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An analysis of simultaneous heat and water vapor exchange through a flat paper plate crossflow total heat exchanger

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Abstract—An analysis of simultaneous exchange of heat and water vapor in a crossflow-type total heat exchanger which is made of a Japanese paper impregnated with some kind of hygroscopic agent is carried out by employing a permeability coefficient based on an analogy between heat and mass transfer. It is shown that the predictions are matched well to experiments for the temperature and humidity efficiency. In respect to heat and mass transfer processes through the total heat exchanger, a particularly interesting fact is also revealed that the rate of heat transfer is dominated by air rather than the paper, while the rate of mass transfer of water vapor is dominated by the paper rather than air.

1. INTRODUCTION

'SHOJI' (paper sliding screen) paper, a material which exists in Japan from ancient times, is characterized by excellent heat insulation and air permeability properties. It is presumed that the heat insulation properties of the Japanese paper are attributable not to its low thermal conductivity, but rather to its obstruction of the air convection perpendicular to a shoji screen due to the partition function. The air permeability arises from the porosity of the Japanese paper.

A flat paper plate crossflow-type total heat exchanger was invented by Yoshino in 1969 [1] and is already applied in products currently available on the market. It is used in combination with a fan, as an air-conditioning ventilator, which has the same functions as the shoji that is retaining the warmth of the room while ventilating. This type of air-conditioning ventilator is currently attracting attention as an energy-saving device, owing to the conciliation of the two contradictory functions mentioned above. The key point for materialization of the merits of this equipment is to use paper in the exchanger element. In spite of the low thermal conductivity of the paper, the temperature efficiency of the heat exchanger is little reduced on account of its thin thickness, while the problem of leaking of air and other gases due to the gas permeability of paper is solved by impregnating hydrophillic resin and hygroscopic agents in the paper to fill up its pores. In consequence of taking this measure the gas permeability of the paper is reduced remarkably, while the water vapor permeability remains almost unchanged.

Total heat exchange means simultaneous exchange of both heat and water vapor, that is enthalpy exchange. The temperature efficiency of the total heat exchanger can be calculated with ease by usual methods, but to the author's knowledge, there is no precedent for the analysis of humidity exchange in heat exchangers. Impermeable packing paper and coated paper heretofore in use have practically no pinholes and are hard to permeate and therefore the mass transfer resistance at the air side is negligibly small compared to that in the paper or film. Such being the case, the overall moisture permeability coefficient is dominated by the moisture permeability coefficient in the paper. However, in the case of materials which allow the easy permeation of water vapor like Japanese paper submitted to special treatment, it is necessary to analyze the moisture exchange by considering each component of the moisture permeability coefficient corresponding to the paper and to the air layers of both sides, respectively. This paper describes the analysis of the foregoing type moisture exchange of the total heat exchanger and the outline of its characteristics.

2. STRUCTURE AND CHARACTERISTICS OF THE TOTAL HEAT EXCHANGER

2.1. *Structure*

As is shown in Fig. 1, the total heat exchanger dealt with in this paper is a flat plate crossflow-type exchanger capable of simultaneous exchange of heat and water vapor, where the partition plates are composed of specially treated paper impregnated with hydrophillic resin and hygroscopic agents, and the spacing wavy plates are composed of special kraft paper. Therefore, it has the advantages of easy construction, light weight and cheap cost compared with metallic heat exchangers. The partition plate has the property of selective permeability, which allows a good permeation of water vapor and a negligible permeation of air, carbon dioxide, carbon monoxide, hydrogen sulfide gas and so on. Therefore, the air-conditioning ventilator employing this type of heat exchanger is able to carry out the simultaneous exchange of temperature and humidity between the heated or cooled contaminated air which is exhausted to

