HEAT AND MOISTURE EXCHANGE IN COUNTERFLOW ROTARY AIR DEHUMIDIFIER

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ABSTRACT

This present work has theoretically investigated coupled heat and mass exchange mechanisms in counterflow rotary dehumidifier with the mixed desiccant of **LiCl** and additives, used in air conditioning systems. Mathematical model has been derived, based on one-dimensional transfer model and Polyni potential theory (theory of water chemical potential) and solved, using digital computer. The profiles of temperature and absolute humidity distribution in the matrix have been obtained. Analysis shows, that the direction of mass transfer is opposite to positive water vapour partial pressure. The results are presented in the form of the "overall effectiveness" of the heat exchanger. Dimensionless operating factors, affecting the efficiency of the heat exchanger have been investigated. The received results offer scope for estimation of variation range of rotary dehumidifier optimum operating conditions and suitable climatic zones for this unit.

INTRODUCTION

Drying of air in hot and humid regions may be a promising low cost alternative to traditional air conditioning. Rotary air dehumidifiers (Fig. 1) are often used in open-cycle air conditioning systems. However the substantiation of potential efficiency of heat and mass transfer processes and optimum operating conditions of these apparatus requires the detailed theoretical analysis.

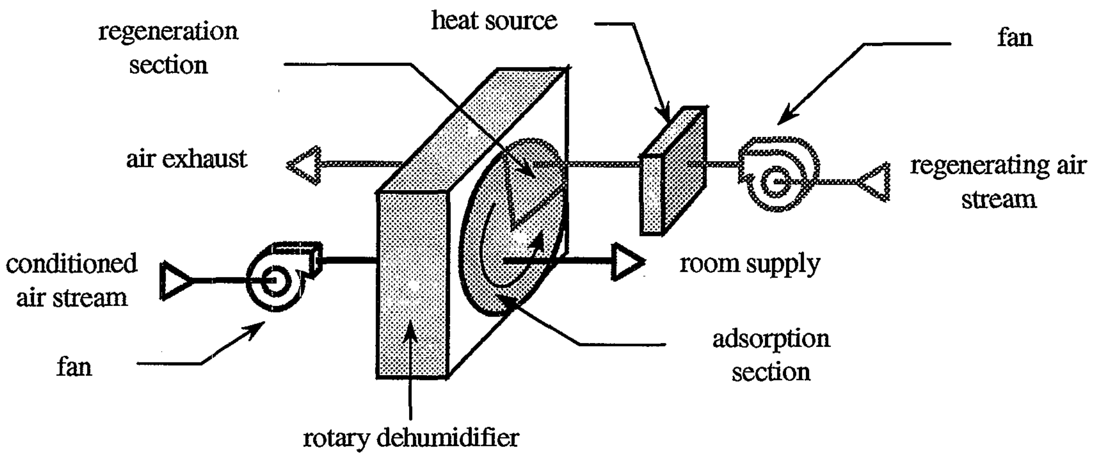

Fig. 1. Counterflow rotary dehumidifier

It is necessary to note that performance prediction from theoretical model for a regenerative dehumidifier is much more difficult than for similar energy recovery regenerator [1, 2].

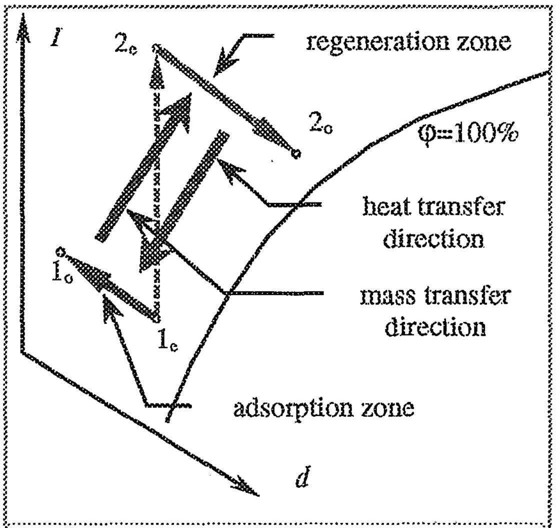

Fig. 2. *I-d*-chart, showing the air state paths of regenerating and conditioned air streams

The difficulty arises because the mass transfer from dried air stream to regeneration air flow is realised in direction opposite to positive difference of water vapour partial pressure (Fig. 2). Therefore mathematical model building was based on Polyni potential theory (theory of water chemical potential). The description of heat and mass transfer in terms of water chemical potential provides a better understanding of heat exchanger operation, thus facilitating design and development [3].

