Available online at www.sciencedirect.com

SciVerse ScienceDirect

Procedia Engineering 56 (2013) 510 – 520

Procedia Engineering

www.elsevier.com/locate/procedia

5th BSME International Conference on Thermal Engineering

Mathematical model for a heat pump dryer for aromatic plant

Mohammed Ayub Hossain[a,*], Klaus Gottschalk[b], Mohammad Shoeb Hassan[c,*]

[a,c]*FMPE Division, Bangladesh Agricultural Research Institute, Gazipur-1701, Bangladesh*
[b]*Leibniz-Institut für Agrartechnik, Potsdam-Bornim, Max-Eyth-Allee 100, 14469 Potsdam, Germany*

Abstract

A mathematical model was developed to evaluate the performance of heat pump dryer for drying of aromatic plants. The model consists of three sub-models; namely, drying model, heat pump model, and performance model. Drying model was developed based on mass balance, heat balance, heat transfer and drying rate equations. Heat pump sub-model consists of some theoretical and empirical equations for estimating the parameters of evaporator, compressor, condenser and expansion valve. The performance sub-model was the equations for prediction of drying efficiency, COP (coefficient of performance), MER (moisture evaporating rate) and SMER (specific moisture evaporating rate). The model was validated with the experimental data. The experiments was conducted in a fixed bed drying of valerian roots (*Valeriana officinalis* L.) in cooperation with a agricultural company (Agrargenossenschaft Nöbdenitz e.G., Thüringen) in Thüringen, Germany. Data logger was used to record the temperature, relative humidity, humidity ratio and enthalpy of air at different positions of the dryer equipped with different types of sensors. The average drying air temperature was 36.84°C and relative humidity was about 20%. About 89 hours were required to reduce the moisture content of valerian roots from 89 to 9% (wb). The simulated results (temperature, relative humidity and moisture content) agreed well with the experimental results. The average COP, MER and SMER and drying efficiency were 5.45, 140.03 kg/h, 0.038 kg/kWh, and 78.23%, respectively. This model may be used for design data for heat pump dryer for drying of aromatic plants as well as other heat sensitive crops.

Selection and peer review under responsibility of the Bangladesh Society of Mechanical Engineers

Keywords: Aromatic plant; heat pump dryer; heat and mass transfer; simulation; valerian root.

Nomenclature

C_p	specific heat (kJ/kgK)
C_{pv}	specific heat of water vapour (kJ/ kgK)
C_{rp}	specific heat of refrigerant at constant pressure (kJ/ kgK)
C_{rv}	specific heat of refrigerant at constant volume (kJ/ kgK)
D	diameter of tube (m)
f_r	friction factor of refrigerator
G_a	mass flow rate of air (kg/s)
H	enthalpy of refrigerant (kJ/kg)
H_a	enthalpy of air (kJ/kg)
h_c	convective heat transfer coefficient (W/m^{2}K)
h_{fg}	latent heat of vaporization of water vapour (W/m^{2}K)
h_r	rdiative heat transfer coefficient (W/m^{2}K)
k	thermal conductivity (kW/mK)
k_0	coefficient of drying rate (h^{-1})

* Corresponding author. Tel.: +88 2 925 2407 fax: +88 2 925 1415.
E-mail address: mahossain.fmpe@gmail.com

1877-7058 © 2013 The Authors. Published by Elsevier Ltd. Open access under CC BY-NC-ND license.
Selection and peer review under responsibility of the Bangladesh Society of Mechanical Engineers
doi:10.1016/j.proeng.2013.03.154

L	length (m)
L_p	latent heat of vaporization (kJ/kg)
M	moisture content of product (kg/kg, d.b.)
M_e	equilibrium moisture content (kg/kg, d.b.)
m_r	mass flow rate of refrigerant (kg/s)
n	polytrophic index
P	pressure (Pa)
Q	heat transfer rate (kW)
r	exponent of Page equation
t	temperature (°C/K)
u	air velocity (m/s)
W	power (kW)
w	humidity (kg/kg)
z	bed thickness (m)
Greek symbols	
α	heat transfer coefficient (kW/m^2K)
∇	change
η_d	drying efficiency of dryer
θ	drying time (s)
ρ	density (kg/m^3)
Subscripts	
a	air
c	condenser
com	compressor
d	dryer
e	external
i	internal
t	tube
r	refrigerant

1. Introduction

Drying is the removal of moisture from a product to its desired or safe moisture content. Heat is normally supplied to the product by heated air by natural means or artificially and the vapour pressure or moisture concentration gradient thus created causes the movement of moisture from inside of the product to the surface. During drying the temperature of the drying air must be kept below the recommended values depending on the intended use of the product. Excessive high temperature drying causes both physical and chemical changes ultimately deteriorate the quality of the product (Bala, 1997). 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 ultimate 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.