NOMENCLATURE

- A_f spacing plate area [m^2]
- A_p partition plate area [m^2]
- A_t total water vapor transfer area [m^2]
- c fluid capacity rate [W K^{-1}]
- C_m fluid mass capacity rate [kg s^{-1} Torr^{-1}]
- C_{pa} specific heat of air [J kg^{-1} K^{-1}]
- C_{pv} specific heat of water vapor [J kg^{-1} K^{-1}]
- D diffusion coefficient [m^2 s^{-1}]
- d_h hydraulic equivalent diameter [m]
- G mass velocity of water vapor [kg m^{-2} s^{-1}]
- h heat transfer coefficient [W m^{-2} K^{-1}]
- h_m mass transfer coefficient [kg m^{-2} s^{-1} Torr^{-1}]
- i enthalpy [J kg^{-1}]
- K_{pa} permeability coefficient of water vapor in air under atmospheric pressure [kg m^{-1} s^{-1} Torr^{-1}]
- K_{pp} permeability coefficient of water vapor in paper [kg m^{-1} s^{-1} Torr^{-1}]
- k_p thermal conductivity of paper [W m^{-1} K^{-1}]
- L latent heat of water vapor [J kg^{-1}]
- l pipe length; also side length of partition plate [m]
- M weight of one mole of substance [kg]
- n number of partition plates [dimensionless]
- p partial pressure [Torr]
- Δp_{ln} logarithmic mean overall partial pressure difference [Torr]
- Q volumetric flow rate [m^3 s^{-1}]
- q_i recovered enthalpy transfer rate [W]
- q_L recovered latent heat transfer rate [W]
- q_s recovered sensible heat transfer rate [W]
- R universal gas constant [Torr m^3 kmol^{-1} K^{-1}]
- r coordinate in radial direction [m]
- Re Reynolds number [dimensionless]
- S cross-sectional area of channel [m^2]
- T temperature [K]
- U average overall heat transfer coefficient [W m^{-2} K^{-1}]
- U_m average overall mass transfer coefficient [kg m^{-2} s^{-1} Torr^{-1}]
- v_a air velocity [m s^{-1}]
- W mass flow rate of water vapor; also that of air [kg s^{-1}]
- X specific humidity [kg kg^{-1}]
- x, z coordinates [m].

Greek symbols

- δ partition plate thickness [m]
- ε_i enthalpy efficiency [dimensionless]
- ε_L humidity efficiency [dimensionless]
- ε_s temperature efficiency [dimensionless]
- ρ density [kg m^{-3}]
- ϕ fin efficiency [dimensionless].

Subscripts

- 1, 2 two fluid sides
- calc. calculation
- e exit condition
- FA feed air to room
- i inlet condition
- meas. measurement
- OA fresh air from outside
- RA exhaust air from room
- w water vapor.

the outdoors from the room and the fresh air which is supplied into the room, when they pass through the exchanger. The exchange of heat and moisture is carried out through the partition plate, where the spacer plate made of kraft paper plays the role of a fin particularly for the transfer of water vapor.

2.2. *Heat recovery and its efficiency*

The efficiency of the foregoing total heat exchanger differs somewhat depending on the geometries of equipment and the operating conditions, but the values with standard air flow rate are as follows: temperature efficiency $\varepsilon_s = 60$–80%; humidity efficiency $\varepsilon_L = 50$–70%; and enthalpy efficiency $\varepsilon_i = 55$–75%. The operating temperature ranges normally from -10 to 50°C.

When using the total heat exchanger as an air-conditioning fan, air supplied to an air-cooled room during the summer is pre-cooled and dehumidified, while air supplied to a heated room during the winter is pre-heated and humidified. Therefore, air is supplied at a state close to that prevailing in the room in either case. The amount of heat recovered by the total heat exchanger, that is, the reduction of the fresh air load is evaluated as follows.

