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Investigating energy consumption and quality of rough rice drying process using a grain heat pump dryer

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Abstract

In this study, energy consumption and quality of rough rice drying process were investigated using a laboratory designed and fabricated grain heat pump dryer. Designing of the heat pump dryer was carried out based on a newly developed non-equilibrium model for the numerical simulation of rough rice drying in a deep-bed mode. For this purpose, the partial differential equations of rough rice drying were solved in MATLAB software. After solving the governing drying equations and heat pump simulation in HYSYS software, a 184 W compressor, an evaporator, an expansion valve and two condensers, were used to fabricate the heat pump system. Rough rice was selected as the material for evaluation of the setup. The rough rice fissuring difference before and after drying decreased about 40% in heat pump dryer mode in comparison with conventional dryer (6 vs. 10%). Moreover, despite of consuming power by compressor in heat pump dryer mode, the total power consumption reduced about 10% compared to the hot air dryer alone. Therefore, it is concluded that the fully closed loop heat pump dryer system could be applied for rough rice drying.

Keywords: Coefficient of Performance, Energy, Heat Pump Dryer, Mathematical Modeling, Specific Moisture Extraction Rate.

Abbreviations:

$\dot{Q}_{cd}$	rate of heat delivered in the condenser (kW)
$\dot{m}_{air}$	mass flow rate of air (kg s^{-1})
$\dot{W}_f$	input power to the centrifugal fan (kW)
$\dot{W}_c$	input power to the compressor (kW)
$\dot{W}_h$	input power to the electrical heater (kW)
$\dot{V}_{air}$	volumetric flow rate of air (m^3 s^{-1})
$COP_{hp,h}$	heating coefficient of performance of the heat pump (-)
$C_{p,air}$	specific heat of air (kJ kg^{-1} °C^{-1})
$\dot{m}_d$	drying rate (g h^{-1})
$SMER_{hp}$	specific moisture extraction rate for heat pump (kg kW^{-1} h^{-1})
$SMER_{ws}$	specific moisture extraction rate for whole system (kg KW^{-1} h^{-1})
$T_{co,air}$	average air temperature leaving the condenser (°C)
$T_{ci,air}$	average air temperature entering to the condenser (°C)
HPD	heat pump dryer
EHD	electrical heated dryer