The present work is concerned with mathematical simulation of coupled heat and mass transfer in counterflow rotary air dehumidifier with matrix, composed wholly of industrial cardboard, impregnated with lithium chloride and special additives.

METHODS

In many problems, including considered one, an engineer is not interested in receiving thermodynamic variables distributions normal to the heat exchanger plate surface, but bulk average values. Therefore we gave preference to one-dimensional transfer model (α-model). In this case air stream in matrix passages is considered as a gaseous fluid flow with constant temperature and mass transfer potential in direction normal to plate surfaces, which are equalled to bulk average values.

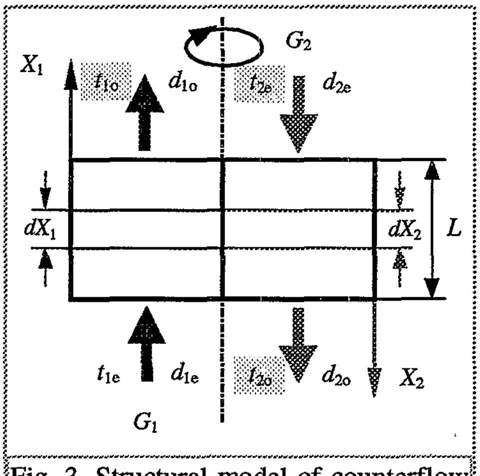

Fig. 3. Structural model of counterflow rotary dehumidifier

A regenerative exchanger transfers heat and water vapour between two air streams by means of a porous matrix through which the air streams are passed alternately. Equations of heat and mass transfer model in the counterflow rotary air dehumidifier are realised in orthogonal coordinates system, as shown in Fig.3.

Now let us formulate a number of assumptions, concerning to peculiarities of examined heat and mass transfer processes realisation and taken as a basis of developed model:

- driving force of mass transfer is a gradient of water chemical potential $\mu = RT \ln \varphi$;
- air flow is an ideal gas mixture of dry air and water vapours;
- the matrix is modelled as being of parallel passage form;
- matrix plates are assumed to be the relative thin colloidal capillary - porous medium;
- longitudinal molecular diffusion of water vapour in air and longitudinal heat transfer at the expense of thermal conductivity are negligible.

According to the assumptions of α-model coupled heat and water vapour transfer in investigated dehumidifier is described by governing differential equations of energy and mass balances made up for the both air streams and material of hygroscopic matrix.

$$\begin{cases} \dfrac{\partial t_j}{\partial \bar{\tau}} = \mathrm{Ho}_j \left[\mathrm{NTU}_j \left(t_{wj} - t_j \right) - \dfrac{\partial t_j}{\partial \bar{X}_j} \right] \\ \dfrac{\partial \mu_j}{\partial \bar{\tau}} = \mathrm{Ho}_j \left[\left(\mathrm{NTU}'_\mu \right)_j \left(\mu_{wj} - \mu_j \right) + \left(\partial\mu/\partial T \right)_{d_j} \mathrm{NTU}_j \left(t_{wj} - t_j \right) - \dfrac{\partial \mu_j}{\partial \bar{X}_j} \right] \\ \dfrac{\partial t_{wj}}{\partial \bar{\tau}} = \mathrm{NTU}_{wj} \left(t_j - t_{wj} \right) + q_{sorp} \left(\dfrac{c'_\mu}{c} \right)_{wj} \left(\mathrm{NTU}'_\mu \right)_{wj} \left(\mu_j - \mu_{wj} \right) \\ \dfrac{\partial \mu_{wj}}{\partial \bar{\tau}} = \left(\mathrm{NTU}'_\mu \right)_{wj} \left(\mu_j - \mu_{wj} \right) \end{cases} \qquad \text{...(1)}$$