Recovered sensible heat transfer rate q_s(W)

$$q_s = C_{pa}W(T_{FA}-T_{OA}) = C_{pa}W(T_{RA}-T_{OA})\varepsilon_s. \quad (1)$$

Recovered latent heat transfer rate $q_L(W)$

$$\begin{aligned} q_L &= W[(L+C_{pv}T_{FA})X_{FA}-(L+C_{pv}T_{OA})X_{OA}] \\ &= W[(L+C_{pv}T_{RA})X_{RA}-(L+C_{pv}T_{OA})X_{OA}]\varepsilon_L. \quad (2)\end{aligned}$$

Recovered total heat (enthalpy) transfer rate $q_i(W)$

$$\begin{aligned} q_i &= q_s+q_L \\ &= W(i_{FA}-i_{OA}) = W(i_{RA}-i_{OA})\varepsilon_i \quad (3)\end{aligned}$$

where C_{pa} and C_{pv} are the isobaric specific heats of dry air and water vapor, respectively, W is the mass flow rate of processed air, T_{FA}, T_{OA} and T_{RA} are the

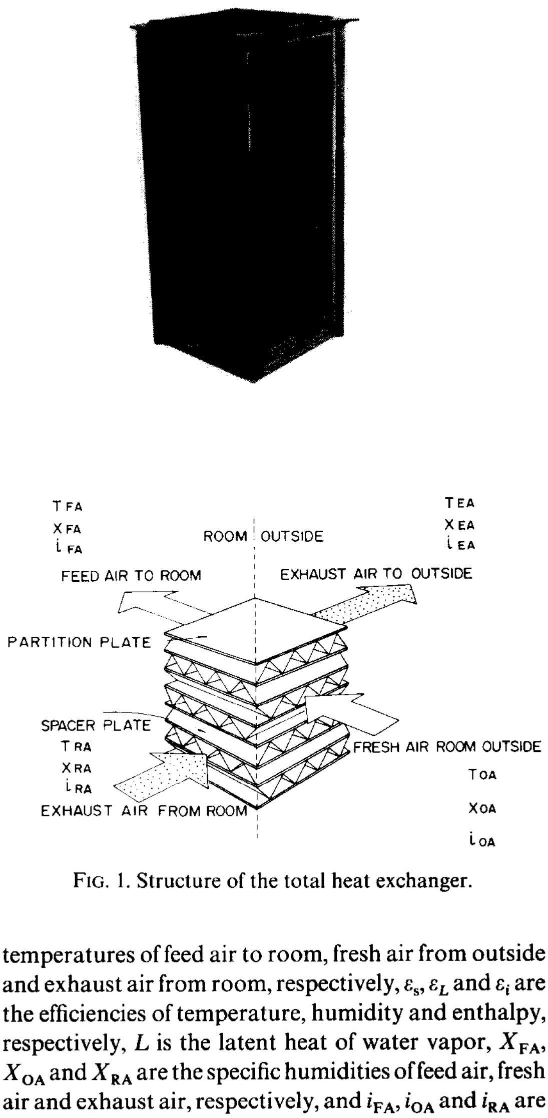

FIG. 1. Structure of the total heat exchanger.

temperatures of feed air to room, fresh air from outside and exhaust air from room, respectively, ε_s, ε_L and ε_i are the efficiencies of temperature, humidity and enthalpy, respectively, L is the latent heat of water vapor, X_{FA}, X_{OA} and X_{RA} are the specific humidities of feed air, fresh air and exhaust air, respectively, and i_{FA}, i_{OA} and i_{RA} are the enthalpies of feed air, fresh air and exhaust air, respectively.

It must be noted that only the sensible heat is recovered in the usual type heat exchanger, whereas in the total heat exchanger a considerable improvement of efficiency is obtained by adding the latent heat to the recovered sensible portion. The ratio of the recovered total heat transfer rate q_i to the recovered sensible heat transfer rate q_s is given as $(i_{FA}-i_{OA})/C_{pa}(T_{FA}-T_{OA})$ from equations (1) and (3), and during the summer, for example, q_i gets approximately four times as large as q_s, because $C_{pa} \cong 1$ kJ kg^{-1} K^{-1} and $(i_{FA}-i_{OA})/(T_{FA}-T_{OA}) \cong 4$ kJ kg^{-1} K^{-1}.

