CHAPTER III

HEAT PUMP DRYER

The conventional heat pump dryer (without loop thermosyphon) is shown in Figure 3.1, which consists of a heat pump system and a conventional dryer system. The heat pump system consists of a compressor, a condenser, an expansion valve and an evaporator. The conventional dryer consists of a blower, a heater and a dryer. In this chapter, the characteristic of heat pump dryer is studied, and how each part has a relationship and affects the other part. This study of heat pump dryer involves modification and construction of the mathematical model and evaluation of simulation results through comparison with experimental data.

Figure 3.1 Conventional Heat pump Dryer

(CV: Control volume)

This chapter can be separated two parts: the first part is the construction of the mathematical model of heat pump system. This model is used to simulate the optimum condition of heat pump for the product. The model has been developed using the heat transfer equation of heat pump which uses R-12 is working fluid. Vongvanichakul (1997) developed the kinetics of drying whole longan model, which was applied with the heat pump model. The details of equations and assumptions for constructing the model are presented below. The second part of this chapter describes the construction of heat pump dryer and testing the characteristic of the heat pump dryer to evaluate the model and compare the results of the heat pump dryer with and without loop thermosyphon interms of energy saving.

3.1 Mathematical Model

For the design of the heat pump system and to study the characteristics of the heat pump dryer, whole longan was selected as the material. The 120-kg whole longan was the capacity selected for this research. This scale is suitable for verifying the program. Quality can be controlled uniformly, since one single drying unit can supply this amount per batch.

3.1.1 Objectives of the mathematical model

3.1.1.1 To construct a simulation model, which can be used to simulate variable parameters for the design and construction of the heat pump dryer.

3.1.1.2 To simulate the heat pump dryer for the minimum energy consumption at any necessary condition and establish optimum condition for the design and construction of the LTS.

3.1.2   Scope of the mathematical model

For the mathematical model, following assumptions are made:

3.1.2.1 Thermal equilibrium exists between moist air and whole longan.

3.1.2.2 The wall of the cabinet dryer is adiabatic.

3.1.2.3 The internal energy change of the cabinet dryer and whole longan are negligible.

3.1.2.4 The drying air which exits the dryer has relative humidity less than 100%, if more than 100% the program will change the conditions for drying.

3.1.2.5 The initial and final moisture content of 300% and 25 % dry basis is suggested.

The variable parameters were:

-The airflow rate varies from 400 to 750 m^3/hr, in steps of 50 m^3/hr. (for 120-kg whole longan), which is dependent on the product.

-The recirculation air ratio (RAR) varies from 0 to 100%, in step of 10%.

-The ambient air varies from 25 to 40 ℃, in step of 5 ℃.

-The by-pass air ratio varies from 0 to RAR-5%, in step of 5%.

-The relative humidity varies from 20 to 80 %, in step of10%.

-The drying temperature varies from 70 to 80 ℃, in step of 5 ℃.

3.1.3   Structure of the mathematical model and computer program

The mathematical model of the heat pump dryer system has been modified and constructed for studying the characteristics of heat pump system. The model can be separated into two main parts: A) the conventional dryer model, B) the heat pump system model. Particular attention is given for developing a theoretical model of heat pump dryer. Figure 3.1 shows the schematic diagram of a heat pump dryer. The details of the conventional dryer model and heat pump system model are as follows:

A) Conventional dryer model

The drying characteristics depend on the type of dryer and product. In this study, the drying chamber model for cabinet drying of whole longan is developed. The drying kinetic equation developed by Vongnichakul (1997) is shown below:

$$M_f = M_i - (M_{in} + M_{eq})\Delta tK \exp(-Kt) \tag{3.1}$$

$$K = 0.0023(T_{di} + 273.15) - 0.739 \tag{3.2}$$

$$M_{eq} = A[RH_{di}/(1 - RH_{di})]^B \tag{3.3}$$

$$A = 2.3015 - 0.00615 (T_{di} + 273.15) \tag{3.4}$$

$$B = -1.3453 + 0.00507 (T_{di} + 273.15) \tag{3.5}$$

Where,

M_f = moisture of whole longan after drying (dry basis, dicimal)

M_i = moisture of whole longan before drying (dry basis, dicimal)

M_{in} = moisture of whole longan begin drying (dry basis, dicimal)

M_{eq} = equilibrium moisture content of whole longan (decimal)

Δt = time interval, hour

t = drying time , hour

K = constant

T_{di} = drying temperature, ℃

RH_{di} = relative humidity, decimal

The moisture content of whole longan during drying can be calculated by Equation 3.1 with respect to the time. The moist air properties at outlet drying chamber can be calculated by using the principle of mass and energy conservation to control volume1 in Figure 3.1.