Heat pump dryers have the potential to operate more efficiently, and at lower temperatures than conventional dryers. Rossi et al. (1992) made a comparison of vegetable drying by the heat pump dryer and a conventional dryer using an electrical heater and found that an energy saving of 40% and the processing time reduced by 40.7%**.** Mellmann and Fürll (2008) reported that energy costs accounted for 30 to 55% of production costs. Ziegler et al. (2009) conducted experimental study for drying of medicinal plants and spices using batch dryer coupled with heat pump. They conducted different modes of operation such as first day, heat pump drying with closed cycle system then subsequently conventional drying. They reported that in early stage of drying, primary energy consumption can be reduced by 40-55% using heat pump with internal heat recovery. Studies of heat pump dryers by computer simulation have received attention in recent years. Many theories and designs have been put forward to describe heat and mass transfer taking place in the heat pump dryer system. A simplified mathematical model of a heat pump assisted crop dryer need to be developed. The developed heat pump dryer model will be used to investigate the performance and verify its accuracy with respect to the experimental results. Prasertsan et al. (1996) developed mathematical models for heat pump dryer characteristics of four configurations, two open and two closed systems. They reported that ambient conditions and dryer efficiency played an important role in the system behaviour. Closed cycle was found more efficient than fully open cycle configuration. Phani et al. (2002a) developed simulation model for heat pump assisted drying of crops. Therefore, this study was undertaken to develop a simulation model for heat pump drying of selected aromatic plant.

2. Mathematical models

The mathematical model of a heat pump dryer consists of three sub-models; namely, drying model, heat pump model, and performance model. Heat and mass balance of both refrigerant and air circuits are used for development of mathematical models. The schematic view of a heat pump dryer is shown in Fig. 1.

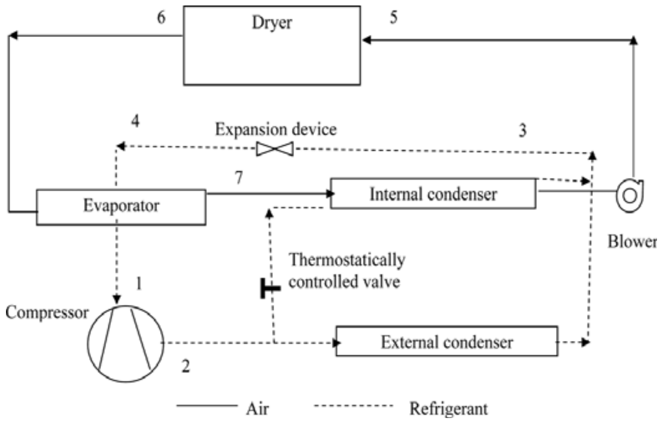

Fig. 1. Schematic view of a heat pump dryer.

2.1 Drying model

The physically-based drying model generally consists of several partial differential equations- mass balance equation, heat balance equation, heat transfer equation and drying rate equation, (Bala, 1997). This model is developed by a set of partial differential equations to describe the heat and mass transfer between drying air and product and also within a single layer of product during a small time increment. The following assumptions are made for simplification of the model.

- The air duct and drying chamber are thermally insulated
- Ambient conditions and specific heat capacity of air are constant
- The inlet air properties of a layer are the same for the outlet air of the previous layer
- Pressure of air in the system and mass flow rate of circulating dry air are constants
- Thermal equilibrium exists between the drying air and product
- The condition of air entering the evaporator is the same as that leaving the dryer and air entering the dryer is same as that leaving the condenser.

Consider an elemental layer dz of the bed (z, z + dz) of unit cross-sectional area with airflow G_a from z to z + dz (Fig. 2). There are four unknowns: changes of product moisture content, humidity, product temperature and air temperature (Hossain et al., 2010). Therefore, four differential equations are required to solve these unknowns.

