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A HOLLOW FIBER MEMBRANE-BASED LIQUID DESICCANT AIR DEHUMIDIFICATION SYSTEM FOR SOLAR ENERGY RECLAMATION

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ABSTRACT

A mathematical model is proposed to study the heat and mass transfer performance of a hot water driven hollow fiber membrane-based liquid desiccant air dehumidification system. The system can be extended to solar energy cooling in summer, especially in hot and humid regions. The benefit with this technology is that the liquid desiccant is not in direct contact with the process air, therefore the problem of liquid droplets cross over is prevented. With the model, the effects of various operating parameters like fresh air temperature and humidity on system performance, in terms of SDP (specific dehumidification power) and COP (coefficient of performance), are discussed.

INTRODUCTION

Air conditioning in buildings has accounted for 1/3 of the overall energy use in the world. The heavy burden on economic growth and environmental protection has forced people to reduce energy use and to use renewable energy sources. Solar energy driven liquid desiccant air cooling system is just such a promising alternative to the traditional air conditioning systems. It uses low-grade hot water (which can be produced by solar energy) rather than high-grade electricity. It has the ability to storage energy by liquid desiccant, which solves the intermittent nature of solar energy. However traditional liquid desiccant system [1,2] has the serious problem of liquid droplets cross over, which is rather corrosive. In recent years, membranes have been used to solve this problem [3]. In this system, membranes are used to separate the liquid desiccant from air stream. Moisture in air can be absorbed by the liquid desiccant inside membrane fibers, but liquid droplets crossover is prevented. The system is promising.

To analyse the system performance, in this study, a mathematical model is proposed to simulate a hot water driven hollow fiber membrane-based liquid desiccant air dehumidification system. With the model, the effects of various operating parameters like fresh air temperature and humidity on system performance, in terms of SDP (specific dehumidification power), and COP (Coefficient of Performance), are discussed.

SYSTEM DESCRIPTION

Figure 1 is the system modelled. As seen, it is similar to a traditional liquid desiccant air dehumidification system, but two hollow fiber membrane modules, one for air dehumidification and the other for solution regeneration, are employed. The former one is called the dehumidifier and the latter one is called the regenerator. The structures for the two are the same. One air stream is fresh air that is dehumidified, and the other one is regenerating air that is in fact another fresh air. The cycle for the solution is 1-2-3-4-1. It comprises of four processes. (1) Process 1-2, dehumidification. In the solution container there is cool and concentrated liquid desiccant at state 1. LiBr solution is used as the desiccant solution. It is pumped to the tube side fibers in the dehumidifier. In the dehumidifier, the liquid desiccant absorbs water vapor from the air flowing in the shell side. The air is dehumidified. The solution becomes diluted with a temperature rise to state 2, after absorbing absorption heat. (2) Process 2-3, solution heating. The solution at state 2 is then heated by a hot water bath which is set to 75℃. The temperature rises and the solute mass fraction remains unchanged. This is so called pure heating process. (3) Process 3-4, regeneration. The hot solution at state 3 flows through the hollow fibers in the regenerator. Its water content is driven to the regeneration air flowing in the shell side. Solution becomes concentrated with a temperature decrease to state 4, after releasing absorption heat. The Regeneration air is exhausted to the ambient. (4) Process 4-1, solution cooling. The warm and concentrated solution at state 4 flows to a cooler by city water. It is further cooled and transported to the solution container at state 1. The solute mass fraction remains constant in this step. This is called pure cooling process. After this step, the cycle is completed.

Besides solution, the thermodynamic paths for the air streams are also depicted. The process for the dehumidified air is expressed as 5-6. The process for the regeneration air is expressed as 5-7.