where $\bar{\tau} = \tau/\tau_r$ – dimensionless time coordinate; $\bar{X} = X/L$ – dimensionless axial coordinate along plate in direction of corresponding air stream; j = counter of stage (j = 1 – subscript for conditioned air stream; j = 2 – subscript for regenerating air stream); $\bar{X}_2 = 1 - \bar{X}_1$
Dimensionless complexes, involved in set of governing equation (1), are defined as

$$\mathrm{NTU} = (\alpha \cdot F)/(G \cdot c_p); \qquad \mathrm{NTU}_w = (\alpha \cdot F \cdot \tau_r)/(M_w \cdot c_w); \ \mathrm{Ho} = (w \cdot \tau_r)/L;$$
$$\mathrm{NTU}_\mu = (\beta_\mu \cdot F_{por})/(G \cdot c'_\mu); \quad \left(\mathrm{NTU}'_\mu \right)_w = (\beta_\mu \cdot F_{por} \cdot \tau_r)/\left[M_w \cdot (c'_\mu)_w \right]. \qquad \text{...(2)}$$

To derive the unique decision the set of simultaneous partial differential equations (1) is supplemented by the initial conditions, establishing matrix temperature and water chemical potential distribution in direction of air stream

$$t_{w1} \Big|_{\substack{\bar{\tau}=0 \\ \bar{X}_1 = 0-1}} = t_{w1i}\left(\bar{X}_1 \right); \qquad \mu_{w1} \Big|_{\substack{\bar{\tau}=0 \\ \bar{X}_1 = 0-1}} = \mu_{w1i}\left(\bar{X}_1 \right), \qquad \text{...(3)}$$

and boundary conditions, prescribing values of inlet air thermodynamic parameters,

$$t_1 \Big|_{\substack{\bar{X}_1=0 \\ \bar{\tau}>0}} = t_{1e} \ ; \quad d_1 \Big|_{\substack{\bar{X}_1=0 \\ \bar{\tau}>0}} = d_{1e} \ ; \quad t_2 \Big|_{\substack{\bar{X}_2=0 \\ \bar{\tau}>0}} = t_{2e} \ ; \quad d_2 \Big|_{\substack{\bar{X}_2=0 \\ \bar{\tau}>0}} = d_{2e} = d_{1e}. \qquad \text{...(4)}$$

The boundary conditions used in modelling of multistage cyclical processes are supplemented by switching conditions

$$t_{w2}\left(\bar{X}_2 \right) \Big|_{\bar{\tau} = \Delta\bar{\tau}_1 + (k-1)} = t_{w1}\left(\bar{X}_1 \right) \ ; \qquad \mu_{w2}\left(\bar{X}_2 \right) \Big|_{\bar{\tau} = \Delta\bar{\tau}_1 + (k-1)} = \mu_{w1}\left(\bar{X}_1 \right) \ ;$$
$$t_{w1}\left(\bar{X}_1 \right) \Big|_{\bar{\tau} = k} = t_{w2}\left(\bar{X}_2 \right) \ ; \qquad \mu_{w1}\left(\bar{X}_1 \right) \Big|_{\bar{\tau} = k} = \mu_{w2}\left(\bar{X}_2 \right) \ , \qquad \text{...(5)}$$

where k = counter of cycles; $\Delta\bar{\tau}_1$ = relative duration of dehumidification stage.

Taking into account non-linear character of the pre-set relationships between thermodynamic constants c'_μ, $(c'_\mu)_w$, $(\partial\mu/\partial T)_d$, q_{sorp}, β_μ and heat and mass transfer potentials, the boundary value problem with dynamic boundary conditions was solved numerically.

Numerical investigation was executed with the help of modified Runge-Kutta method, distinguished by sufficient accuracy and stability at the decision of similar type problems [4]. The approximate decision was found in given space-time domain, divided by a grid with variable step size into rectangular elements. The computation was carried out by space layers along air flow direction starting with the first. The changes of thermodynamic variables of air stream and matrix at transition from one time node to another were defined by Runge-Kutta method, and the partial derivatives with respect to $\overline{X}$ were replaced with finite difference analog.

To take an approach to numerical modelling by a natural way, it is necessary to calculate adsorption stage, then regeneration stage and after that repeat this procedure for the next cycle. Such sequence of evaluation is repeated until the temperature and water chemical potential distributions obtained at the end of adjacent periods remain practically unchanged. The main elements of the algorithm of regenerative dehumidifier performance computation are shown in Fig. 4.