3. ANALYSIS OF HUMIDITY EXCHANGE

3.1. *Permeability coefficient*

If it is supposed that the mass transfer of water vapor can be handled independently of the heat transfer by air in the analysis of the mechanism of transfer of water vapor by permeation because of a small amount of water vapor in the air, then a clear correspondence is established between the mass transfer coefficient and the heat transfer coefficient, in view of the analogy between the two cases.

According to Fick's law, the diffusable mass velocity G of water vapor in air is expressed as follows supposing that the water vapor is an ideal gas

$$G = -\frac{DM_w}{RT}\frac{dp_w}{dx} = -K_{pa}\frac{dp_w}{dx} \tag{4}$$

where

$$K_{pa} = \frac{DM_w}{RT}. \tag{5}$$

In the above equations, D is the diffusion coefficient of water vapor in air under atmospheric pressure, M_w is the weight of one mole of water vapor, R is the universal gas constant, and p_w is the partial pressure of the water vapor. In this paper K_{pa} is termed the permeability coefficient of the water vapor in air. From equation (5) the permeability coefficient of water vapor in air under atmospheric pressure is evaluated as shown in Table 1. The diffusion coefficients listed in Table 1 are quoted from the *International Critical Tables* [2].

It is also possible to make theoretical predictions of the permeability coefficient of water vapor in paper, K_{pp}, that is defined in a similar manner to equation (4) in the low humidity range [3, 4]. In the present paper, however, the moisture permeability of paper was estimated experimentally by means of changing the partition plate thickness of the total heat exchanger and then observing the difference in humidity efficiency holding all the other parameters fixed.

K_{pp} generally depends on the relative humidity; it is almost constant below about 40% relative humidity and increases with increasing relative humidity [3]. The experimental results show that for the special papers impregnated with chemicals, K_{pp} is in the range 2.8×10^{-9}–9.2×10^{-9} kg m^{-1} s^{-1} Torr^{-1} under

Table 1. Diffusion coefficient D, and permeability coefficient K_{pa} of water vapor in air at atmospheric pressure

Temperature T (K)	Diffusion coefficient* $D \times 10^5$ (m^2 s^{-1})	Permeability coefficient $K_{pa} \times 10^8$ (kg m^{-1} s^{-1} Torr^{-1})
273	2.20	2.32
283	2.34	2.39
293	2.49	2.45
303	2.64	2.51
313	2.80	2.58
323	2.95	2.64
333	3.12	2.70
343	3.28	2.76
353	3.49	2.82
363	3.62	2.88
373	3.80	2.94

* $D = D_0(T/T_0)^{1.75}$, where D_0 is the value of D at $T = T_0$ (=273 K).

ambient temperature and ordinary relative humidity range.

3.2. *Prediction*

3.2.1. *Method of analysis.* The flow of either fresh or exhaust air through the present total heat exchanger is assumed to be laminar, because the Reynolds number based on the hydraulic equivalent diameter ranges from 100 to 500 under standard volumetric flow. The prediction for the mass transfer of water vapor is carried out by using the general solution of the water vapor diffusion equation through a tube. In other words, by employing the partial pressure distribution of water vapor given by the diffusion equation (4), the mass flow rate of water vapor W at the channel wall is expressed as

$$W = -\frac{4S}{d_h}\int_{z=0}^{z=l} K_{pa}\left(\frac{\partial p_w}{\partial r}\right)_{r=d_h/2} dz \tag{6}$$

where S is the cross-sectional area of the channel, d_h is the hydraulic equivalent diameter, K_{pa} is the permeability coefficient of water vapor in air, z is the coordinate axis in the flow direction, and l is the pipe length.