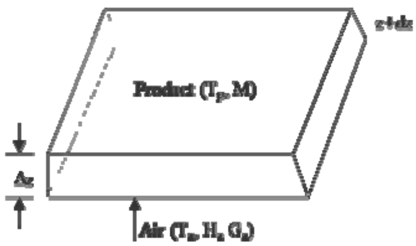

Fig. 2. Element of product (valerian roots) bed.

2.1.1 Mass balance equation

The flow of moisture into the element (z) at time dθ is $G_a w_a(z)d\theta$ and out of the element (z+dz) is $G_a w_a(z+dz)d\theta$. The moisture lost from the product in the element of bed dz at time θ is $-\rho_p M(\theta)dz$ and at time (θ + dθ) is $-\rho_p M(t + dt)dz$. The mass balance equation is therefore given by:

Moisture lost by product = moisture gained by air.

$$-\rho_p M(\theta+d\theta)dz - (-\rho_p M(\theta)dz) = G_a w_a(z+dz)d\theta - G_a w_a(z)d\theta \tag{1}$$

Using Taylor series expansion for w_a and M and ignoring all terms of dz^2 and $d\theta^2$ and higher, Equation (1) reduces to

$$G_a\left(\frac{\partial w_a}{\partial z}\right)dzd\theta = -\rho_p\left(\frac{\partial M}{\partial t}\right)dzd\theta \tag{2}$$

Therefore, the change of humidity in the finite difference form, the Equation (2) becomes as

$$\Delta w_a = -\left(\frac{\rho_p}{G_a}\right)\left(\frac{\Delta M}{\Delta\theta}\right)\Delta z \tag{3}$$

2.1.2 Heat balance equation

Heat balance equation of the drying element (layer) is the change in enthalpy of air equal to change in enthalpy of product. Therefore, change of enthalpy of air of element (z, z+dz) at time dθ is:

$$G_a[C_{pa} + C_{pw} w_a(z+dz) t_a(z+dz) + L_a w_a(z+dz)]d\theta - G_a[C_{pa} + C_{pw} w_a(z) t_a(z) + L_a w_a(z)]d\theta \tag{4}$$

Again, change of enthalpy of product at time (θ+dθ) and θ is:

$$-\rho_p[C_{pp} + C_{pl} M(\theta+d\theta) t_p(\theta+d\theta]dz + \rho_p[C_{pp} + C_{pl} M(\theta) t_p(\theta)]\,dz \tag{5}$$

Applying the Taylor series expansion and neglecting second order and higher terms, expressions (4) and (5) are equated to and change of air temperature in the finite difference form as:

$$\Delta t_a = \frac{-\dfrac{\rho_p}{G_a}\Delta t_p[C_{pp} + C_{pl}(M+\Delta M)](\dfrac{\Delta z}{\Delta\theta}) + \dfrac{\rho_p}{G_a}\Delta M(C_{pw} t_a + L_a - C_{pl} t_p)(\dfrac{\Delta z}{\Delta\theta})}{C_{pa} + C_{pw} w_a - \dfrac{C_{pw}\rho_p}{G_a}\Delta z(\dfrac{\Delta M}{\Delta\theta})} \tag{6}$$

2.1.3 Heat transfer rate equation

Heat transfer between air and product = change in sensible heat of product + change in enthalpy in air just after drying and just prior to drying (evaporation). Therefore, the expression can be written mathematically as:

$$h_{cv}(t_a - t_p)\,dzd\theta = \rho_p[C_{pp} t_p(\theta) + C_{pl} M t_p(\theta) + (C_{pw} + L_p) M(\theta)\,dzd\theta - C_{pl} M(\theta) t_p)]dz \tag{7}$$

Applying the Taylor series expansion and neglecting the second and higher order terms Equation (7) becomes in the finite difference form and after simplification, the change of product temperature becomes

$$\Delta t_p = \frac{2(t_a - t_p) + \Delta t_a + \dfrac{2\rho_a}{h_{cv}}(C_{pw} + L_p - C_{pl} t_p)\dfrac{\Delta M}{\Delta\theta}}{1 + \dfrac{\rho_p}{h_{cv}\Delta\theta}(C_{pp} + C_{pl} M)} \tag{8}$$