The described algorithm was realised in digital computer program including 14 modules. The computer program evaluation tests were performed with the help of calculations of different variants, describing problems with the familiar analytical solutions. Besides the accuracy of numerical method was checked by computing of identical variants on various space-time domain grids. Convergence test showed that the optimum number of grid nodes and, consequently, minimum time of computation within the given tolerance of accuracy (0,5% thermal and mass balance error) was reached at the step size of 0,01 in $\overline{X}$ direction and 0,0002 in $\overline{\tau}$ direction

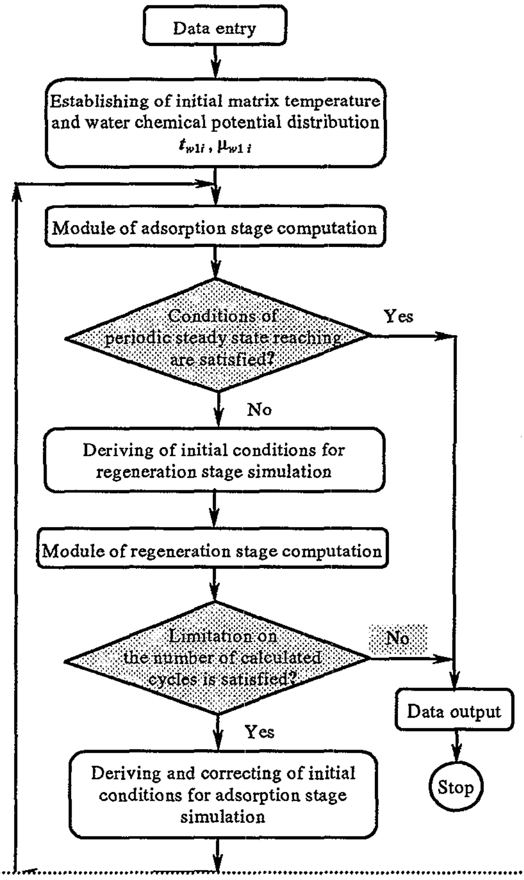

Fig. 4. Flowchart of dehumidifier mathematical model equations solution

RESULTS and DISCUSSION

On the basis of numerical investigations performed over a wide range of dimensionless complexes values (2) and under different boundary conditions (4) it was established that coupled heat and water vapour transfer in hygroscopic matrix was realised under "*combined wave mode*". The regularities of this mode are clearly exhibited under "slow" wheel rotation (Fig. 5). In this case matrix thermodynamic parameters at the end of each stage attain equilibrium conforming to the temperature and humidity of air stream on the entry to regenerator matrix.

Interaction between air stream and adjacent matrix on the frontal side of regeneration section causes humidification of regenerating air flow and cooling it up to t_{1e}, therefore values of air stream water chemical potential are made greater than initial one on the entry of matrix μ_{1e}. Converse process of adsorption arises during consequent air passing along matrix surface. Warming-up of matrix and increasing of its water chemical potential result in rising of absolute humidity and temperature of air stream and consequently more deep heating and humidifying occur in remote part of matrix . Revealed stabilisation of air stream water chemical potential causes equalising of air flow and matrix thermodynamic parameters

Thus, the investigated process is characterised by the presence of two active coupled heat and mass transfer zones – frontal and rear ones. Repeated adsorption of water vapour takes place in the first zone, therefore air and matrix temperature increases from initial value

t_{1e} to equilibrium one t_2^*. The second zone is formed by the wave front of the matrix complete warming-up. The raising of matrix element and its contained air temperature from equilibrium value t_2^* to t_{2e} causes water vapour desorption. It is necessary to note that velocity of repeated adsorption wave propagating through the matrix is much more greater then velocity of rear wave front ($v_f \approx 23 v_r$). This difference in waves velocities results in formation of the interstitial zone, in which air flow and matrix are in equilibrium condition at temperature t_2^* and water chemical potential μ_2^*.