Introduction

Drying is the most common method for preserving food and agricultural crops used in practice (Midilli et al., 2002). The optimization of drying operation results in improving the quality of the output product, reducing the cost of processing and energy consumption, as well as optimizing of the throughput (Madamba et al., 1994). The conventional drying methods of agricultural products require great energy (Karathanos and Belessiotis, 1999). Some thermal and physical properties of agricultural products such as moisture diffusion, heat and mass transfer, activation energy and specific energy consumption are also important for the proper dryer design (Aghbashlo et al., 2008). Heat pumps have been extensively used by industry for many years, but less for drying process. The modeling of heat pump dryers (HPDs) for design purpose has attracted research interest for more than 20 years. Yet the literature remains divided on what design features are appropriate for different application of HPDs (Oktay and Hepbasli, 2003). About 6.2×10^9 J energy is used to produce one ton food crop with current technology. More than 24% of this amount is needed for the drying process (Wang and Chang, 2001). An improved dryer with energy recovery can reduce the total energy requirement. A heat pump is attractive, because it can deliver more energy as heat than it consumes in electrical energy. HPDs offer several advantages over traditional oil-fired, gas-fired or conventional electrical heated dryers (EHDs) for drying of food products and agricultural materials such as grains. These advantages include higher energy efficiency, better quality, as well as the ability to operate independently of outside ambient weather conditions. Several researchers have studied heat pump grain drying. In a research the fuel efficiency of a grain dryer through heat recovery was improved (Lai and Foster, 1977). Two heat recovery methods were used to reclaim the energy from the exhaust air of a laboratory scale grain dryer (0.46 m diameter and 1.22 m high). The results showed that the average heat energy required to remove 1 kg of water from grain was 2954 kJ in a closed loop mode with a heat pump and heat exchanger in operation. In an open loop mode, the energy requirements were 2768, 4792 and 5350 kJ, respectively for heat pump plus heat exchanger, heat exchanger only, and control set-up (no heat recovery). An experimental closed loop heat pump grain drying system was designed and evaluated for energy performance while drying shelled corn and grain sorghum (Wang and Chang, 2001). Four drying tests using three corn lots with initial moisture contents of 25, 20, and 18% (w.b.) and one grain sorghum lot with initial moisture content of 25% (w.b.) were conducted. Temperature of the heated air was controlled in the range of 43-45 °C with air flow rates of 0.26 and 0.25 m^3 h^{-1} per dry mass of grain for corn and grain sorghum, respectively. The average amount of input energy (all electric) required to remove 1 kg of water from the corn lots was 1186 kJ based on grain lot weight differences before and after drying. The grain sorghum drying performance (1156 kJ kg^{-1} water) was lower than that for drying corn. The average thermal energy transfer rate of the air-to-air heat exchanger was 11.47 kW, which represented 28% of the total thermal energy transfer rate. The average efficiency of the air-to-air heat exchanger was about 34%. Prasertsan and Paen-Saby, (1998) reported that the HPD performance was not well understood in many studies. For example, some works reported that the optimum performance of the HPD occurs when the coefficient of performance of the heat pump is at maximum. But others revealed that the maximum coefficient of performance, drying rate and system efficiency did not necessarily occur at same working condition. Some researchers suggested that at the early stage of drying, the closed configuration should be used, but the others concluded that the open configuration performed better. It seems that there is no specific configuration for any single product, but the operation must be changed with respect to the product drying rate and ambient condition (Achariaviriya et al., 2000; Prasertsan et al., 1996; Prasertsan et al., 1997). Practically, the HPD should be an open or partially open system in order to stabilize the system by shedding of the energy from system. This would accelerate the drying throughput rate, but at the expense of low energy efficiency. In order to control and optimize the drying process and designing of the proper drying equipments, it is necessary to use the mathematical models describing the drying kinetic. Various mathematical models have been suggested for thin layer drying of rough rice. However, a simulation model that describes rough rice drying in a deep bed mode has not been reported in the literature. Deep bed models for grain drying simulation can be classified as logarithmic, heat and mass balance and partial differential equation (PDE). The PDE model is more detailed, accurate and valid for cereal drying, while the others are less accurate owing to more assumptions being made in the model derivations. These latter models have not dealt with rough rice drying in a deep bed bath dryer. The objective of this research was to investigate energy consumption and quality of rough rice drying using a laboratory grain heat pump dryer which was designed and fabricated based on a newly developed non-equilibrium model for the numerical simulation of rough rice drying in a deep-bed mode.

Results

Characteristics of the fabricated heat pump dryer

Fig. 1 illustrates a schematic of the designed and fabricated setup (Torki Harchegani, 2008). The results of the setup design can be summarized as follows. The dryer was constructed for grain drying only and for experimental studies. To achieve this goal, the system was designed to operate at various conditions. Air circulation was achieved by means of a centrifugal fan with a 2 hp three phase motor and maximum speed of 2800 rpm. The dryer chamber was a plexiglas cylinder with circular cross section of 14 cm internal diameter and 5 mm thickness and was connected to the air paths with flexible PVC tubes and was located on a digital laboratory scale. The electrical heater of the dryer was constructed of 6 elements with total heat capacity of 4.2 kW. A frequency inverter was used to adjust the drying air velocity. A digital multi-meter was used to measure the electrical power consumption of the heater. A portable digital wattmeter was used to measure the electrical power consumption of the centrifugal fan. A compressor with maximum power of 184 W was selected to supply the power for the working fluid of the heat pump which was R134a. The evaporator and condensers were aluminum finned copper tubed heat exchanger with dimensions of 0.26 m high × 0.24 m wide × 2 rows deep. They had two circuits of 10 tubes each. An expansion valve was used to expand the refrigerant. Three pressure gages were used in the compressor inlet and outlet and in evaporator inlet to measure the refrigerant pressure. A control device was considered to prepare the compressor safe operation condition.

