CHAPTER V

APPLYING LOOP THERMOSYPHON IN A HEAT PUMP DRYER

5.1 Mathematical Model

For studying the heat transfer characteristics of the heat pump dryer with the loop thermosyphon, the mathematical models were connected between the heat pump dryer model (in Chapter III) and loop thermosyphon model (in Chapter IV). The entire program was connected and used to study characteristics of heat transfer of heat pump dryer, and to establish the minimum specific energy consumption. After that compare the energy consumption of heat pump dryer with loop thermosyphon (in this chapter) and heat pump dryer without loop thermosyphon (in Chapter III).

5.1.1 Objectives of the mathematical model

To construct simulation model which can be used to establish the minimum specific energy consumption of the heat pump dryer with loop thermosyphon.

5.1.2 Scope of the mathematical model

The assumptions of mathematical modeling of the heat pump dryer with loop thermosyphon are:

5.1.2.1 Thermal equilibrium exists between moist air and whole longan.

5.1.2.2 The wall of the cabinet dryer is adiabatic.

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

5.1.2.4The drying air which exit the dryer have relative humidity less than 100%, if more than 100% the program will change the conditions for drying.

5.1.2.5The initial and final moisture content of the product are 300 and 25 % dry basis.

5.1.2.6 The heat pump dryer model with loop thermosyphon has the following control parameter, which are the same as of the heat pump without loop thermosyphon described in Chapter III as follow:

-The ambient temperature was 30 °C

-The by-pass air ratio was 15%

-The relative humidity was 70 %.

-The drying air temperature was75 °C

The variable parameters are:

-The airflow rates of 937, 1350 and 1620 m^3/hr (this is the same of the heat pump dryer without loop thermosyphon).

-The recirculation air ratio (RAR) of 30, 35 and 40%.

5.2 Structure of the mathematical model and computer program

The mathematical model of the heat pump dryer with loop thermosyphon system has been modified and constructed for studying the heat transfer characteristics.

Particular attention is given for developing a theoretical model of heat pump dryer with loop thermosyphon by connecting the heat pump dryer model with the loop thermosyphon model. Figure 5.1 shows the schematic diagram of the main program of the heat pump dryer with loop thermosyphon. The details of the step for simulation model as follows:

5.2.1 Entry of data from the user for optimum heat pump dryer and loop thermosyphon

5.2.2 Optimum heat pump dryer with out loop thermosyphon for the minimum specific energy consumption

5.2.3 Optimize the loop thermosyphon for the maximum energy in relation to cost of construction (E by C)

5.2.4 Optimize heat pump dryer with loop thermosyphon for minimum specific energy consumption

5.2.5 Store the data and parametric values for maximum E by C and minimum specific energy consumption.

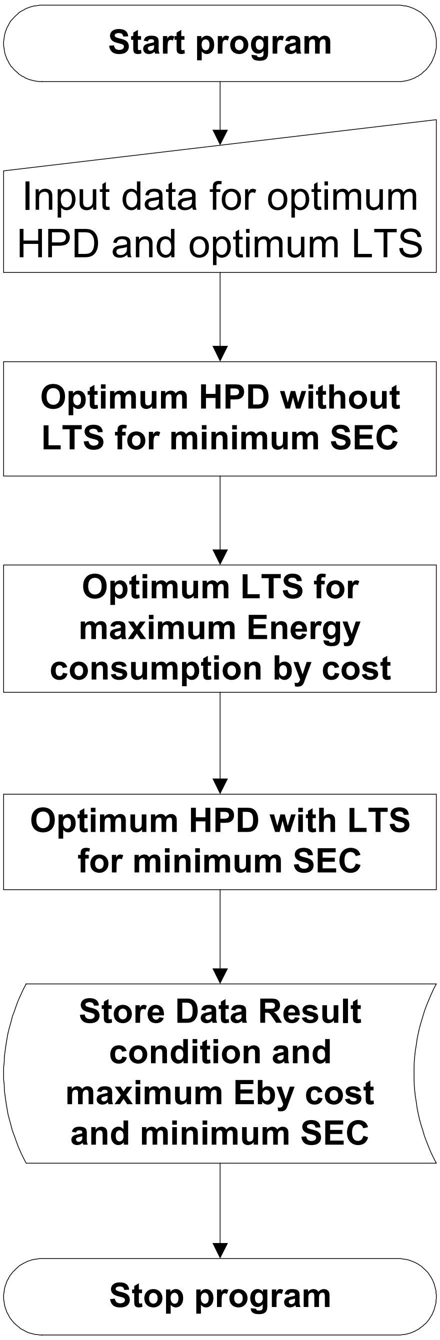

Figure 5.1 Schematic diagram of flow chart of heat pump dryer with loop thermosyphon.