The following equation is obtained, by assuming that the mass transfer coefficient at the channel wall surface is h_m corresponding to the heat transfer coefficient h and that the logarithmic mean overall partial pressure difference of water vapor between the wall surface and the fluid is Δp_{ln}

$$W = h_m \frac{4S}{d_h} l\Delta p_{ln}. \tag{7}$$

The following expression is obtained from equations (6) and (7)

$$h_m = \frac{-1}{l\Delta p_{ln}}\int_{z=0}^{z=l} K_{pa}\left(\frac{\partial p_w}{\partial r}\right)_{r=d_h/2} dz. \tag{8}$$

The mass transfer coefficient h_m [kg m^{-2} s^{-1} Torr^{-1}] defined here is different in units from h_D [m s^{-1}] which is conventionally used in the field of mass transfer [5].

The average overall mass transfer coefficient U_m is calculated by the following expression, in correspondence to the average overall heat transfer coefficient U, obtained from the mass transfer coefficient h_m and permeability coefficient K_{pp} of the channel wall

$$U_m = \frac{1}{A_p\left[\frac{1}{h_{m1}(A_p+\phi A_f)} + \frac{\delta}{K_{pp}A_p} + \frac{1}{h_{m2}(A_p+\phi A_f)}\right]} \tag{9}$$

where A_p is the partition plate area, A_f is the spacer plate (fin) area, ϕ is the fin efficiency (see Section 2.1), δ is the partition plate thickness, and subscripts 1 and 2 denote the two fluids.

The humidity efficiency can be evaluated in a similar manner to that of the number of transfer units (NTU) approach employed in the analysis of the heat exchanger [6]. The capacity rate C_m [kg s^{-1} Torr^{-1}] in mass transfer corresponding to the capacity rate C [J s^{-1} K^{-1}] in heat transfer is given by the following equation (cf. Appendix)

$$C_m = \frac{M_w Q}{RT} \tag{10}$$

where Q is the volumetric flow rate of the moist air [m^3 s^{-1}]. Thus the NTU for the mass transfer can be obtained as

$$NTU = \frac{U_m A_p}{C_m}. \tag{11}$$

3.2.2. *Calculation of humidity efficiency.* The humidity efficiency ε_L is defined as follows in a similar way to the temperature efficiency

$$\varepsilon_L = \frac{X_{1i}-X_{1e}}{X_{1i}-X_{2i}} \tag{12}$$

where the two fluids are distinguished by subscripts 1 and 2, inlet conditions are denoted by a subscript i and exit conditions by e, and X is the specific humidity of air. As an example from among the various geometries of the total heat exchanger, the author selects an external size of 130 mm in width, 130 mm in length, 300 mm in height as shown in Fig. 1. The cross-section through which the air flows is triangular in shape. The hydraulic diameter is the same as that employed in the treatment of heat transfer [5]. Based on the effective size shown in Fig. 2, the hydraulic diameter of a triangular channel is about 1.8 mm. The exchanger total water vapor transfer area on one side is expressed as

$$A_t = A_p + \phi A_f \tag{13}$$

where A_p is the partition plate area ($A_p = nl^2 = 1.35$ m^2) for the number of partition plates $n = 135$, and the effective side length of the partition plate $l = 100$ mm, A_f is the exchanger total fin area ($A_f = 1.89$ m^2), and ϕ is the fin efficiency. Assuming a straight fin with constant cross-section and the moisture permeability coefficient of paper $K_{pp} = 3.8\times10^{-9}$ [kg m^{-1} s^{-1} Torr^{-1}], the fin efficiency ranges from 4.4 to 5.5% over air velocity range $v_a = 0.7$–4.2 m s^{-1}.

The mass transfer coefficient h_m essentially originates in equation (8), but the author obtains it from Hausen's formula representing the average Nusselt number for a uniform temperature and fully developed laminar flow [7] in analogy between heat and mass transfer, where the Reynolds number is defined by the flow of air because the air contains only a small quantity of water vapor, while the Nusselt number and Prandtl number

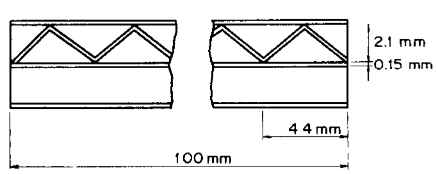

FIG. 2. Part of front view of the total heat exchanger element.