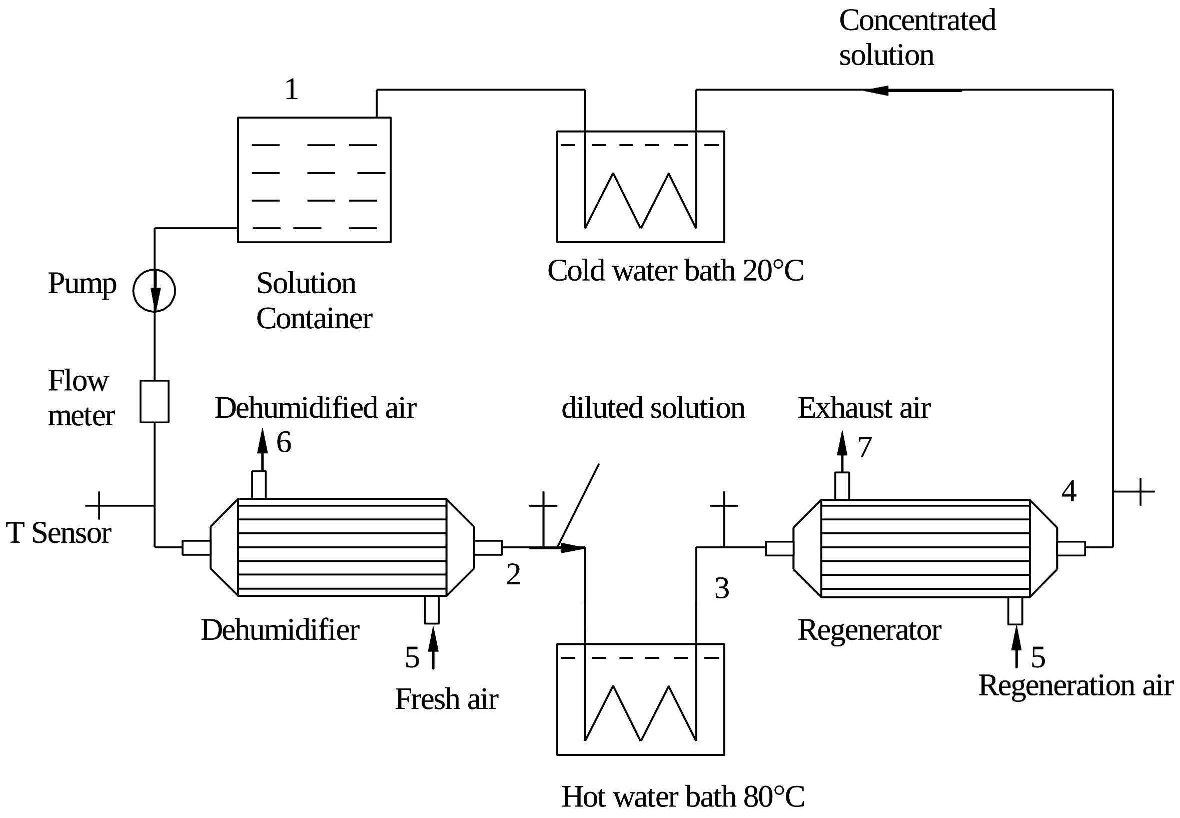

Figure 1 A hot water driven hollow fiber membrane liquid desiccant air dehumidification system

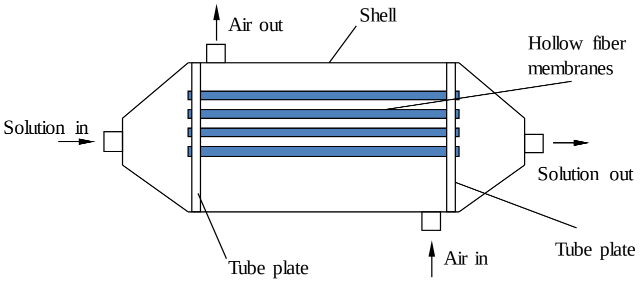

Figure 2 Structure of a hollow fiber membrane module

The structure of a single membrane module is shown in Fig.2. As seen, it is similar to a shell-and-tube heat exchanger. However, membrane tubes are used rather than metal tubes. Both heat and moisture can be transferred from one fluid to another through the membrane tube walls.

Heat and mass transfer in the membrane modules are schematically illustrated in Figure 3. Heat and moisture can be permselectively transported from air to liquid solution in dehumidification, or from liquid solution to air stream in regeneration. Liquid cannot travel through membranes.