From mass conservation, the increase of moisture air is equal to the decrease of moisture content of the product. It can be written as

$$\Delta t.m_d(w_{do} - w_{di}) = m_p (M_i - M_f) \tag{3.6}$$

Where, Δt is time interval, m_d is air flow rate, w_{do} is the humidity ratio of air leaving drying chamber, w_{di} is the humidity ratio of air entering drying chamber, M_i is moisture content before drying, M_f is moisture content after drying and m_p is the dry mass of whole longan.

From the energy conservation, it is assumed that moist air and drying product are in thermal equilibrium. The equation can be derived from the first law of thermodynamics. It is given by Equation 3.7.

$$m_d \cdot [(C_a T_{di} + w_{di}(h_{fg} + C_v T_{di}) - C_a T_{do} - w_{do}(h_{fg} + C_v T_{do})] = 0 \qquad (3.7)$$

Where, T_{di} and T_{do} are the drying air temperature of air entering and leaving drying chamber, respectively. C_a and C_v are the specific heat capacities of air and vapor, respectively. Furthermore, there are minor equations derived from physical properties of moist air.

B) Heat pump model

The pressure-enthalpy diagram shown in Figure 3.2 is applied in this study to develop a heat pump model. There are two circuits in the system: the drying air circuit and the refrigeration circuit. The heat pump model has three major models: evaporator model, condenser model and compressor model. The details of each component are as follows:
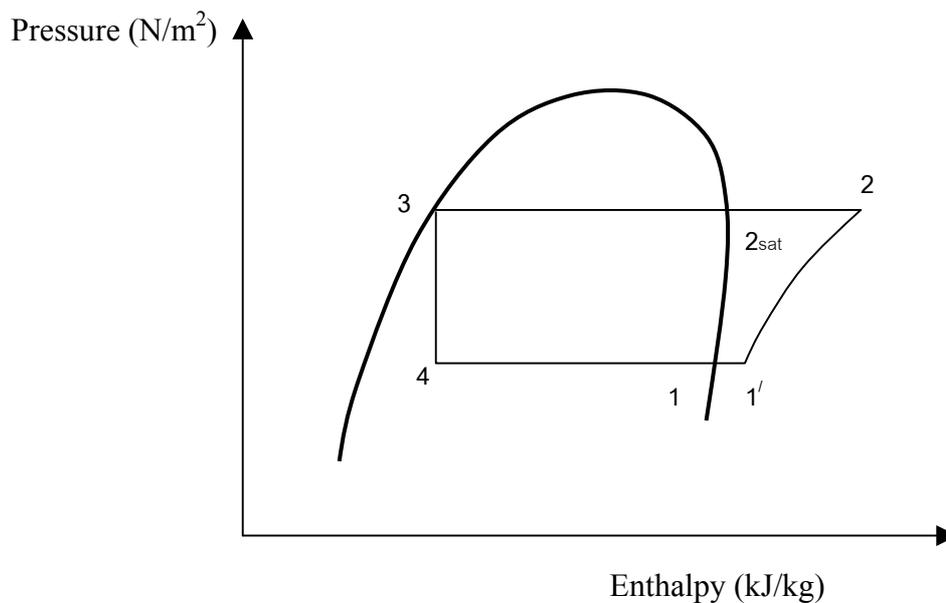
Figure 3.2 Pressure-enthalpy diagrams.