The power consumption, SMER and COP

Table 1 shows the results of the power consumed by the electrical heater, the centrifugal fan and the compressor for heat pump dryer (HPD), and by the electrical heater and the centrifugal fan for electrical heated dryer (EHD). The results of specific moisture extraction rate (*SMER*) examinations are also presented in Table 1. In spite of consuming power by compressor in HPD mode ($\dot{W}_c$), it is observed that the total power consumption for this method (1005 W), i.e., the addition of the electrical heater, fan, and compressor powers, is lower than its corresponding value for the EHD system (1099 W), which is the addition of electrical heater and centrifugal fan powers. The *SMER* for the heat pump system ($SMER_{hp}$) calculated by Eq. 4 was obtained 0.1 kg kW^{-1} h^{-1}. The *SMER* for the whole system ($SMER_{ws}$) in EHD mode (0.023 kg kW^{-1} h^{-1}) was more than that in HPD mode (0.019 kg kW^{-1} h^{-1}). The average inlet air temperatures to and outlet air temperature from condenser, heat transfer rate (thermal power) in condenser, and the calculated coefficient of performance (*COP*) for the heat pump system are presented

Table 1. Power consumption of different components, drying rate, and specific moisture extraction rate (*SMER*) of the heat pump dryer and electrical heated dryer.

Dryer mode	$\dot{W}_h$ (W)	$\dot{W}_f$ (W)	$\dot{W}_c$ (W)	$\dot{m}_d$ (g h^{-1})	$SMER_{hp}$ (kg kW^{-1} h^{-1})	$SMER_{ws}$ (kg kW^{-1} h^{-1})
HPD	585	236	184	19.1	0.1	0.019
EHD	925	174	-	25.2	-	0.023

HPD and EHD stand for heat pump dryer and electrical heated dryer, respectively.

Table 2. Coefficient of performance (*COP*) of the heat pump dryer.

$T_{i,air}$ (°C)	$T_{o,air}$ (°C)	$\dot{Q}_{cd}$ (W)	$COP_{hp,h}$
24.01	33.28	138.3	0.76

Table 3. The results of heat pump dryer simulation in HYSYS software.

Evaporator heat exchange capacity (W)	Condenser heat exchange capacity (W)	External condenser heat exchange capacity (W)	Mass flow of refrigerant (kg h^{-1})	Compressor power (W)
331.7	330.8	174.7	9.56	174 *

* The compressor adiabatic efficiency was considered 0.75.

in Table 2. In the experiments, the mean of thermal power delivered in the condenser ($\dot{Q}_{cd}$) was obtained to be 138.3 W (Table 2), whereas the required $\dot{Q}_{cd}$ predicted by HYSYS software was 330.8 W; given in Table 3 which shows the results of heat pump dryer simulation.

Fissured kernels

Table 4 shows the results of the fissured kernels percentage before drying and after drying by EHD and HPD. As presented, the difference of fissured kernels percentage before and after drying in HPD (6.3%) was about 40% lower than its value for EHD (10.3%). The rice kernel breakage during milling process is the most important problem for this staple food crop. Head rice yield (HRY), i.e., the mass percentage of rice kernels that remain as sound rice after milling, is the most important quality characteristic in rice industry (Yang et al., 2003). HRY reduction decreases the economical values of rice, since broken kernels are typically worth half the value of sound ones (Cnossen et al., 2003). HRY is especially sensitive to drying conditions and is commonly taken as a standard index to assess the effect of drying system on rice quality (Yang et al., 2003). Because of the meaningful reduction of fissured kernels in HPD compared to EHD, this method could be applied for rough rice drying.