Table 2. Comparison between predictions and experimental data for humidity transfer

Volumetric flow rate $Q \times 10^2$ ($m^3 s^{-1}$)	Reynolds number Re	Mass transfer coefficient* $h_{m1} \times 10^5$ ($kg\ m^{-2} s^{-1} Torr^{-1}$)	Conductance of air side* $h_{m1}(A_p + \phi A_f) \times 10^4$ ($kg\ s^{-1} Torr^{-1}$)	Overall conductance $U_m A_p \times 10^5$† ($kg\ s^{-1} Torr^{-1}$)	Mass capacity rate $C_m \times 10^5$‡ ($kg\ s^{-1} Torr^{-1}$)	Number of transfer unit NTU ($U_m A_p/C_m$)	Humidity efficiency‡ $\varepsilon_{L,calc.}$ (%)	
							$\varepsilon_{L,calc.}$ (%)	$\varepsilon_{L,meas.}$ (%)
0.83	75	6.32	0.93	1.97	0.78	2.52	64	62
1.67	150	7.31	1.06	2.08	1.56	1.33	54	51
3.33	300	8.88	1.28	2.23	3.12	0.71	40	38
5.00	450	10.2	1.46	2.33	4.69	0.50	33	30

* $h_{m1} \simeq h_{m2}$.
† Conductance of partition plate, $K_{pp}A_p/\delta = 3.8 \times 10^{-9} \times 1.35/0.15 \times 10^{-3} = 3.43 \times 10^{-5}$ [$kg\ s^{-1} Torr^{-1}$].
‡ C_m and ε_L are values of the hot side fluid (inlet air temperature: 35°C) of the total heat exchanger. The humidity efficiency of the cold side fluid (inlet air temperature: 25°C) is slightly lower than that of the hot side fluid because of small difference of both the mass capacity rates.

are replaced by the dimensionless group $h_m d_h/K_{pa}$ and Schmidt number, respectively.

Then considering that the present total heat exchanger is a type of crossflow with unmixed fluids and using the heat transfer effectiveness as a function of the number of transfer units and the capacity rate ratio [5], the humidity efficiency is obtained. These are summarized in Table 2. The predicted humidity efficiency curve is also shown with the solid line in Fig. 3. The dotted line is the result of the calculation of the temperature efficiency using Hausen's formula [7].

3.3. *Comparison of predictions with experimental data*

A total heat exchanger 130 × 130 × 300 mm in size was installed in a wall partitioning two compartments with different humidity and temperature and the exchange of temperature and humidity was carried out between the two compartments. The temperature and humidity were measured by two dry bulb thermometers and two wet bulb thermometers installed at positions near the inlet and outlet of the heat exchanger, while the air flow rate was measured with an orifice flowmeter. The temperature and humidity efficiencies measured under the conditions shown in Table 2 are represented by solid and open circles, respectively, in Fig. 3. The measured and predicted values of either the temperature or the humidity efficiency coincide with each other within an error of several percent; the measured values of humidity efficiency are also listed in Table 2.

3.4. *Discussion*

In the case of the total heat exchanger operating at the standard flow rate (air velocity in the channel of approximately 2.8 m s^{-1}), the analysis made in this study shows that the thermal resistance at the air side is several times larger than that in the paper, while the mass transfer resistance at the air side is considerably less than that in the paper (as noted in Table 2), suggesting that the mass transfer resistance of the paper

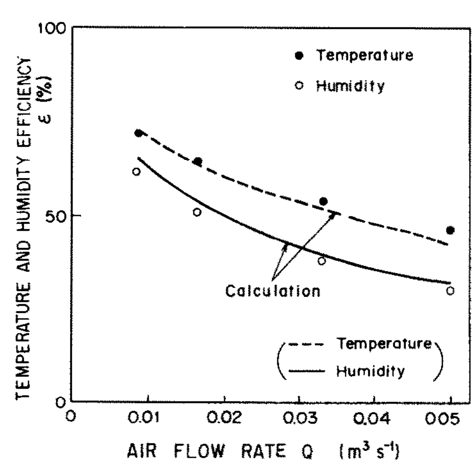

FIG. 3. Comparison between predictions and experimental data for the temperature and humidity efficiency of the total heat exchanger.

is larger than expected. The above-mentioned fact indicates that the influence of the thermal resistance of the paper on heat transfer is small (the calculations indicate that the increase in temperature efficiency is only about 1% when the partitioning paper is replaced with aluminum).

