedure for returning absorbent solution could be eliminated then by the increasing concentration difference during the vapor absorption process. When the Reynolds number of the inlet absorbent film is around 200, optimal conditions for the absorption-evaporation process exist and will provide appropriate conditions for a high efficiency absorption air conditioning system.

2. Overall Absorption Rate and Mass Transfer Coefficient

The calculated overall absorption mass transfer rate, M_T , from Eq. (13) can be deduced from the measured data M_i , C_i , and C_o . The effects of the operating pressure, P_v , and the inlet absorbent Reynolds number, Re_i , on M_T are presented in Fig. 9. As depicted in Figs. 8 and 9. When, the concentration difference, ΔC , decreases, the overall absorption rate M_T increases with increasing Re_i . When Re_i is over 700 the rate of increase of M_T becomes milder. For the same inlet absorbent concentration, a greater Re_i means a higher absorbent flow rate. Eq. (13) indicates that the outlet absorbent concentration, C_o , increases and $(C_i/C_o - 1)$ decreases with increasing Re_i . Hence the absorption rate, M_T , increases with the increase of inlet absorbent mass flow rate, M_i . At $P_v=15$ torr, because the driving potential is much smaller, the absorption rate, M_T , stays constant, disregarding the increase in Re_i . However, when $P_v=13$ torr, M_T tends to decrease as Re_i increases.

Figure 10 depicts the effects of P_v on the concentration difference at different inlet absorbent Reynolds numbers. It indicates that M_T increases as P_v increases, and the variation of P_v has a great effect on the absorption rate M_T at higher Re_i values. The effect of operating pressure, P_v , on the overall mass transfer coefficient, N , at different inlet absorbent Reynolds numbers, Re_i , is shown in Fig. 11. According to the data reduction process of N , the assumption that the bulk temperature of the absorbent is equal to the interfacial temperature of the absorbent ($T_b=T_f$) will result in an overestimation of ΔC_{lm} and ΔT_{lm} , and will in turn lead to unrealistically low N and high U . According to Eq. (9), the exponent $n=0.55$ obtained by Chen and Sun (1997) is adopted in the present study. As shown in Fig. 11, N increases with the increase of Re_i and that of the driving potential with the augmentation of operating pressure, P_v .

3. Overall Heat Transfer Rate and Coefficient

The variation of overall heat transfer rate, Q_T , calculated from Eq. (16) is plotted in Fig. 12 as a function of inlet absorbent Re_i for various operating pressures, P_v , of the absorption compartment. As shown in Fig. 12, the overall heat transfer rate decreases significantly as Re_i increases, and the decreasing trend is augmented as P_v increases. As depicted in Fig. 9, when Re_i octuples from 100 to 800, the absorption rate rise is less than double. Since the heat transporting to the absorbent film, per unit mass of vapor condensation, is greater as Re_i is smaller, the outlet absorbent temperature should decrease with increasing Re_i . This is confirmed by the measured

data of the present study. The bulk temperature difference between the absorbent film and the cooling water, and the overall heat transfer rate decreases as Re_i increases. The reason is that the pressure driving potential decreases with the decrease in P_o . This experimental result is different from that of Chen and Sun (1997). Since the inlet Reynolds number Re_i of the absorbent increases, the contact for absorbent film to transport the absorption heat from the film surface to the cooling coil decreases. The absorption heat might not be transported totally, and remains in the absorbent film. Therefore, the different result of Q_T from Chen and Sun (1997) is obtained. Nonetheless, they found that the overall heat transfer rate Q_T increased as Re_i increased.

Figure 13 shows the effect of Re_i on the overall heat transfer coefficient, U , under different P_v . The variation trend is opposite to that presented in Fig. 12. Although decrease in the overall heat transfer rate, Q_T , corresponds to decrease in the pressure driving potential ΔP , ΔT_{lm} increases significantly as ΔP decreases. Therefore, with some compensation for the ΔT_{lm} , the value of U decreases.

