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A Study on Cooling and Dehumidification Process of a Heat Pump Drying System at Low Temperature

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ABSTRACT: The heat pump drying technology at low temperature has been become widespread due to advantages of product's quality in postharvest conditions, comparing to the conventional drying method. The efficiency of the heat pump dryer is predominately affected by cooling and dehumidification of moisture air moving through evaporator coils. Nevertheless, there is a lack of heat and mass transfer correlations for solving the problems. First attempts to fill the gaps of the heat and mass transfer problem are presented in this paper. A mathematical model is established from fundamental mass and energy equations. Numerical solutions of the model were solved by means of the Taylor series method. The results obtained in the study are the dehumidification of moist air through plain tube coils at the steady state. The other cases such as the dehumidification at finned-tube coils and dehumidification with air frost at the transient state will be shown in next papers.

INTRODUCTION

Heat pump dryers have the potential to operate more efficiently, and at lower temperatures than conventional dryers The application of heat pumps for drying has received continuous attention since it possesses two-fold beneficial characteristics. Heat is normally supplied to the product by heated air by natural means or artificially and the vapour pressure or moisture concentration gradient, and thus, created causes the movement of moisture from inside of the product to the surface. Excessive high temperature drying causes both physical and chemical changes ultimately deteriorate the quality of the product [1]. There are some heat sensitive materials such as medicinal and aromatic plants, those should be dry very carefully, otherwise the medicinal and aromatic qualities will be deteriorated which are the goal of the product. To maintain the quality, these heat sensitive crops should be dried at low temperature. The problems of low temperature drying are, it takes long time and during this drying period products may deteriorate caused by microorganisms specially fungus [2].

Through the evaporator in the heat pump dryer, sensible and latent heat from the dryer are exhausted, and the energy is recovered. Condensation the vapour of water in moist air occurring at the surface of the evaporator reduces absolute humidity of the working air, and thus, increasing the effectiveness of product drying. Therefore, the cooling and dehumidification processes of moist air play a role in the system's efficiency. However, there is a lack of correlations for describing heat and mass transfer in the evaporator, due to the complication of the phenomena. For instance, a researcher developed simulation models of heat pump dryer by means of the finite-difference method to examine the state of the working fluids and heat and mass transfer [3]. Another researcher established a mathematical model and simulation code to investigate the performance of a transcritical CO_2 heat pump dryer [4]. The model takes into account detailed heat and mass transfer and pressure drop phenomena occurring in each component of the system.

In the paper, a model based on fundamental mass and energy equations is formed. Numerical solutions of the model were solved by the Taylor series method. The results obtained in the study are the dehumidification of moist air through plain tube coils at the steady state. The other cases such as the dehumidification at finned-tube coils and dehumidification with air frost at the transient state will be shown in next papers [5, 6].

MODELING OF THE COOLING AND DEHUMIDIFICATION

Objectives

Fig. 1 shows the mechanism of heat and mass transfer when moist air moves through cold water outside of a tube. On top of the cold water surface, a thin layer of saturated (or near saturated) moist air is formed. As the water is colder, the layer has saturated vapor pressure lower than vapor pressure of the air. The difference between these pressures causes water to be condensed onto the colder water surface.

In order to establish a model, the following assumptions have been made as follows:

- Thermal properties of most air are constant
- Steady state
- The model is one-dimensional (refrigerant boils at the given temperature, i.e. t_r=const)
- Thermal resistance of condensate water is negligible.
- Heat infiltration from the environment is ignored.

The main purpose of the model is to investigate the change of thermodynamics properties of the air (e.g. temperature, humidity, etc), while varying the external conditions along the coil and to subsequently determine the air's outer parameters values.

Heat and mass transfer equations

Considering air flow exchanges energy to a cold water surface (i.e. Fig. 1). On top of the cold water surface, a thin layer of saturated (or near saturated) moist air is formed. Since the water is colder, the layer being saturated vapor pressure is lower than that of the air. The difference of the two partial pressure causes water to be condensed on the tube's surface [7-10].

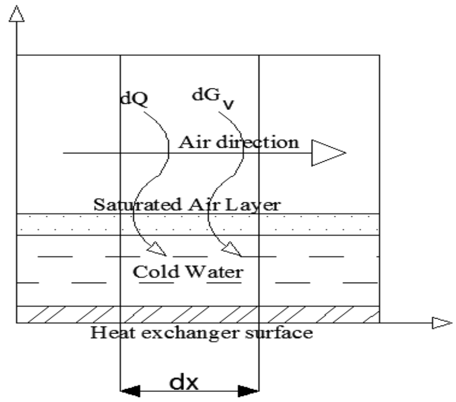

Figure 1. Heat and mass transfer mechanism of moist air outside the tube.