2.1.4 Drying rate equation

The rate of change of moisture content of a thin layer product inside the dryer can be expressed by an appropriate thin layer drying equation. No thin layer drying equation was available in the literature for drying of valerian (*Valeriana officinalis* L.) roots. Asparagus (*Asparagus racemosus* Wild.) root is similar in nature and widely used as medicinal purposes. The Page equation was found suitable to predict moisture content of asparagus root in thin layer (Bala et al., 2010). The Page equation is as follows.

$$\frac{M-M_e}{M_o-M_e}=\exp(-k_p t_p^r) \tag{9}$$

The Page equation in differential form is

$$\frac{dM}{dt}=-k_p r\left(\frac{M-M_e}{M_0-M_e}\right)\theta^{r-1} \tag{10}$$

Taking the mean value of M and (M+ΔM) the Equation (10) in the finite difference form can be expressed as (Hossain et al., 2005).

$$\Delta M=-k_p n.\Delta\theta(M-M_e)(X+\Delta\theta)^{r-1} \tag{11}$$

Where, $X=\left[\frac{-\ln\left(\frac{M-M_e}{M_0-M_e}\right)}{k_p}\right]^{\frac{1}{r}}$

The values of k and r may the obtained in terms of drying air temperature from the work of (Bala et al., 2010).

2.2 Heat pump model

The heat pump is a complex device with continuous changing air and refrigerant properties during operation. To make the model simpler, the following assumptions are made.

- The refrigerant at outlet of evaporator and condenser are saturated vapour and saturated liquid, respectively
- The compression and the expansion of the refrigerant are isentropic and isenthalpic processes, respectively
- The tubes connecting the heat pump components are insulated and the pressure drop in the pipe system is negligible
- The wall of the component housing is adiabatic
- Heat pump is operated at steady-state.

2.2.1 Evaporator model

The rate of moisture condensed or removal at the evaporator surface can be computed using the following equation.

For ideal condition:

$$\mathrm{m_{ew}}=\mathrm{m_a}(\mathrm{w_{do}}-\mathrm{w_{es}}) \tag{12}$$

In an evaporating coil, all the airs are not cooled due to not coming in contact with the cooling coil. The value of bf (bypass factor) for an efficient process at the evaporator should be in the range of 0.10-0.15 (Phani et al., 2002b).

$$\mathrm{m_{ew}}=\mathrm{m_a}(1-bf)(\mathrm{w_{do}}-\mathrm{w_{es}}) \tag{13}$$

Now, the energy transfer between the refrigerant air at the evaporator is given by the following equation.

$$\mathrm{Q_e}=\mathrm{m_r}(\mathrm{H_1}-\mathrm{H_4})=\mathrm{m_a}\mathrm{C_{pa}}(\mathrm{t_{ei}}-\mathrm{t_{eo}}) \tag{14}$$

Where, H_1=saturated vapour enthalpy (H_v) and H_3=liquid enthalpy (H_l). Therefore H_1 and H_3 can be obtained from the following empirical equations (Singh and Heldman, 2009).

$$H_1 = H_v = a_8 + a_9 T_{re} + a_{10} T_{re}^2 + a_{11} T_{re}^3 + a_{12} \tag{15}$$

$$H_3 = H_l = a_4 + a_5 T_{rc} + a_6 T_{rc}^2 + a_7 T_{rc}^3 \tag{16}$$

The forced convection heat transfer coefficient on the air side of a flat-finned tube coil is calculated by the correlation proposed by Rich (1973).

2.2.2 Compressor model

A mathematical model of a reciprocating compressor is used to predict the change in enthalpy of refrigerant during compression and energy consumption. The main parameters of compressor are volumetric efficiency (η_v), refrigerant mass flow rate (m_r) and compression power (W_{com}). These can be calculated using the following equations.

$$\eta_v = 1 + n - n\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} \tag{17}$$

$$W_{com} = m_r (H_2 - H_1) \tag{18}$$

$$H_2 = H_1 + \nabla H \tag{19}$$

∇H can be obtained from the following equation (Singh and Heldman, 2009).

$$\nabla H = \frac{P_1 v_{rl}}{3600}\left(\frac{n}{n-1}\right)\left\{\left(\frac{P_2}{P_1}\right)^{(n-\frac{1}{n})} - 1\right\} \tag{20}$$

Here, n is the ratio of specific heat of refrigerant at constant pressure (C_{rp}) and specific heat at constant volume (C_{rv}) i.e. $n = \frac{C_{rp}}{C_{rv}}$.