5.3 Experimental study

5.3.1 Objectives

5.3.1.1 To experimentally investigate the heat transfer characteristics of the heat pump dryer with the loop thermosyphon.

5.3.1.2 To verify the mathematical model by the experimental results for the specific energy consumption of the heat pump with loop thermosyphon.

5.3.2 Scope of the experiment work

The scope of the experimental work is the same as of simulation model because it must use the results from the experiments to verify the simulation program. After constructing the loop thermosyphon, it was evaluated and installed in the HPD as shown in Figure 4.7

All the temperature and velocity sensors were installed in the system to monitor the necessary data. TESTO model 454 with accuracy ± 0.4 °C and $\pm$ 2 %RH was used to record the temperature and RH. In addition, the type-K thermocouples were also employed with the COMARK C8510 data logger to monitor the temperature at all the points. The consumed electrical power was measured using DIGICON MG51 wattmeter with an accuracy of ± 1%. The air velocity was also recorded by using TSI model 8384-M flow meter with ± 3% accuracy.

The experiment began by starting the heat pump and blower. The circulating air was heated when it passed through the condenser of the HPD. The electric heater ensured the desired operating temperature. This hot and dry air then passed through the dryer and resulted in having lower temperature and higher humidity. This relatively cooler and damp air then passed through the evaporator section of the LTS and rejected heat to the evaporator section of the LTS. The cooler air then rejected some moisture once it passed though the evaporator of the heat pump. The exit air from the evaporator section was cool and dry. This cool and dry air gained some heat as it passed through the condenser section of the LTS. The relatively hotter and dry air then was recirculated through the blower and the cycle repeated. All the necessary data were monitored.