The air is cooled by 2 ways. One is by convective heat transfer caused by the temperature differences between the air and the water surface, and another one is due to latent heat by water condensation.

Mass transfer equation

The equation of mass transfer can be written in term of partial pressure:

$$dG_v = -\beta(W - W_s)dF \qquad (1.1)$$

To address to the x-coordinate, we introduce function $f(x)$:

$$f(x) = \frac{dF(x)}{dx} \qquad (1.2)$$

Ignoring the thermal resistance of the condensate film, the temperature of the coil's outer surface is assumed being equal to the condensate layer surface temperature. The relationship can be shown by the Antoine expression:

$$P_s = \exp\left(12 - \frac{4026.4}{t_w + 235.5}\right) \qquad (1.3)$$

Absolute humidity ratio air flow is calculated by the formula (number 0.95 means that the layer is only in 95% saturated condition)

$$W_s = 0.621\frac{0.95P_s}{P_a - 0.95P_s} \qquad (1.4)$$

Combine equations (2.2) and (2.4) to (2.1), with W = G_v/G_d the mass transfer equation is read:

$$\frac{dG_v}{dx} = -\beta\left(\frac{G_v}{G_d} - 0.621\frac{0.95\exp\left(12-\frac{4026.4}{t_w+235.5}\right)}{P_a - 0.95\exp\left(12-\frac{4026.4}{t_w+235.5}\right)}\right)f(x)(1.5)$$

Heat transfer equation

Heat transfer is caused by the change of the air flow enthalpy:

$$dQ = G_d dh \qquad (2.1)$$

From the left side of the equation (2.6):

$$dQ = -\alpha_1(t - t_w)dF + r\,dG_v \qquad (2.2)$$

The first term of the equation (2.7) is sensible heat transferred by convection and the second one is latent heat of condensation process.

The right side of equation can be written by the specific enthalpy dh:

$$G_d dh = G_d\left(C_{pd}dt + r\frac{dG_v}{G_d} + C_{pv}t\frac{dG_v}{G_d} + C_{pd}dt\frac{G_v}{G_d}\right) (2.3)$$

Substituting equations (2.7) and (2.8) into equation (2.6), with $dF = f(x)dx$ and then simplifying to get the following equation:

$$\frac{dt}{dx} = \frac{-\alpha_1(t - t_w)f - C_{pv}t\frac{dG_v}{dx}}{G_d C_{pd} + G_v C_{pv}} \qquad (2.4)$$

Heat balance

In the model the equations are derived by balancing heat transferred from the air to the coil outer surface and from there to refrigerant boiling inside of the coil:

$$k_{\alpha_2}(t_w - t_r)dF = \alpha_1(t - t_w)dF - rdG_h \qquad (3.1)$$

We change equation (2.10) to the algebraic form as:

$$\frac{\alpha_1 + k_{\alpha_2}}{r}t_w - \beta 0.621\frac{0.95\exp\left(12-\frac{4026.4}{t_w+235.5}\right)}{P_a - 0.95\exp\left(12-\frac{4026.4}{t_w+235.5}\right)} = -\beta\frac{G_v}{G_d} + \frac{k_{\alpha_2}}{r}t_r + \frac{\alpha_1}{r}t \ (3.2)$$

Function f

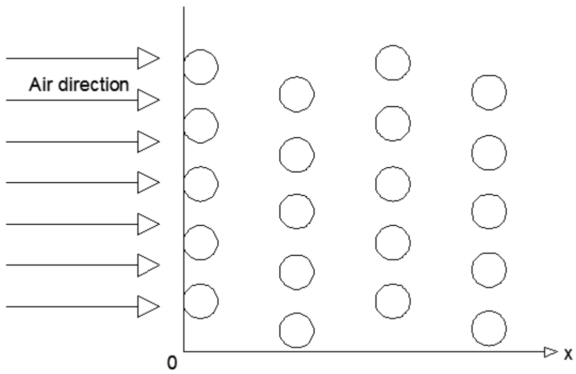

Figure 2. Configuration of the coil and air direction

In the case plain tube coil the function $f(x)$ can be written as:

$$\mathrm{f(x)} = \begin{cases} 2Rnl\dfrac{1}{\sqrt{R^2-\left((2k-1)R+(k-1)s-x\right)^2}} \\ \forall(\ 2(k-1)R+(k-1)s \le x \le 2kR+(k-1)s) \\ 0 \quad \forall(2kR+(k-1)s) \le x \le 2kR+ks) \end{cases} \tag{4}$$

Boundary condition

The boundary condition is the air temperature and absolute humidity before entering the coil: $G_v(x=0)$ and $t\ (x=0)$.