2.2.3 Condenser model

The condenser model is used to predict the energy transfer and mass flow rate of refrigerant in the condenser coil. Generally, two condensers are used as heat exchanger-internal and external (Fig. 1). The rejected heat by the external condenser can be computed using following equation.

$$Q_{ce} = m_r (H_2 - H_3) \tag{21}$$

The heating load of internal condenser may be calculated using the following equation. The condenser model is used to calculate the energy transfer and mass flow rate of refrigerant in the condenser coil.

$$Q_{ci} = m_a (C_{pa} + w_{di} C_{pv})(t_{co} - t_{eo}) \tag{22}$$

Heating load in the external condenser (Q_{ce}) may be computed from the following equation

$$Q_c = Q_{ci} + Q_{ce} \tag{23}$$

Heat transfer coefficients of the refrigerant side of the condenser for single phase are calculated as (ASHRAE, 1997).

2.2.4 Expansion valve model

In this model, capillary tube is used as the expansion device for pressure reduction. The expansion process is assumed to be isenthalpic, i.e.

$$H_4 = H_3 \tag{24}$$

The frictional pressure drop and friction factor of each part is calculated and the required length of capillary tube is determined using the following equations (ASHRAE, 1997).

$$\nabla P_r = \frac{\rho_r f_r \nabla L u_r^2}{2D} \tag{25}$$

2.3 Performance model

The drying and dehumidification performance of the heat pump dryer are evaluated by the drying efficiency of the dryer (η_d), coefficient of performance of the heat pump (COP), moisture extraction rate (MER) and specific moisture extraction rate (SMER). These may be computed using the following equations (Pal and Khan, 2008).

$$\eta_d = \frac{t_{di} - t_{do}}{t_{di} - t_w} \tag{26}$$

$$COP = \frac{Q_{con}}{W_{com}} = \frac{H_2 - H_3}{H_2 - H_1} \tag{27}$$

$$MER = \frac{m_w}{\theta} \tag{28}$$

$$SMER = \frac{m_w}{W_{Com} + E_b} \tag{29}$$

2.4 Solution procedure

The drying chamber was vertically divided into a number of sections (trays) (L=j.Δx) along the direction of air flow in the dryer. The drying time was also splitted into a number of intervals (θ = i.Δθ). Using k and r values, the changes of moisture content of valerian roots for time interval, ΔM was calculated from equation (11). The change in air humidity was computed using equation (3). The drying air temperature and product temperature was computed using equation (6) and equation (8) respectively. This process would be repeated section by section until the end of the section was reached. This process would be then repeated for each time increment.

The saturated vapour enthalpy and liquid enthalpy was obtained from equations (15) and (16), respectively. The energy transfer between refrigerant and air was determined using equation (14). The value of H_2 was determined using equation (19). Q_{ce} and Q_{ci} was computed from equations (21) and (22), respectively. Then Q_c was determined using equation (23). Then the pressure drop was computed from equation (25). Now, the dryer efficiency (η_d), COP, MER and SMER were calculated using equations (26), (27), (28), and (29), respectively. The whole solution process was solved using the Qbasic computer program.

2.5 Experimental procedure

The experiments was conducted to fixed bed drying of valerian roots (*Valeriana officinalis* L.) in cooperation with a large agricultural company (Agrargenossenschaft Nöbdenitz e.G., Thüringen) in Thüringen, Germany. Before starting an experimental run, the whole apparatus was operated for several hours. The fresh and washed valerian roots were loaded in the drying chamber at a thickness of about 250 mm. Loading and unloading was done by a crane. Data logger (Model: ALMENO, Ahlborn Mess-und Regelungstechnik GmbH, Germany) was used to record the temperature, relative humidity, humidity ratio and enthalpy of air at different positions of the dryer using equipped with different types of sensors. The data was recorded at one minute interval. Different types of sensors were set at inlet and outlet points of drying chamber, evaporator, condenser and auxiliary heater. The air conditions of ambient air were also recorded simultaneously. The initial and final moisture content of valerian roots were determined by gravimetric method.