a. The evaporator model

The evaporator model can be derived from the performance data and the equations of heat transfer by using the principle of mass and energy conservation. It can be developed from the equation of heat transfer in both drying air and refrigerant circuits by using the first law of thermodynamics to the control volume 3 in Figure 3.1. Heat rejected from drying air during the process from state 4 to state 1 (see Figure 3.2) is equal to the heat absorbed by the refrigerant. It is written as:

$$Q_{ae} = Q_{re} \tag{3.8}$$

Where, Q_{ae} is the rate of heat rejected from drying air at evaporator and Q_{re} is the rate of heat absorbed by refrigerant at the evaporator. They are given by Equations 3.9 and 3.10

$$Q_{ae} = (RAR - BAR)m_d[C_a(T_{eo} - T_{mix}) + (w_{eo} - w_{mix})h_{fg} + w_{mix}C_v(T_{eo} - T_{mix})] \tag{3.9}$$

$$Q_{re} = m_r.(h_1 - h_4) \tag{3.10}$$

Where,

RAR	is recirculation air ratio
BAR	is the by pass air ratio
w_{eo}	is the humidity ratio of air leaving evaporator
w_{mix}	is the humidity ratio of mixed air before entering evaporator
T_{eo}	is the temperature of air leaving evaporator
T_{mix}	is the temperature of mixed air before entering evaporator
h_{fg}	is latent heat at reference temperature of 0 °C
m_r	is mass flow rate of refrigerant
h_1, h_4	are specific enthalpy of refrigerant leaving and entering evaporator, respectively.

The equation of heat transfer by using the principle of mass and energy conservation to the control volume 3 shown in Figure 3.1.is given by:

$$(RAR - BAR)m_d W_{eo} =$$
$$(RAR - BAR)m_d(1 - BF)W_{con} + (RAR - BAR)m_d(BF)W_{do} \tag{3.11}$$

Where, w_{con} is the humidity ratio of drying air with contacting cooling coils which can be calculated from Equation 3.12.

$$W_{con} = 0.62189(P_{vs(con)}/(101.325 - P_{vs(con)})) \tag{3.12}$$

The temperature of drying air with contacting cooling coils (T_{con}) is equal to the dew point. It is given by:

$0\ ^{\circ}\text{C} \geq T_{con} \geq -50\ ^{\circ}\text{C}$:

$$T_{con} = 5.994 + 12.4\ln(P_{vs(con)}) + 0.427[\ln(P_{vs(con)})]^2 \tag{3.13}$$

$50\ ^{\circ}\text{C} \geq T_{con} \geq 0\ ^{\circ}\text{C}$:

$$T_{con} = 6.983 + 14.38\ln(P_{vs(con)}) + 1.079[\ln(P_{vs(con)})]^2 \tag{3.14}$$

The corresponding energy balance yields

$$(RAR - BAR)m_d[C_a T_{eo} + W_{eo}(h_{fg} + C_v T_{eo})]$$
$$= (RAR - BAR)(BF)m_d[C_a T_{do} + W_{do}(h_{fg} + C_v T_{do})] \tag{3.15}$$
$$+ (RAR - BAR)(1 - BF)m_d[C_a T_{con} + W_{con}(h_{fg} + C_v T_{con})]$$

Where, T_{con} is the temperature of air contacting cooling coils, BF is by pass factor which depends upon the cooling coil construction, that is, the size of tube, size and type of fin, and the tube and fin spacing. For this type of cooling coil, the bypass factor of 0.2 is used.

$$(RAR - BAR)m_d W_{eo} =$$
$$(RAR - BAR)m_d(1 - BF)W_{con} + (RAR - BAR)m_d(BF)W_{do} \tag{3.16}$$

The BF is bypass factor, which depends upon the cooling coil construction, that is, the size of tube, size and type of fin and the tube and fin spacing.

Heat transfer rate at the evaporator with phase change of refrigerant is also applied for calculate on. It can be written as:

$$Q_e = U_e A_e [(T_{do} - T_{er}) - (T_{eo} - T_{er})]/[\ln\{(T_{do} - T_{er})/(T_{eo} - T_{do})\}] \quad (3.17)$$

Where, Q_e is heat transfer rate at evaporator (watt), A_e is area of evaporator (m^2), T_{do}, T_{eo} and T_{er} are temperature out dryer(in evaporator), out evaporator and working fluid, respectively.

$$Q_e = \dot{m}(h_{ei} - h_{eo}) \quad\quad\quad (3.18)$$

Where, $\dot{m}$ is the mass flow rate of working fluid (kg/s), h_{ei} and h_{eo} are at enthalpy entry and exit of the evaporator.