On the other hand, the calculations reveal that the mass transfer resistance of the paper is large, and therefore the moisture permeability coefficient of the paper has a pronounced influence on the humidity efficiency. Owing to the efforts to conciliate the contradictory demand of minimizing the permeability coefficient of air, carbon dioxide and other noxious gases through paper, and that of maximizing the moisture permeability coefficient of paper, the permeability coefficient of carbon dioxide in the specially treated paper is currently 2.8×10^{-11} kg m^{-1} s^{-1} Torr^{-1}, which is of the order of 1/100–1/30 of the moisture permeability coefficient. The permeability coefficient of the water vapor in air is 2.45×10^{-8} kg m^{-1} s^{-1} Torr^{-1} at 20°C as is noted in Table 1 as against the permeability coefficient of carbon dioxide in air estimated by equation (5): 3.83×10^{-8} kg m^{-1} s^{-1} Torr^{-1} at 20°C.

Incidentally, as for the carbon dioxide leak rate, the results of calculations similar to those of the moisture efficiency indicate that the carbon dioxide exchange efficiency ranges from 5 to 7% agreeing with experimental values within errors of 3–4%.

4. CONCLUSION

Based on an assumption that the temperature and humidity exchange are made independently in the flat plate crossflow-type total heat exchanger made of paper, due to a small quantity of water vapor in the air, an analysis of the humidity exchange has been carried out utilizing the analogy between heat and mass transfer. The humidity efficiency thus estimated agrees with the experimental data within errors of several percent.

The foregoing analysis also shows that the thermal conductivity of paper is approximately twice as large as that of air, whereas the average permeability coefficient of water vapor through paper is approximately a quarter of that in air. In other words, it can be said that the rate of heat transfer in the present heat exchanger is dominated by air rather than by the paper, while the rate of mass transfer of water vapor is dominated by the paper rather than by air.

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REFERENCES

1. M. Yoshino, Japanese patent Publication No. 47-19990, Heat exchanger (1972).
2. E. W. Washburn, *International Critical Tables*, Vol. 5, p. 62. National Research Council of U.S.A., McGraw-Hill, New York (1926).
3. K. Takahashi, K. Nakajima, H. Kusakawa and O. Tanaka, Vapor permeability of porous fiber materials, *Kagaku Kogaku Ronbunshu* **3**, 510–513 (1977).
4. K. Takahashi, H. Kusakawa and O. Tanaka, Vapor permeability of porous fiber materials—an investigation by a bundle model of tortuous capillaries, *Kagaku Kogaku Ronbunshu* **5**, 391–396 (1979).
5. E. R. G. Eckert and R. M. Drake, *Heat and Mass Transfer* (2nd edn.), pp. 159, 460–478. McGraw-Hill, New York (1959).
6. W. M. Kays and A. L. London, *Compact Heat Exchangers* (2nd edn.), pp. 13–63. McGraw-Hill, New York (1964).
7. H. Hausen, *Z. Ver. Dt. Ing., Beih. Verfahrenstechnik* 91–98 (1943).