3. Results and discussion

Simulated and measured air temperature in dryer during drying of valerian roots is shown Fig. 3. The total drying time was about 89 hours. The average, maximum and minimum ambient temperature was 4.05, 12.19 and -4.67°C, respectively. Average dryer inlet temperature was 34.80°C. The maximum drying inlet temperature (39.92°C) was below the optimum drying air temperature (40°C) for valerian roots. So, there was no possibility for quality deterioration of valerian roots during drying. Sometimes, the measured temperature fluctuations were observed. These might be due to that at that time the door of the drying chamber was remained open and temperature felled down. The average simulated temperature (34.92°C) was very close to the average measured temperature (34.80°C). There was no significant difference between measured and simulated temperatures ($p \leq 0.00023$) by t-test. So, the simulated temperature in the dryer agreed well with the measured temperature.

Simulated and measured outlet temperatures of evaporator and condenser during drying of valerian roots are given in Fig. 4. The average measured (40.89°C) and simulated (40.10°C) condenser temperature was found very close to each other and there was no significant difference between them by t-test ($p \leq 0.000364$). But the measured condenser temperature was found to fluctuate with the drying time. This may be due to that the door of condenser opened several times for adjusting. Similar result was observed for evaporator ($p \leq 0.10065$) outlet measured and simulated temperatures.

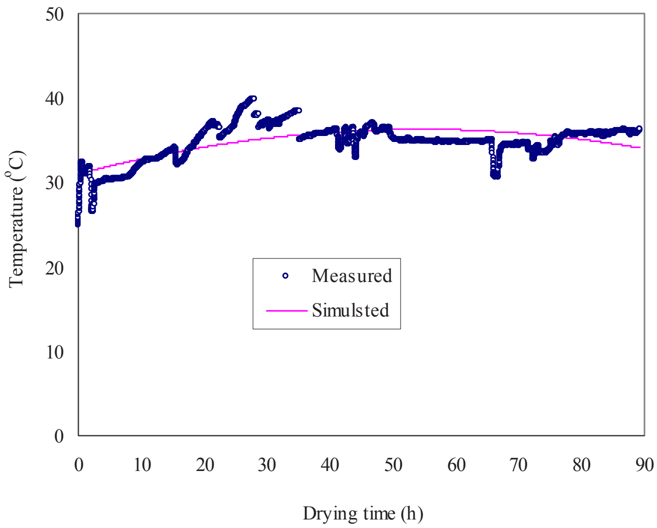

Fig. 3. Simulated and measured air temperature in dryer during drying of valerian roots.

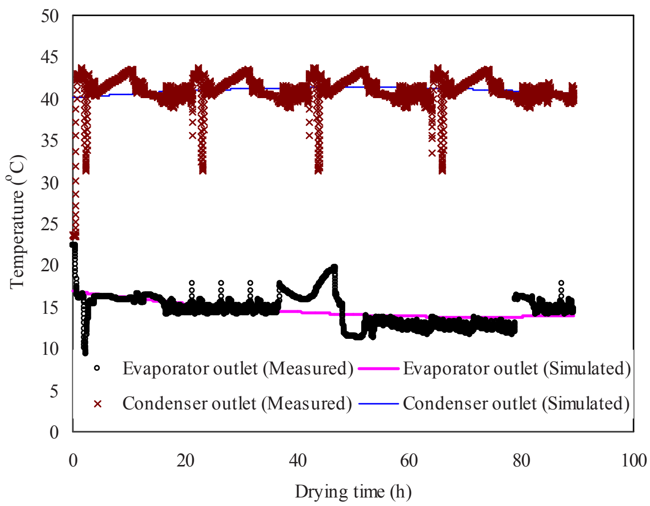

Fig. 4. Simulated and measured outlet temperature of evaporator and condenser during drying of valerian roots.

Simulated and measured inlet and outlet relative humidity in the dryer during drying of valerian roots are shown in Fig. 5. The difference of dryer inlet and outlet measured and simulated relative humidity (p≤0.0256) was found statistically insignificant. The outlet measured relative humidity (p≤0.0219) was found to fluctuate during drying period. This may be due to that inversion of valerian roots were carried out several times for better and uniform drying. But reasonable good agreement was found for measured and simulated relative humidity for both inlet and outlet air.