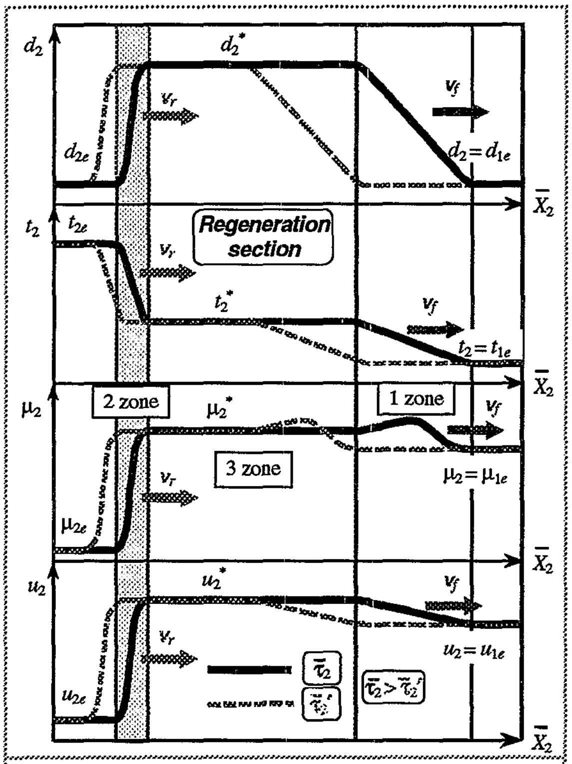

Fig. 5. Formation of coupled heat and mass transfer zones in regeneration section of dehumidifier

The similar pattern of "combined wave mode" was revealed in humidifying section. In this case repeated desorption of water vapour occurred in the frontal zone, while process of adsorption was carried out in the second (rear) zone generated by the wave front of the matrix complete cooling-up. It was established, that the second zone of active coupled heat and mass transfer propagated along the matrix in regeneration section with velocity approximately 1,5 times larger than those in processed section. This circumstance was taken into account in defining of optimum period of time spent by matrix correspondingly in conditioned and regenerating air stream.

The criteria generalisation of numerical experiment results allowed to reveal extreme influence of dimensionless operating complexes NTU, $\overline{W}$, $\overline{W}_2/\overline{W}_1$ on absolute humidity and temperature changes efficiency for conditioned air stream Δd_1, E_t. As shown in Fig. 6 the absolute humidity change efficiency achieves maximum value in a range of number of heat transfer units for air stream $15 < \text{NTU} < 30$. The noted values are by an order of magnitude greater than corresponding range of optimum NTU variation for energy recovery regenerators, therefore this is inhibitory to the extrapolate of presented in the literature energy recovery regenerators investigations results for regenerative dehumidifier performance optimization.

The analysis of "combined wave mode" regularities has allowed to substantiate an extreme character of function $\Delta d_1 = f(\text{NTU})$. At a small air flow capacity (NTU > 30), when thermodynamic state of air stream on exit from regenerator matrix are essentially determined by duration of the first "combined wave" front propagating along the matrix, the increasing of air flow capacity causes the rising of water vapour transfer efficiency at the expense of increasing of the first active heat and mass transfer zone propagation velocity v_f, that stipulates the shift of average outlet air stream absolute humidity in direction of low values (d_1^*). At a high air flow capacity (NTU < 15) the opposite trend is revealed (Fig. 6). In this case water vapour transfer efficiency depends upon the time of second wave front reaching matrix terminal parts, i.e. inversely proportional to the velocity of the second zone traversing v_r. Therefore increasing of air flow capacity causes decreasing of air dehumidification degree.

The received results offer scope for estimation of variation range of rotary dehumidifier optimum operating conditions ($15 < \text{NTU} < 30$; $2 < \overline{W} < 40$; $0{,}2 < \overline{W}_2/\overline{W}_1 < 1$) and suitable climatic zones for this unit.