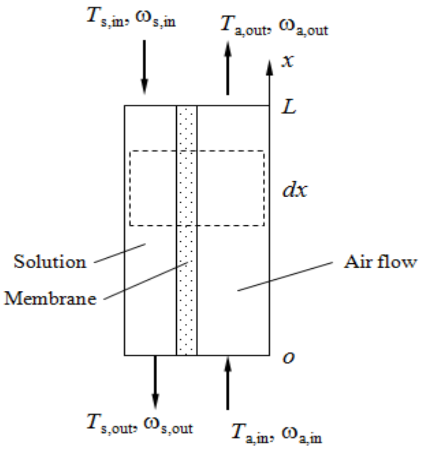

Figure 3 Heat and mass transfer through membranes

Heat and moisture conservation in the air stream are

$$\dot{m}_{a} c_{pa} \frac{dT_{a}}{dx} = \frac{h_{tot} A_{tot}}{L}\left(T_{s} - T_{a}\right) \tag{1}$$

$$\dot{m}_{a} \frac{d\omega_{a}}{dx} = \frac{\rho_{a} k_{tot} A_{tot}}{L}\left(\omega_{s} - \omega_{a}\right) \tag{2}$$

where in the equations, A_{tot} is the total area of the outer surface of fibers (m^2), x is axial coordinate (m) and L is fiber length (m).

In the equations, ω_s is the equilibrium humidity of air with solution at T_s and water mass fraction ($1-X_s$). In air dehumidification operations, the change in the liquid mass flow rate in the module can be neglected. Heat conservation in solution

$$\dot{m}_{s} c_{s} \frac{dT_{s}}{dx} = \frac{h_{tot} A_{tot}}{L} \left(T_{s} - T_{a} \right) + \dot{m}_{a} h_{abs} \frac{d\omega_{a}}{dx} \tag{3}$$

where in the equation, the first term on right side is convective heat transferred from solution to air, and the second term is absorption heat extracted by water desorption. Variable $\dot{m}_s$ is the total mass flow rate of solution (kg/s).

Equilibrium humidity of solution is a function of temperature and solute fraction, or

$$d\omega_{s} = \frac{\partial \omega_{s}}{\partial T_{s}} dT_{s} + \frac{\partial \omega_{s}}{\partial X_{s}} dX_{s} \tag{4}$$

Solution side pressure drop

$$\Delta p_{s} = f_{s} L \frac{\rho_{s}}{2} \frac{u_{s}^{2}}{d_{i}} \tag{5}$$

in which, f_s, u_s and d_i are friction factor for solution side, mass flux and inner diameter of fibers.

For laminar flow in round tubes, friction factor and Reynolds number satisfies [4]

$$f_{s} \mathrm{Re}_{s} = 64 \tag{6}$$

Air side pressure drop

$$\Delta p_{a} = N_{L} m_{2} f_{a} \frac{\rho_{a} u_{a,\max}^{2}}{2} \tag{7}$$

in which N_L is the number of fibers along the air flow direction, m_2 is a correction factor, f_a is friction factor across tube banks listed in, $u_{a,\max}$ is maximum mass flux in air flow direction.

Specific dehumidification power (*SDP*), and *COP* of the whole system are defined to evaluate the performances of this system. *SDP* and *COP* express the dehumidification ability of the device and they can be calculated by the following equations:

$$SDP = \frac{\rho_{a} V_{a} \left(\omega_{a,deh,i} - \omega_{a,deh,o} \right)}{A_{tot}} \tag{8}$$

$$COP = \frac{\dot{m}_{a} \left(e_{a,deh,i} - e_{a,deh,o} \right)}{\dot{m}_{w} c_{pw} \left(T_{het,i} - T_{het,o} \right)} \tag{9}$$

where ρ_a is air density, V_a is air volumetric flow rate and ω is humidity ratio, e is enthalpy (kJ/kg). Subscripts "a, s, deh, i, w" and "o, het" refer to air, liquid desiccant, dehumidifier, inlet, water and outlet, heater, respectively. The detailed calculation of other parameters can be found in [4].

Eqs. (1)-(4) are the governing partial differential equations for heat and moisture transfer in the module. As heat and mass transfer are coupled, they are solved by control volume based finite difference method. A grid independence test is found that 40×40 grids are enough for numerical accuracy (less than 0.1% difference compared to 50×50 grids). After the solution of the modules, the state points of the system are determined, and then the system performance is evaluated. Temperature, humidity, and solution concentration of liquid desiccant are governed by equilibrium thermodynamic graph as shown in Fig.3. The model for the module has been experimentally validated in [4]. Following are the performance analysis.