Numerical solution

The differential equations were solved numerically by the Taylor series method with the 1st degree of accuracy and simulated by the Matlab software:

$$G_v(x_o+h) = G_v(x_o) + \Delta x\frac{dG_v}{dx} \tag{6.1}$$

$$t(x_o+h) = t(x_o) + \Delta x\frac{dt}{dx} \tag{6.2}$$

A flow chart of the basic calculation procedure in the model of the heat and mass transfer with moist air as working fluid is presented in Fig. 3.

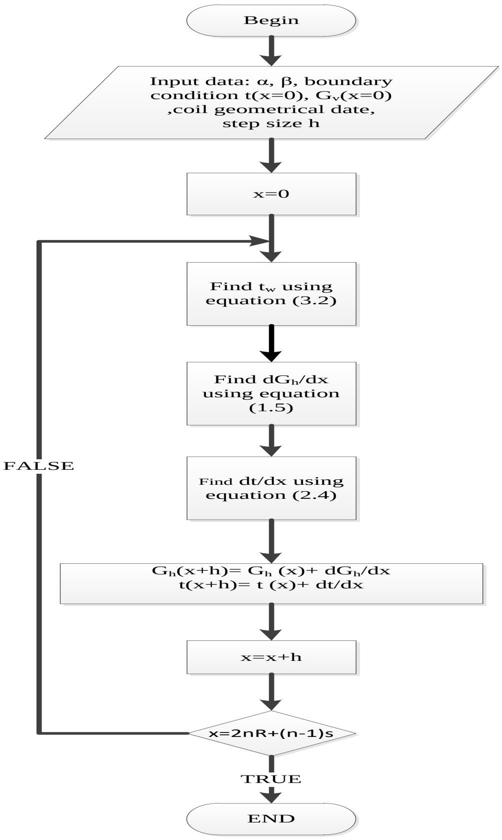

Figure 3. Flowchart of the simulation program

RESULTS AND DISSCUSIONS

Inputs of the model

The inputs parameters given in Table 1 and 2 have been kept constant. The model has been investigated for two example cases. The variation of several quantities along the flow length with different inputs has been studied.

Table 1. Input of the model for the 1st case.

No.	Parameter	Unit	Value
1	Air inlet temperature	ºC	32
2	Air inlet relative humidity	%	50
3	Air inlet mass flow rate	kg/s	0.05
4	Tube's outer diameter	cm	1
5	Length of tube each row	m	0.5
6	Number of tubes each row	-	6
7	Number of rows	-	6
8	Distance between two row	cm	3
9	Refrigerant boiling temperature	ºC	3
10	Air-side heat transfer coefficient	W/m^2K	60
11	Mass transfer coefficient	kg/m^2s	0.086
12	Overall heat transfer coefficient of tube and refrigerant	W/m^2K	1000

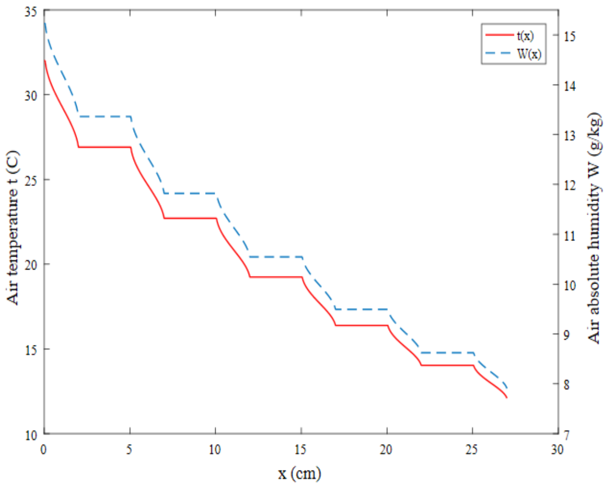

Figure 4. Variation of absolute humidity and temperature of moist air along the coil for the 1st case.

Fig. 4 and 5 present the change of the parameters: air temperature and absolute humidity when air goes through the evaporator coil with different input data. Generally speaking, both cases have a similar trend. At the same time, the temperature and humidity behaviors (i.e. $t(x)$ and $W(x)$) from each case are both similar and equivalent to another one.

Table 2. Input of the model for the 2nd case.