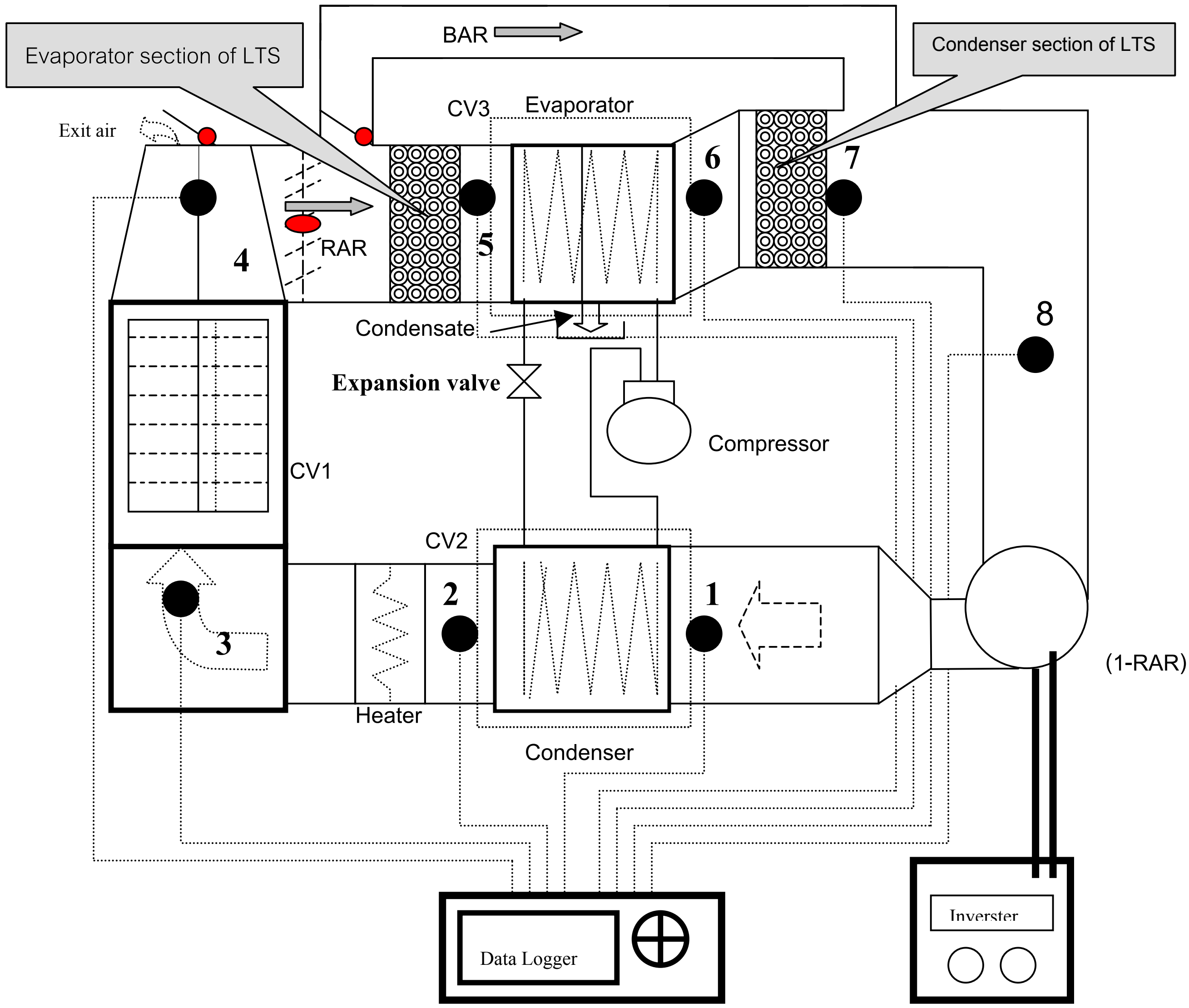

Figure 5.2 Experimental set up of loop thermosyphon integrated with heat pump dryer.

5.4 Verification of the mathematical model of heat pumps dryer with loop thermosyphon.

The experimental results were compared with the simulated results of whole longan, which used heat pump dryer with loop thermosyphon, as shown in Figures 5.3, 5.4 and 5.5. The drying conditions are drying temperature of 75 °C, airflow rate 937 m^3 per hour, by-pass air ratio of 0.15, recirculation air ratio of 40%, initial and final moisture content of 300 and 25 % dry basis, respectively. The moisture content of whole longan, the exits air temperature of the condenser and the evaporator and the specific energy consumption are used for verification of the mathematical model.

Figure 5.3 shows the evolution of moisture content obtained from the experimental and simulated results. The simulated moisture content is nearly the same as that of the experiments.