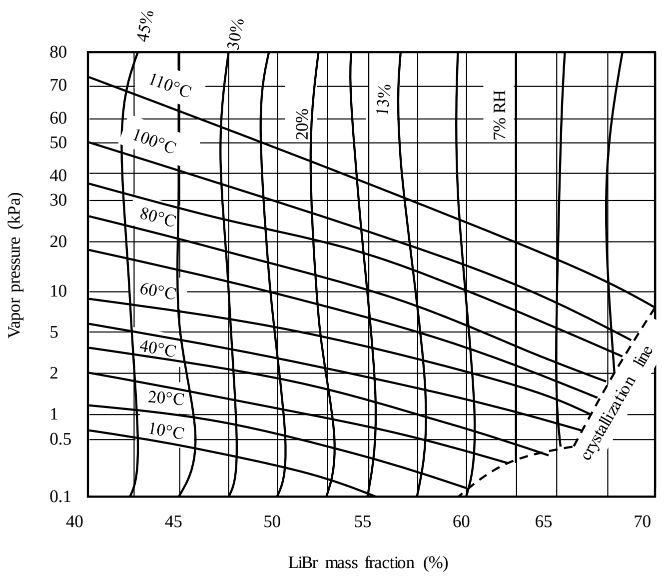

Figure 3 Thermodynamic state graph for LiBr solution

RESULTS AND DISCUSSION

The dehumidification system is simulated by the model and the operating conditions for the module are shown in Table 1. Temperature and relatively humidity of outdoor air are typical weather conditions in hot and humid Southern China. Volumetric air flow rate of 150m^{3}h^{-1} is selected for fresh air requirement of a 100m^2 office building with 4-5 occupants. The ambient fresh air is also used for solution regeneration. The volumetric flow rate is equal to that of outdoor fresh air for pressure balance in rooms. Inlet solution concentration is set to 0.4 and inlet temperature is set to 20℃ for the dehumidifier. Volumetric flow rate of heating and cooling water is 54 Lh^{-1} and the inlet solution temperature of dehumidifier is kept to 20℃.

Table 1 Geometrical and physical properties of the module

Parameters	Description or value
Dehumidifier/regenerator	
Module length (y_0)	380 mm
Module width (x_0)	200 mm
Fiber inner diameter (d_i)	1.1 mm
Fiber outer diameter (d_0)	1.5 mm
Total area (A_{tot})	7m^2
Bundle arrangement	Staggered
Membrane thickness (δ)	150μm
Moisture diffusivity (D_{vm})	1.6e-9m^2/s

Air flow rates are varied to study the effects of air flow rates on performances while other parameters are fixed. The regeneration air flow rates are equal to fresh air. The specific dehumidification power and COP under varying inlet air temperatures and humidity are plotted in Figs.5 and 6 respectively.