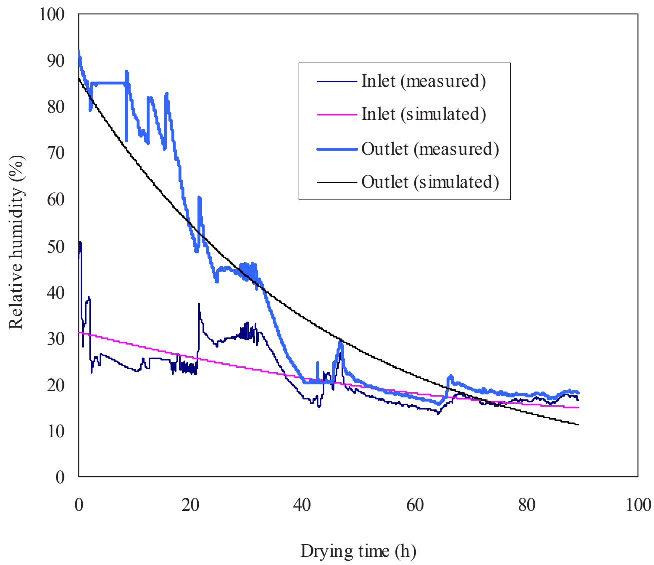

Fig. 5. Simulated and measured inlet and outlet relative humidity in the dryer during drying of valerian roots.

Simulated enthalpies of air in the compressor, condenser, evaporator, and dryer during drying of valerian roots are shown in Fig. 6. Average enthalpy of air in the compressor was found the highest (63.07 kJ/kg) followed by evaporator (46.91 kJ/kg) and dryer (28.26 kJ/kg). The average lowest enthalpy (23.55 kJ/kg) was found in the condenser. All the simulated enthalpy lines followed the similar pattern during drying period.

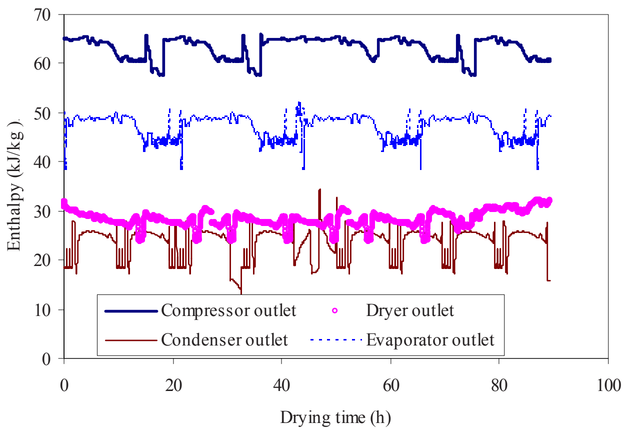

Fig. 6. Simulated enthalpy of air in the compressor, condenser, evaporator, and dryer during drying of valerian roots.

Variation of measured and simulated moisture content of valerian roots with drying time is shown in Fig. 7. Drying curves followed the exponential trend. There were some fluctuations in moisture content due to the fluctuations of drying air temperatures. The initial moisture content of valerian roots was 6.00 kg/kg dry basis (86%, wet basis) and dried to 0.10 kg/kg dry basis (9%, wet basis). There was no significant difference between measured and simulated moisture contents by t-test (p≤0.01222). Therefore, the simulation agreed well with the measured moisture content of valerian roots during drying.

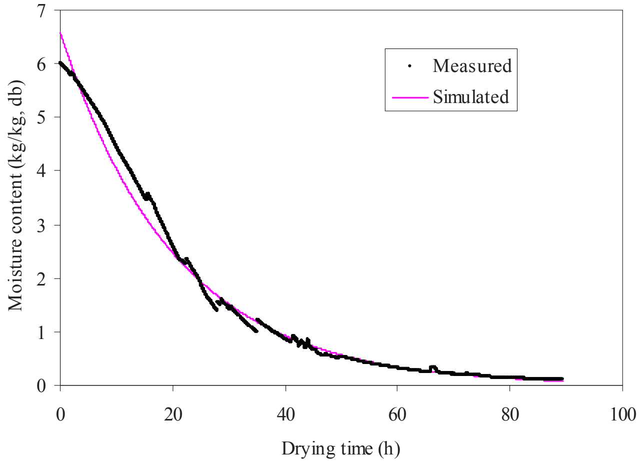

Fig. 7. Measured and simulated moisture content of valerian roots during drying.