No.	Parameter	Unit	Value
1	Air inlet temperature	°C	25
2	Air inlet relative humidity	%	60
3	Air inlet mass flow rate	kg/s	1
4	Tube's outer diameter	cm	1
5	Length of tube each row	m	1
6	Number of tubes each row	-	16
7	Number of rows	-	20
8	Distance between two row	cm	2
9	Refrigerant boiling temperature	°C	2
10	Air-side heat transfer coefficient	W/m^2K	60
11	Mass transfer coefficient	kg/m^2s	0.086
12	Overall heat transfer coefficient of tube and refrigerant	W/m^2K	1000

As the air moves through the coil, both temperature and specific humidity line reduce upwards to x. The decrease is non-continuous. Temperature and humidity of the air flow decrease only while staying in contact with the coil's outer surface (i.e. $f(x) \neq 0$). In the empty region between two rows ($f(x) = 0$), these parameters remain constant. In the regions in which the heat and mass transfer occurs under the effect of function $f(x)$, the temperature and humidity graph's slope gradually decline from 90° to minimum at the center of the tube center, before increasing again to 90° at the end of the tube. However, the slope does not increase again after the tube's center, once the heat and mass transfer became less effective due to the flow mechanics. The obtained result is due to the assumption heat and mass transfer coefficient being constant with coordinate of x.

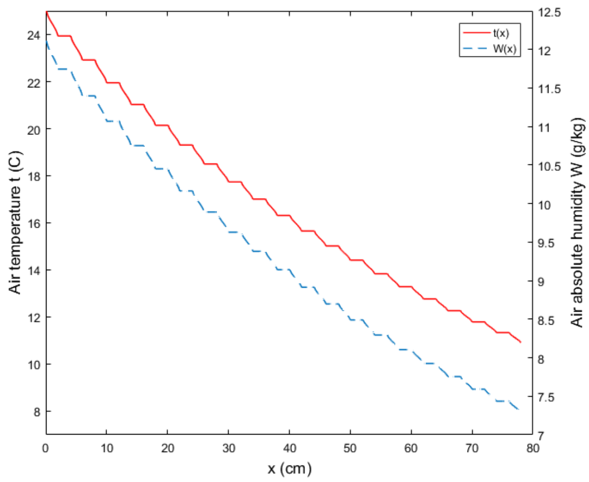

Figure 5. Variation of absolute humidity and temperature of moist air along the coil for the 2nd case.

Furthermore, both $t(x)$ and $W(x)$ curves (when decrease) are steeper in area nearer to the base point than posterior area (for instance, in the 1st case, the temperature drops in the 1st row is 5 °C, while in the last row is 1.6 °C). This can be explained as the further the air flows inside the coil, the colder and the drier the air achieved, which cause temperature and humidity differences between the air and the coil's surface to be decreased as well. Since both these two differences are two main factors behinds heat and mass transfer process, their decrease cause heat/mass transfer to be less effective than before. This effect cause both the $t(x)$ and $W(x)$ have the forms relatively similar to the exponential function curve, which is commonly found in the open literature.

CONCLUSIONS

The heat and mass transfer in the evaporator (dehumidifier) has been studied in this work for a better understanding of the phenomena in the heat pump dryers. Based on the fundamental equations of heat and mass balances, a mathematical model was established. The model has the advantage of being simple and compact, and thus, requires a short simulation time. Some different cases have been investigated as changing the input data for the model. The results show the temperature and absolute humidity behaviors changing with different factors when it is cooled and dried through the coils of the evaporator. Nevertheless, several assumptions are made at the cost of accuracy, especially the constant heat and mass transfer coefficient assumption. The other cases such as the dehumidification at finned-tube coils and dehumidification with air frost at the transient state will be shown in next papers.

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NOMENCLATURE

G	Mass flow rate (kg/s)
W	Absolute humidity (kg/kg)
β	Mass transfer coefficient (kg/m^{2}s)
F	Coil's outer surface area (m^2)
t	Temperature (°C)
φ	Relative humidity (%)
h	Specific enthalpy (J/kg)
c_p	Specific heat (J/kg·K)
r	Latent heat (J/kg)
α_1	Heat transfer coefficient of air (W/m^{2}K)
$k_{\alpha 2}$	Overall heat transfer coefficient of refrigerant and tube (W/m^{2}K)
Δx	Calculation step size (mm)
R	Tube outer radius (m)

n Number of tubes each row
l Tube's length each row (m)
s Distance between 2 rows (m)
N Number of rows

Subscripts

- Moist air flow
s Saturated moist air layer
v Water vapor
d Dry air
w Coil's surface
r Refrigerant

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