V. CONCLUSION

The heat and mass characteristics of the new vertical twin-parallel-tube absorber-evaporator unit operated under different pressures are investigated experimentally. Key parameters involved in the study include the Reynolds number of the inlet absorbent solution and the operation pressure for vacuum absorption. The method of analogy between heat and mass transfer adjacent to the film surface is used to iteratively calculate the interfacial temperature and concentration of the absorbent film, and the mass and heat transfer coefficients are derived accordingly. Important results obtained are summarized as follows:

- (1) Concentration difference ($C_i - C_o$) of the absorbent film between inlet and outlet of the absorber increases sharply as the inlet Reynolds number, Re_i , of absorbent decreases below 200, where ideal operation conditions for absorption-evaporation processes exist.
- (2) The concentration difference, ($C_i - C_o$), the overall absorption rate, M_T , and the overall heat transfer rate, Q_T , increase with increasing operating pressure. As operating pressure decreases, the logarithmic mean temperature difference, ΔT_{lm} , increases significantly and the overall heat transfer coefficient, U , of the absorber decreases consequently.
- (3) Total absorption rate of the absorber increases as the inlet Reynolds number of the absorbent solution increases, and tends to increase slightly against higher values of Re_i . The total heat

transfer rate of the absorber decreases with increasing Re_i .

NOMENCLATURE

C	absorbent solution concentration by weight	%
c_p	absorbent solution specific heat	kJ/kg $^{\circ}$ C
c_{pw}	water specific heat	kJ/kg $^{\circ}$ C
D	mass diffusivity	m 2 /sec
d	outer diameter of vertical tube	m
E_T	Ackermann correction factor	
g	gravitational acceleration constant	m/sec 2
h	enthalpy of absorbent solution	kg/kJ
k	thermal conductivity	kW/m $^{\circ}$ C
L	total wetting length	m
Le	Lewis number, $Le=Pr/Sc=\alpha/D$	
M	absorbent solution mass flow rate	kg/sec
m_a	local water vapor absorption rate	kg/sec-m 2
M_T	overall water vapor absorption rate	kg/sec
N	overall mass transfer coefficient	m/sec
n	constant	
Nu	Nusselt number	
P	equilibrium pressure of absorbent solution	torr
Pr	Prandtl number, $Pr=\nu/\alpha$	
P_v	operation pressure	torr
q_a	local heat flux	kW/m 2
Q_T	overall heat transfer rate	kW
Re	Reynolds number, $Re=4\Gamma/\mu$	
Sc	Schmidt number, $Sc=\nu/D$	
Sh	Sherwood number, $Sh=N(\nu^2/g)^{1/3}/D$	
T	absorbent solution film temperature	$^{\circ}$ C
U	overall heat transfer coefficient	kW/m 2 - $^{\circ}$ C
u	velocity distribution function	m/sec
z	longitudinal coordinate	m

Greek letters

Γ	mass flow rate per unit width	kg/sec-m
α	thermal diffusivity	m 2 /sec
β	local mass transfer coefficient	m/sec
δ	film thickness	m
ϕ	porosity	
η	local heat transfer coefficient	kW/m 2 - $^{\circ}$ C
μ	absorbent solution viscosity	kg/sec-m
ν	absorbent solution kinematic viscosity	m 2 /sec
θ	interface contact angle	
ρ	absorbent solution density	kg/m 3
σ	surface tension factor	N/m
ΔC_{lm}	logarithmic mean concentration difference	%
ΔP	driving potential pressure difference	torr
ΔT_{lm}	logarithmic mean temperature difference	$^{\circ}$ C

Subscripts

a	absorption
B	bulk condition
b	absorbent solution
c	cooling water
e	thermodynamic equilibrium condition
f	interfacial
h	heating water
i	inlet
o	outlet
v	water vapor
w	pure water

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垂直雙圓管吸收－蒸發器之熱質傳分析

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摘 要

本研究建立了一垂直雙圓管吸收-蒸發器，以更理想之條件研究於不同操作條件下，下落液膜之熱質傳效應。為達成吸收液膜均勻下落於吸收器表面，本系統中採用具傾斜面的環型吸收液分佈器，使圓管表面均勻潤濕。本研究應用了熱質傳間類比關係來分析吸收過程中吸收液膜表面溫度與濃度變化，進而得到熱傳係數與質傳係數。本研究之結果發現增加系統操作壓力時，能提昇吸收液膜出入口濃度差，且在吸收液入口雷諾數低於200時，吸收液出入口濃度差陡然上升，而此一區間便是吸收過程的最佳操作點。由本文的研究可應用於吸收式空調系統之最佳化操作條件。

關鍵詞：表面潤濕，熱質傳類比關係，下落液膜，操作壓力。