Discussion

As presented in Table 1, the power consumed by the electrical heater in HPD mode is much lower than its value in EHD mode. This is due to increase in the inlet air temperature to the fan when heat pump system is used. In spite of the constant value of the air velocity during the experiments, the higher power consumption by the centrifugal fan in HPD mode is related to the higher air pressure drop in this mode compared to the EHD mode. Although, the total power consumption of the HPD was lower than its value for the EHD, because of reducing the evaporating capability of the drying air as a result of inefficient evaporating in the heat exchanger, the $SMER_{ws}$ of the EHD was more than its value for the HPD mode (Table 1). The measured mean value of the heat delivered in the condenser (as shown in Table 2) was meaningfully smaller than its required value predicted by HYSYS software (Table 3). This is due to the limitation in selection of the heat exchangers (evaporator and condenser). Moreover, the amount of heat exchanged between the external condenser and the ambient air should be also considered in computing of the COP. This was calculated to be 174.7 W by HYSYS software (Table 3). However, this heat was not measurable in practice. As mentioned the percentage of fissured kernels after drying with HPD was considerably less than the corresponding value for EHD. During drying process, the grain kernels lose their moisture to the drying air due to higher vapor pressure within the kernels (Sarker et al., 1996). As the kernel surface moisture content decreases, the initial water transfers toward the surface by diffusion in order to eliminate the moisture gradients inside the material. This phenomenon continues until the end of drying operation. Due to moisture reduction, the surface cells undergo more shrinkage than internal ones, which in turn cause differential stresses inside the kernel. These stresses, if sufficiently large, cause the kernel to fissure (Cnossen et al., 2003). In this research, due to slower drying of rough rice in HPD method the moisture content gradients were less than those with EHD method and hence, the fissured kernels percentage in HPD was approximately 40% lower than its value for EHD.

Materials and methods

Dryer set-up design

In order to increase the capability of the considered laboratory dryer, its design was carried out based on fluidized bed dryers (FBDs) with active hydrodynamic. To design the dryer, the most common rough rice variety in Isfahan province (central Iran), namely Sazendegi was selected. The samples were prepared from Isfahan Center for Research of

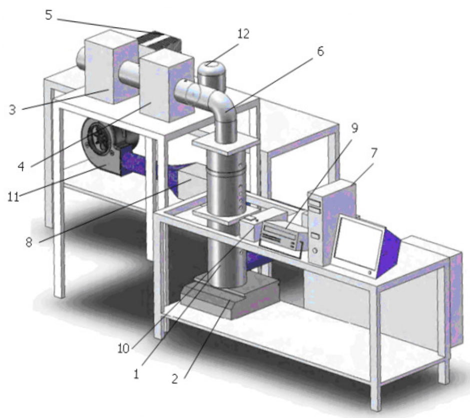

Fig 1. Schematic diagram of the setup: (1) drying chamber, (2) digital scale, (3) condenser, (4) evaporator, (5) external condenser, (6) flexible tube, (7) PC, (8) electrical heater, (9) digital multi-meter, (10) frequency inverter, (11) centrifugal fan and (12) compressor.

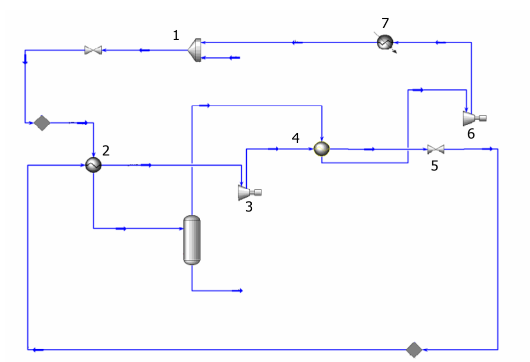

Fig 2. Heat pump dryer arrangement without external condenser: (1) dryer, (2) evaporator, (3) compressor, (4) condenser, (5) expansion valve, (6) centrifugal fan and (7) electrical heater.

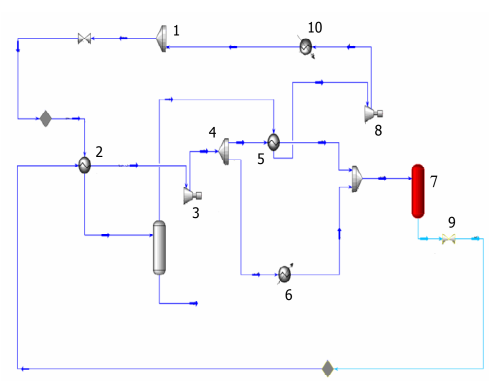

Fig. 3. Heat pump dryer arrangement with an external condenser parallel to the main condenser: (1) dryer, (2) evaporator, (3) compressor, (4) division valve, (5) condenser, (6) external condenser (7) refrigerant reservoir, (8) centrifugal fan, (9) expansion valve and (10) electrical heater.