The average MER (moisture evaporating rate) and SMER (specific moisture evaporating rate), COP (coefficient of performance), and drying efficiency are given in Table 1. The average MER and SMER were found to be 140.03 kg/h and 0.038 kg/kWh, respectively. The average COP was 5.45. It is noted that the COP of industrial dryer should not below 5. The drying efficiency was found to be 78.23%.

Table 1. Performance parameters of heat pump dryer during drying of valerian roots

Performance parameters	Average value
MER (kg/h)	140.03
SMER (kg/kWh)	0.038
COP	5.45
Drying efficiency (%)	78.23

4. Conclusion

The average drying air temperature was 36.84°C and relative humidity was about 20%. About 89 hours were required to reduce the moisture content of valerian roots from 89 to 9% (wb). The simulated results (temperature, relative humidity and moisture content) agreed well with the experimental results. The average COP, MER, SMER and drying efficiency were 5.45, 140.03 kg/h, 0.038 kg/kWh, and 78.23%, respectively. This model may be used for design data for heat pump dryer for drying of aromatic plants as well as other heat sensitive crops.

Acknowledgement

This research was carried out with the financial assistance of Alexander von Humboldt Foundation, Germany and physical support provided by Leibniz Institut für Agrartechnik Potsdam-Bornim, Germany. The authors are very thankful to them.

References

[1] Bala B K. Drying and Storage of Cereal Grains. New Delhi: Oxford and IBH Publishing Co. Pvt. Ltd.;1997.
[2] Rossi S J, Neves L C, Kieckbusch T G. Thermodynamic and energetic evaluation of heat pump applied to the drying of vegetable. Proceedings 8th International Drying Symposium (IDS92), Montreal, Canada, August 2-5, 1992; pp. 1475-1484.
[3] Mellmann J, Fürll C. Trocknungsanlagen für Arznei- und Gewürzpflanzen – spezifischer Energieverbrauch und Optimierungspotenzial (Drying facilities for medicinal and aromatic plants–specific energy consumption and potential for optimisation). Zeitschrift für Arznei- und Gewürzpflanzen 2008;13(3):127-133.
[4] Ziegler Th, Niebling F, Teodorov T, Mellmann J. Wärmepumpentrocknung von Arznei- und Gewürzpflanzen – Möglichkeiten zur Steigerung der Energieeffizienz/ Heat pump drying of medicinal and spice plants – possibilities of energy efficiency enhancement. Zeitschrift für Arznei- und Gewürzpflanzen 2009; 14(4):160-166.

[5] Prasertsan;, S, Saen-Saby P, Ngamsritrakul, P, Prateepchaikul G. Heat pump dryer Part 1: Simulation of the models. International Journal of Energy Research 1996; 20: 1067-1079.

[6] Phani K A, Greg J S, Sokhansanj S. Performance study of a heat pump dryer system for specialty crops- Part 1: Development of a simulation model. International Journal of Energy Research 2002a; 26: 1001-1019.

[7] Hossain M A, Gottschaslk K, Amer, B M A. Mathematical modeling for drying of tomato in hybrid dryer. The Arabian Journal for Science and Engineering 2010; 35(2B): 239-262.

[8] Bala B K, Hoque M A, Hossain M A, Uddin M B. Drying characteristics of asparagus roots (*Asparagus racemosus* Wild.). Drying Technology 2010; 28: 533-541.

[9] Hossain M A, Woods J L, Bala B K. Simulation of solar drying of chilli in solar tunnel drier. International Journal of Sustainable Energy 2005; 24(3): 143-153.

[10] Phani K A, Sokhansanj S, Schoenau G J S. Performance study of a re-circulating cabinet dryer using a household dehumidifier. Drying Technology 2002b; 20(8): 1673-1689.

[11] Singh R P, Heldman D R. Introduction to Food Engineering. Fourth Edition. Academic Press, An imprint of Elsevier; 2009.

[12] Rich D G. The effect of fin spacing on the heat transfer and friction performance of multi-row, smooth plate fin-and-tube heat exchangers. ASHRAE Transactions 1973; 79(1): 137-145.

[13] ASHRAE. ASHRAE Handbook: Fundamentals. American Society of Heating, Refrigerating and Air- Conditioning Engineers. Inc. 1791 Tullie Circle, N. E., Atlanta, GA 30329; 1997.

[14] Pal U S, Khan M K. Calculation steps for the design of different components of heat pump dryers under constant drying rate condition. Drying Technology 2008; 26(7): 864-872.