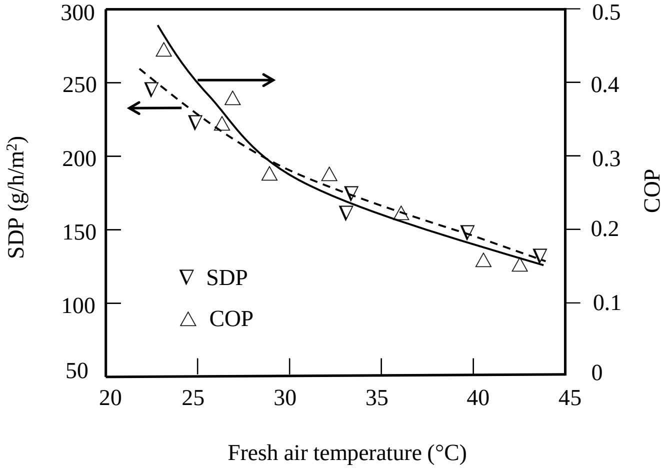

Figure 5 Effects of fresh air temperature on performance

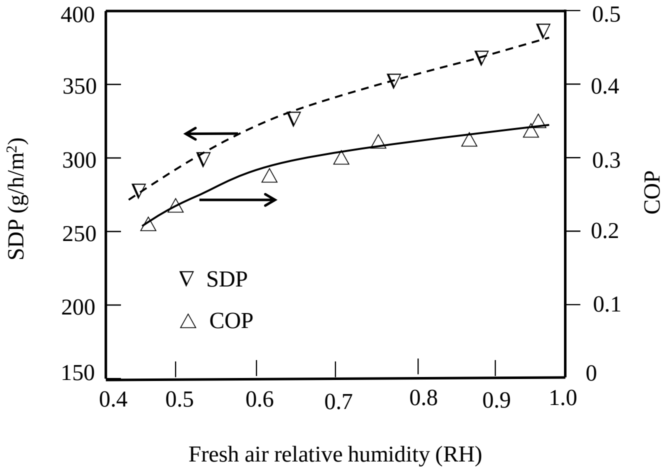

Figure 6 Effects of fresh air humidity on performance

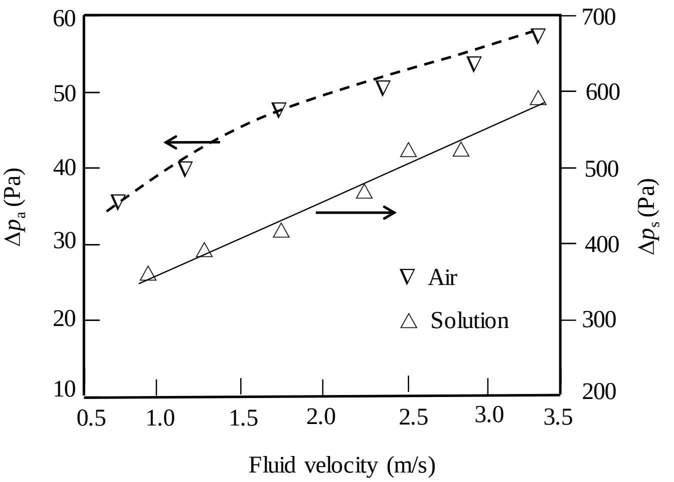

Figure 7 Effects of fluid velocity on pressure drop through the membrane modules

As outdoor air temperature changes from time to time, the influences should be considered. As seen, generally, the higher the air temperature is, the lower the *SDP* and COP are. The reason is that the working zone moves to a higher temperature one. The fluid cycles at higher temperatures. The ability to dehumidify air decreases. At the same time, the higher air temperature and the much more latent heat released during the dehumidification process all contribute to the rise in solution temperature. Therefore, the temperature of the liquid desiccant is higher and the coefficient of performance is lower. On the other hand, the higher solution temperature would make the regeneration temperature rise and consequently a lower performance. The SDP is about 150 to 300 kg/h/m^2, which indicates that the new system is compact and high efficient.

For humidity, the higher the fresh air humidity is, the higher the SDP and the COP are. This is because, the higher the fresh air humidity is, more moisture is dehumidified and the better the performance. Generally, the performance of the membrane system is comparable to a packed column air dehumidification system. However the liquid droplets crossover problem can be prevented.

The pressure drops through the modules with varying air and solutions flow rates are shown in Fig.7. Generally, the pressure drops are low. The pressure drop in air side is from 30 to 60Pa, while the pressure drop in liquid side is from 350 to 600 Pa. A 20W blower and a 5W plastic pump are used respectively to drive the air stream and the liquid stream. The pressure drop induced energy use is trivial compared to the energy saving by the low-grade hot water system. A test prototype is being built in our laboratory to help to commercialize the system. Detailed heat and mass transfer analysis for component design is conducted [5,6].

CONCLUSION

A mathematical model is developed to simulate a hot water driven and membrane-based liquid desiccant air dehumidification system. The membrane module is modeled in detail to describe the performances of the whole system. The system performances are investigated under different operating conditions. Following conclusions can be made:

The temperature and humidity of the fresh air have a tremendous impact on system performance. The higher the temperature is , the less the dehumidification capability and the COP.

For humidity, the higher the fresh air humidity is, the higher the SDP and the COP are. Generally, the performance of the membrane system is comparable to a packed column air dehumidification system. The pressure drops are acceptable. However the liquid droplets crossover problem can be prevented.

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