APPENDIX

The fluid capacity rate C_m in mass transfer is defined as follows corresponding to the fluid capacity rate in heat transfer [6]

$$C_m = \frac{\mathrm{d}W}{\mathrm{d}p_w} = \frac{Q\,\mathrm{d}\rho_w}{\mathrm{d}p_w} \tag{A1}$$

where W is the mass flow rate of water vapor, p_w is the partial pressure of water vapor, Q is the volumetric flow rate of air, and ρ_w is the density of water vapor in air. For a mixture of air and water vapor, ρ_w can be expressed by the following equation based on the ideal gas assumption

$$\rho_w = \frac{M_w p_w}{RT}. \tag{A2}$$

Partially differentiating equation (A2) with respect to p_w under constant temperature yields

$$\left(\frac{\partial \rho_w}{\partial p_w}\right)_T = \frac{M_w}{RT} \tag{A3}$$

and hence the substitution of equation (A3) into equation (A1) gives

$$C_m = \frac{M_w W}{RT}. \tag{A4}$$

ANALYSE DE L'ECHANGE SIMULTANE DE CHALEUR ET DE VAPEUR D'EAU A TRAVERS UNE FEUILLE DE PAPIER D'ECHANGEUR DE CHALEUR TOTALE A COURANTS CROISES

Résumé—On donne une analyse de l'échange simultané de chaleur et de vapeur d'eau dans un échangeur de chaleur totale, type courants croisés, qui est fait de papier japonais imprégné avec différentes sortes d'agents hygroscopiques ; un coefficient de perméabilité est basé sur une analogie entre les transferts de chaleur et de masse. On montre que les prédictions sont bien vérifiées par les expériences pour la température et l'efficacité de l'humidité. De part les mécanismes de transfert de chaleur et de masse à travers l'échangeur de chaleur totale, un fait particulièrement intéressant est aussi révélé selon lequel le transfert de chaleur est dominé par l'air plutôt que par le papier, tandis aue le transfert de vapeur d'eau est dominé par le papier plutôt que par l'air.

UNTERSUCHUNG DES GLEICHZEITIGEN AUSTAUSCHES VON WÄRME UND WASSERDAMPF IN EINEM KREUZSTROM-WÄRMEÜBERTRAGER, DER AUS GLATTEN PAPIERFLÄCHEN AUFGEBAUT IST

Zusammenfassung—Der gleichzeitige Austausch von Wärme- und Wasserdampf in einem Kreuzstrom-Wärmeübertrager wird untersucht. Dieser besteht aus Japan-Papier, welches mit einem hygroskopischen Wirkstoff imprägniert worden ist. Bei der Analyse wird ein Permeabilitäts-Koeffizient verwendet, der auf der Analogie zwischen Wärme- und Stofftransport beruht. Der berechnete Wirkungsgrad, der mit den Temperaturen bzw. Feuchtigkeitsgehalten gebildet wird, stimmt gut mit Versuchsergebnissen überein. Für den Wärme- und Stofftransport im gesamten Wärmetauscher zeigt sich als besonders interessante Tatsache, daß das Wärmetransportvermögen stärker von der Luft als vom Papier bestimmt wird, während das Stofftransportvermögen bezüglich des Wasserdampfes eher vom Papier als vom Luftstrom abhängt.

АНАЛИЗ СОВМЕСТНЫХ ПРОЦЕССОВ ПЕРЕНОСА ТЕПЛА И ВОДЯНОГО ПАРА ЧЕРЕЗ ПЛОСКИЕ БУМАЖНЫЕ ПЛАСТИНЫ В ТЕПЛООБМЕННИКЕ С ПЕРЕКРЕСТНЫМ ТОКОМ

Аннотация—С помощью коэффициента проницаемости, рассчитываемого по аналогии между тепло-и массопереносом, проведен анализ совместных процессов переноса тепла и водяного пара в перекрестном теплообменнике, изготовленном из японской бумаги, пропитанной специальной гигроскопической жидкостью. Показано, что результаты расчетов температуры и влажности хорошо согласуются с экспериментальными данными. Что касается процессов тепло-и массопереноса во всем теплообменнике, то установлен очень интересный факт, что интенсивность теплопереноса определяется скорее передачей тепла через воздух, а не через бумагу, а интенсивность массопереноса—через бумагу, а не через воздух.