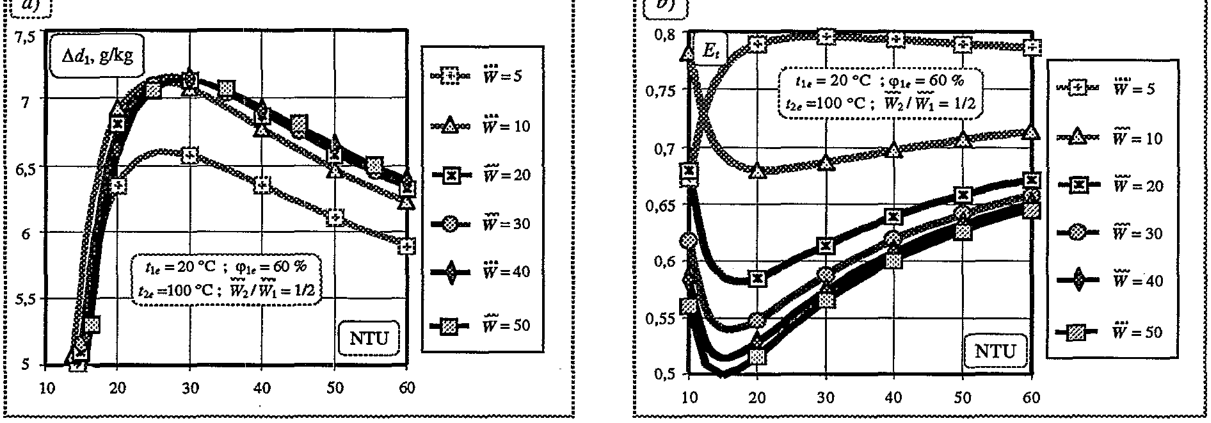

Fig. 6. Effects of dimensionless operating complexes NTU, $\overline{W}$, $\overline{W}_2/\overline{W}_1$ on absolute humidity and temperature changes efficiency for conditioned air stream Δd_1, E_t.

NOMENCLATURE

c = specific heat capacity, [J/(kg·K)]; d = absolute humidity of air, [kg/kg (g/kg)]; c'_μ - specific isothermal mass capacity (sorbability), related to water chemical potential difference, [mol/J]; E_t = temperature efficiency; F = transfer area of matrix, exposed to air stream, [m^2]; F_{por} = summarised transfer area of macro- and micro-porous per unit mass of dry matrix material, [m^2/kg]; G = air mass flow rate, [kg/s]; q_{sorp} = heat of adsorption per unit mass of sorbed water, [J/kg]; L = plate length in X direction, [m]; M_w = mass of dry matrix material, [kg]; R = universal gas constant [J/(mol·K)]; t = temperature, [°C]; T= absolute temperature, [K]; u = sorbed water content of matrix material, mass ratio to dry matrix material; $\overline{W}$ = air stream to matrix specific heat capacity rate ratio = $(G \cdot c_p \cdot \tau_r)/(M_w \cdot c_w)$; $\overline{W}_2/\overline{W}_1$ = regeneration air stream to conditioned air stream specific heat capacity rate ratio = $(G \cdot c_p)_2/(G \cdot c_p)_1$; X = distance in air flow direction, [m]; α = sensible heat transfer coefficient, [W/(m^2· K)]; β_μ = vapour transfer coefficient, related to water chemical potential difference, [(kg·mol)/(J·m^2·s)]; μ = water chemical potential, [J/mol]; $(\partial\mu/\partial T)_d$ = temperature coefficient of water chemical potential, [J/(mol·K)]; φ = relative humidity, percent; τ = time, [s]; τ_r = period time of cycle, [s].

SUBSCRIPTS

e = state of air stream on entry to regenerator matrix; o = average state of air stream on exit from regenerator matrix; p = at constant pressure; w = for regenerator matrix.

REFERENCES

1. Banks P.J. 1985. Prediction of heat and mass regenerator performance using non-linear analogy method, *Journal of heat transfer*, Vol. 1, pp. 222–238.
2. Jing Ding, Yingke Tan, Eric J. Hu. 1996. An open-cycle desiccant air conditioning system for improving indoor quality, *Proceedings of 7th International Conference on Indoor Air Quality and Climate - Indoor Air'96*, Vol. 1, pp. 319-324.
3. Bogoslovskij, V.N. 1996. Theory of Moisture Potential, *Proceedings of 7th International Conference on Indoor Air Quality and Climate - Indoor Air'96*, Vol. 3, pp. 891-896.
4. Iked, A. 1977. Heat and mass transfer in the non-isothermal fixed bed adsorption column with non-linear equilibrium, *Bulletin of the Faculty of Engineering, Yokohama National University*, Vol. 26, pp. 117-139.