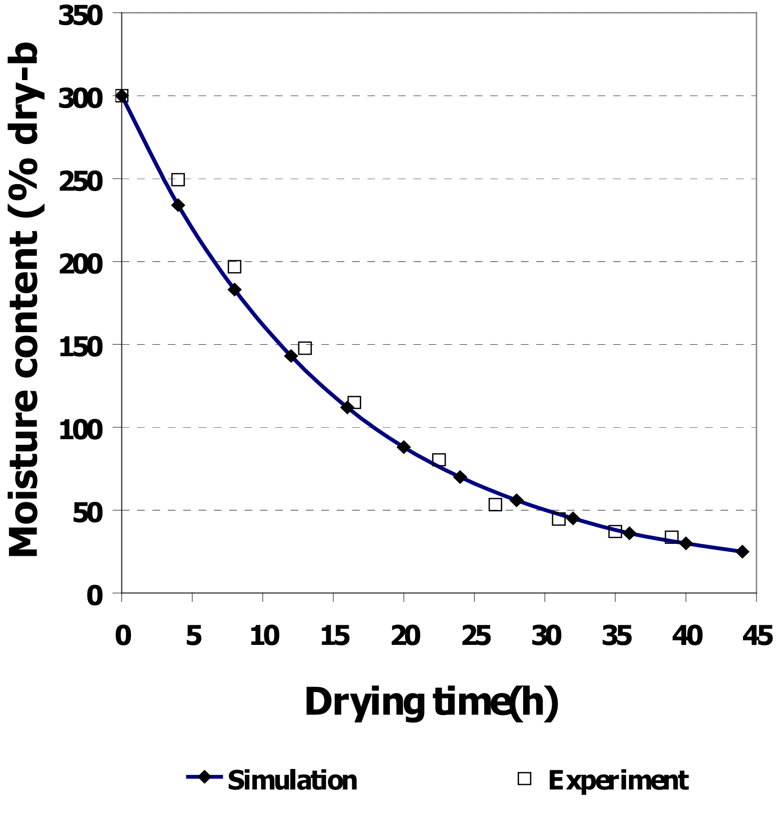

Figure 5.3 Drying kinetics of whole longan using heat pump dryer with LTS. (These results were simulated under the condition: Tdi = 75 °C, Md = 937 m^3/hr, RAR = 40%, BAR = 15%, the initial and final moisture content of the product were 300 and 25 % dry-basis).

Figure 5.4 shows the exit air temperature of the condenser and the evaporator. The value of the exit air temperature of the condenser from the experiment was higher than the result from simulation. This is attributed to the ambient temperature of air used in the experiment which is higher than the air temperature of the simulate program. The simulation program set the ambient temperature, as 30 °C. The exit air temperature of the evaporator is nearly the same for all the drying times.

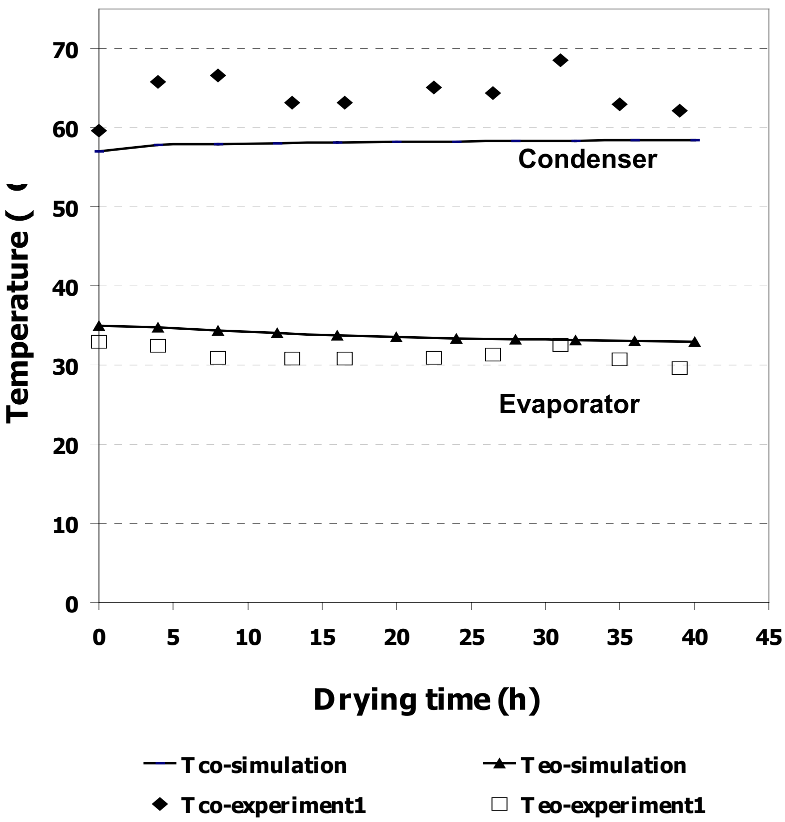

Figure 5.4 Comparison of the exit air temperature of the condenser and the evaporator of HPD with LTS. (These results were simulated under the condition as: Tdi = 75 °C, Md = 937 m^3/h, RAR = 40%, BAR = 15%, the initial and final moisture content of the product were 300 and 25% dry-basis, Tco = Temperature of exit air of the condenser, Teo = temperature of exit air of the evaporator.).

Figure 5.5 shows the specific energy consumption of the heat pump dryer with loop thermosyphon. The operating conditions are the same, as those of the experiments conducted with heap pump dryer without loop thermosyphon in Chapter III. It can be seen that in each condition, the value from simulation and experiment correspond well when the recircuration air ratio increase from 30 to 35%. But from 35 to 40 % the result from experiment was different from the result of mathematical model especially for airflow rate of 1350 m^3/hr. The details are as follow:

For airflow rate 937 m^3/hr. When the recircuation air ratio increased the specific energy consumption decrease in simulated and experimental data. This is because of the recirculation air ratio increase, it means that the recovery the waste heat increases. So, the specific energy consumption decreases. The result from model has a few differences. When the recirculation air ratio in crease from 35 to 40%, the specific energy consumption a few increase. This is because at the recirculation air ratio 35% is the minimum specific energy consumption of heat pump dryer with loop thermosyphon system.