Agricultural Science and Natural Resources. Moisture content of the samples was determined according to the ASAE standard (2001). Table 5 shows the measured dimensions and densities of Sazendegi variety at different moistures contents. Equivalent diameter, sphericity and statically porosity of Sazandegi rough rice were calculated by the related equations (Mohsenin, 1986). Table 6 shows the calculated parameters at different moisture contents. With respect to the measured and determined physical properties of the selected product, dryer setup was designed based on the governing equations for the FBDs, which its detail can be found in Sadeghi et al. (2004).

Heat pump design

The selection of the best heat pump configuration is the most important challenge in designing of the heat pump dryers. The most efficient configuration of heat pump is affected by ambient condition, drying rate, and stage of product drying. It is so hard to achieve a desirable configuration and drying condition for best performance of HPDs (Saensabi and Prasertsan, 2003). At the first step, fully closed loop was selected for the desired heat pump system. The properties (temperature and humidity) of inlet and outlet air in heat pump are essential for designing heat pump systems. Based on the selected configuration for the setup, the heat pump inlet air is the same as the dryer outlet air. So, it was necessary to simulate the grain drying in a deep bed mode and determine the dryer outlet air properties. To simulate grain drying in this condition, a differential volume was considered in an arbitrary position of a batch bed of grain. After solving four partial differential equations for air enthalpy, material enthalpy, air humidity and moisture content of the material in MATLAB software, the dryer outlet air properties were obtained for different conditions which its detail can be found in Naghavi (2008). In next step, the obtained results were used for heat pump dryer simulation which was conducted in HYSYS software. R134a was selected as the working fluid of the heat pump due to its plus points (Miroslaw, 1993). High and low pressures of the working fluid in the heat pump were considered to be 1000 and 100 kPa, respectively. According to the minimum fluidization velocity and safe air temperature for drying Sazandegi rough rice variety, air velocity and temperature were selected 1 m s^{-1} and 35 °C, respectively. As the first idea, an evaporator and a condenser shown in Fig. 2 were considered for the heat pump system. The system was not stable with this arrangement of heat exchangers. Therefore, in order to stabilize the system through shedding of the energy from system, an external air-cooled condenser was added to the system as shown in Fig. 3. After running the program, it was revealed that the considered arrangement was appropriately stable.

Drying experiments

Drying experiments were carried out in deep bed mode (20 cm depth) and two methods of HPD and EHD. The air velocity and temperature were 1 m s^{-1} and 35 °C, respectively. The weight of the bed was noted at 5 min intervals. Final moisture content of the samples was considered as 11-12% (w.b.), which is suitable for storage of rough rice.

Table 4. Rough rice fissured kernel percentage before and after drying process.

Treatments	Fissured kernels (%)
Before drying	9.3±0.88
After drying by EHD	19.3±1.76
After drying by HPD	15.3±1.20

HPD and EHD stand for heat pump dryer and electrical heated dryer, respectively.

Table 5. Dimensions and densities of Sazandegi rough rice variety.

Moisture content (%, w.b.)	Length (mm)	Width (mm)	Thickness (mm)	Bulk density (kg m^{-3})	True density (kg m^{-3})
11-13	8.65	2.80	1.84	578.9	1116.6
13-15	8.74	2.44	1.86	573.3	1135.9
15-17	8.65	2.48	1.92	578.3	1141.2

Table 6. Equivalent diameter, sphericity and porosity of Sazandegi rough rice variety.

Moisture content (%, w.b.)	Equivalent diameter (mm)	Sphericity (-)	Porosity (-)
11-13	3.41	0.39	0.48
13-15	3.41	0.39	0.49
15-17	3.46	0.40	0.48

Determination of fissured kernels

To determine the fissured kernels after drying and compare two drying methods (HPD and EHD), 48 h after drying operation, 100 kernels of each sample were manually husked and then, the fissured kernels were determined by a binocular.