b. Condenser model

The condenser model can be developed by using the principle of energy conservation of the control volume of heat pump system. The control volume 2 (CV.2) in Figure3.1, heat reject from refrigerant during the process from state 2 to state 3 (see Figure 3.2) is equal to heat absorbed by drying air. It is written as

$$Q_{rc} = Q_{aci} \quad\quad\quad (3.19)$$

$$Q_{rc} = \dot{m}(h_{co} - h_{ci}) \quad\quad\quad (3.20)$$

$$Q_{aci} = m_d[(C_a(T_{co}-T_{ci}) + w_{ci} C_v(T_{co}-T_{ci})] \quad\quad\quad (3.21)$$

Where, Q_{aci} is the rate of heat rejected from the drying air at the condenser and Q_{rc} is the rate of heat absorbed by the refrigerant at condenser. w_{ci} is the humidity ratio of

air entering condenser, w_{co} is the humidity ratio of air leaving condenser, T_{ci} is the temperature of air entering condenser, T_{co} is the temperature of air leaving condenser, h_{ci} and h_{co} are specific enthalpy of refrigerants entering and leaving condenser, respectively.

c. Compressor model

The compressor model is given in equation 3.22

$$P = Q_c - Q_e \qquad\qquad (3.22)$$

Where, P is power consumption in the compressor and a_1 to a_9 are constants. The computer program used in this thesis was a Pascal program. When the specified data entered, the program could compute and present the result in form of a parametric profile.

The personal computer used for writing the program consisted of various components, they are:

- A windows' XP Operating system
- A 1.8 GHz CPU. Pentium III
- 256 MB of RAM
- A 20 GB Hard disk
- A Pascal Computation program

3.2 The flow chart of the program computer

The computer program of this part consists of three units. The first flow chart of the program as show in Figure 3.3 is used to calculate the energy consumption of drying process, needed for the design specification of the heat pump. The second flow chart of the program as shown in Figure 3.4 is used to calculate the specific energy consumption of the heat pump dryer. The third flow chart of the program as shown in Figure 3.5 is used to determine the minimum specific energy consumption of the heat pump dryer. All of the calculations in these programs can be obtained using the following steps.

3.2.1 The first flow chart programs. (Shown in Figure 3.3)

This procedural chart is used to calculate the energy consumption of the system. After defining the specification of the heat pump such as working fluid (R-12) the following steps are taken:

3.2.1.1 Receiving the data entered from the user (whole longan in this case).

3.2.1.2 Checking the condition of humidity ratio, relative humidity at the exit, if correct then calculates the results. Next is to check if the moisture of the product is equal to the specified value, if it is agrees record the result and stop the program.

3.2.2 The second flow chart program (Shown in Figure 3.4).

This procedural chart is used to calculate the specific energy consumption of the heat pump system.

3.2.3 The third flow chart program (Shown in Figure 3.5).

This procedural chart is used to simulate the optimum specific energy consumption and condition for the construction and testing of the heat pump dryer. The next step is to define the condition for the loop thermosyphon. Each step of the simulation is as follows:

For studying the effect of the variable in the operation of the heat pump and determining the optimum specific energy consumption of the heat pump dryer, following parametric values were used:

Mass of the product:

- 120 kg whole longan

The variable parameters were:

-Ambient temperature (Ta) to vary from 25 to 40 ℃, in step of 5 ℃.

- Relative humidity (RH) varying from 20 to 80 %, in step of 10%.

- Drying temperature (Tdi) varying from 70 to 80 ℃, in step of 5 ℃.

- Air flow rate (Md) varying from 400 to 750 m^3/hr, in step of 50 m^3/hr.

- Recirculation air ratio (RAR) varying from 0 to 100%, in step of 10%.

- By-pass air ratio (BAR) varying from 0 to RAR-5%, in step of 5%.

The steps of the simulation model are:

3.2.3.1 Input the initial data for the heat pump dryer.

3.2.3.2 Calculate the specific energy consumption of the heat pump system. After that change the condition for new result of specific energy consumption. Then compare the result and select the minimum value. Record the result. For simulating the suitable condition for construct the loop thermosyphon, the minimum specific energy consumption of the heat pump dryer was considered. The parameters specified are:

The controlled parameters are:

- Mass, 120 kg whole longan

-Ambient temperature (Ta), 30 ℃

- Relative humidity (RH), 70 %.

-Drying temperature (Tdi), 75 ℃

The variable parameters are:

- Airflow rate (Md) used two, three and four time of the optimum value.

- Recirculation air ratio (RAR) varying from 0 to 100%, in step of 10%.

- By-pass air ratio (BAR) varying from 0 to RAR-5%, in step of 5%.