For airflow rate 1320 m^3/hr. When the recircuation air ratio increase the specific energy consumption decrease for experimental result and simulation result. This is because of the recirculation air ratio increase, it means that the recovery the waste heat increases. So, the specific energy consumption decreases. The result from model has a few differences. When the recirculation air ratio in crease from 35 to 40%, the specific energy consumption a few increase. This is because at the recirculation air ratio 35% is the minimum specific energy consumption of heat pump dryer with loop thermosyphon system.

For the airflow rate 1620 m^3/hr. When the recircuation air ratio increase the specific energy consumption decrease in simulated and experimental data. This is because of the recirculation air ratio increase, It means that the recovery the waste heat increases. So, the specific energy consumption decreases. The result from model has a few differences. When the recirculation air ratio in crease from 35 to 40%, the specific energy consumption a few increase. This is because at the recirculation air ratio 35% is the minimum specific energy consumption of heat pump dryer with loop thermosyphon system.

However, it can be concluded that the suitable condition, which the heat pump dryer with loop thermosyphon has agree with the result of Jolly *et al* (1990), Jai *et al* (1990) and prasertsan *et al* (1996).

From this study, it can be concluded that the suitable condition for operating heat pump dryer with loop thermosyphon are at the low airflow rate, but this value must not be lower the lower limit of the value required for heat pump operation. For this research, the recirculation air ratio of 35%, bypass air ratio of 15% and the specific energy consumption was minimum when the airflow rat of 937 m^3/h.

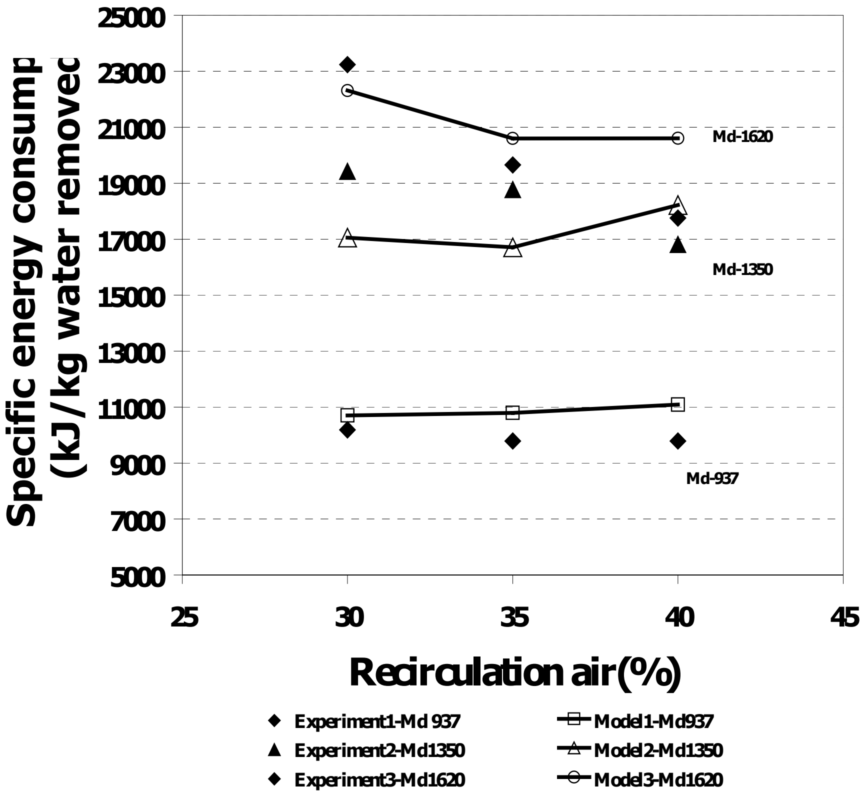

Figure 5.5 Comparison of the SECT between simulation model and experimental data of HPD with LTS.(These results were simulated under the on condition as: Tdi = 75 °C, Md = 937,1350 and 1620 m^3/hr, RAR = 40%, BAR = 15%, the initial and final moisture content were 300 and 25% dry-basis)