Performance evaluation of the heat pump

In order to measure the energy consumption by the electrical heater and the centrifugal fan, a digital multi-meter and a portable digital wattmeter were used, respectively. The multi-meter was connected to a PC for data logging. The wattmeter was placed before the frequency inverter in order to reduce and measure the power consumption of the centrifugal fan. The overall performance of a HPD may be characterized by several criteria. Among them the coefficient of performance (*COP*) and specific moisture extraction rate (*SMER*) are the most important ones (Oktay and Hepbasli, 2003).
COP of the heat pump system was obtained as follows (Oktay and Hepbasli, 2003):

$$COP_{hp,h} = \frac{\dot{Q}_{cd}}{\dot{W}_c} \tag{1}$$

For normal application, the power consumption comes from the compressor of the heat pump. In this study the compressor power consumption ($\dot{W}_c$) was used to represent the input power (Oktay and Hepbasli, 2003). Also instead of measuring the mass flow rate on the refrigerant side, the heat delivered in the condenser ($\dot{Q}_{cd}$), was estimated as follows (Oktay and Hepbasli, 2003):

$$\dot{Q}_{cd} = \dot{m}_{air} C_{p,air} (T_{co,air} - T_{ci,air}) \tag{2}$$

where,

$$\dot{m} = \rho_{air} \dot{V}_{air} \tag{3}$$

To measure the $T_{ci,air}$ and $T_{co,air}$, two thermocouples (K type) were used in air inlet and outlet of condenser, respectively. The thermocouples were connected to a data logger (ls2winul model). Figs. 4 and 5 show the variation of $T_{ci,air}$ and $T_{co,air}$ with respect to the time, respectively.
SMER is defined as the energy required for removing 1 kg of water. It may be related to the input power into the compressor ($SMER_{hp}$) for HPD, or to the total power into the dryer including the fan and heater powers ($SMER_{ws}$) for HPD and EHD, as follows (Oktay and Hepbasli, 2003):

$$SMER_{hp} = \frac{\dot{m}_d}{\dot{W}_c} \tag{4}$$

And

$$SMER_{ws} = \frac{\dot{m}_d}{(\dot{W}_f + \dot{W}_h + \dot{W}_c)} \tag{5}$$

Conclusions

In this study a heat pump dryer was designed and developed based on simulation of rough rice drying process and numerical solution of partial differential equations carried out in MATLAB and HYSYS softwares. The system was evaluated from dried product quality and energy consumption aspects. The results showed that the fabricated heat pump dryer reduced the fissured kernels value about 40%. In spite of consuming power by compressor in HPD mode, the

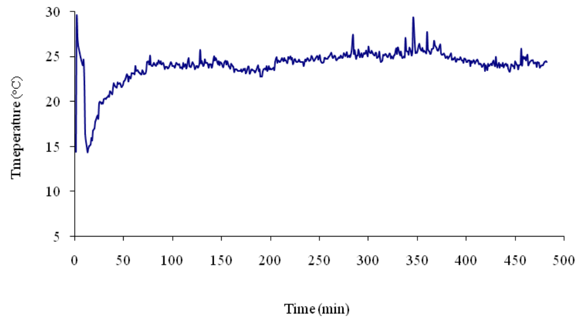

Fig 4. Time variation of air temperature at condenser inlet.

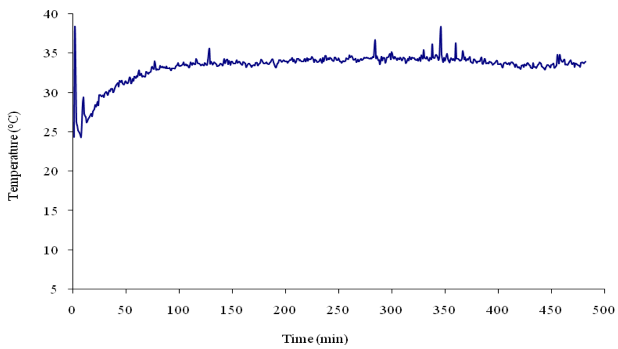

Fig 5. Time variation of air temperature at condenser outlet.

total power consumption reduced about 10% compared to the EHD mode. Since, the most important challenge in rice milling industry is kernels fissuring, HPD method could be applied for rough rice drying. To improve the low obtained *COP* for the fabricated system, the more efficient heat exchangers are needed. Therefore, it is recommended to study the different configuration of the heat pump in order to achieve the best condition of drying in next research steps.

Acknowledgments

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