

Drying kinetics and energy-exergy analysis of an experimental heat pump dryer utilized horseshoe heat recovery heat pipes for drying different grapes

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ABSTRACT

In this study, nine different seeded and seedless grape cultivars (*Vitis* spp) were dried in a closed circuit heat pump-drying (HPD) system. The system's drying performance was determined for each grape cultivar, and exergy analysis was performed. The evaporator efficiency increases when the external condenser is in the "on" position. This revealed that when the system reached the desired drying temperature, the longer the external condenser stayed in the "on" position, the more moisture can be drawn from the product. The heat pump highest coefficient of performance total (COP_{total, hp}) and system coefficient of performance total (COP_{total, sys}) were found to be 3.714 and 3.035, respectively, in the external condenser "on" position. The highest exergy efficiency values of compressor, condenser, TXV, evaporator and drying cabinet were found to be 68 %, 75.41 %, 92.43 %, 77.35 % and 78.28 % respectively, when the external condenser "on" position. The highest exergy efficiencies were determined 75.75 %, 69.81 %, 87.5 %, 86.65 % and 80.46 % respectively, when the external condenser "off" position. The experiments were evaluated in order to study the drying characteristics in HPD system. The changes in moisture ratio during the drying process were compared by six thin-layer drying models and then assessed in terms of various statistical criteria like root-mean-square-error (RMSE), diminished chi-square (χ^2), and determination constant (R^2). Fick's second law of diffusion was used to obtain diffusivity of effective-moisture values, and they were found to range between 4.715×10^{-11} and 1.280×10^{-10} m²/s.

Introduction

Drying is among the important techniques for providing original flavour, preserving nutritional properties, reducing packaging costs and transportation costs, and it also increases the shelf life of fresh foods. Drying technology has the most important share in terms of energy consumption among the processes applied in the processing of post-harvest agricultural products. The hot air drying technique is traditionally a widely used method. However, this method has some disadvantages such as high energy consumption and high drying temperatures. For this reason, alternative innovative drying technologies have been developed and used in recent years. [1–4]. Heat pump systems, which can also be used in combination with renewable energy sources, are used alone or in combination with different methods in the

drying process. Heat pump drying (HPD) technology is a very successful drying method with low energy consumption. In the literature, it has been reported that the energy consumption of HPD systems is 60 %–80 % less than that of the conventional ones when compared under the same temperature conditions [5,6]. Besides, there is a high energy requirement in drying techniques like freeze, hot air, microwave, oven, vacuum, and infrared drying [7]. HPD system can control easily regarding reducing the products to the desired moisture content (MC). One of the most striking advantages of HPD is that it allows easy control of humidity and drying air temperature (DAT). Therefore, it is a suitable method for drying products that are sensitive to temperature and humidity [8]. Depending on the ambient conditions, it is possible to operate HPD systems in the mode of open-loop or closed-loop (the energy of the air rejected out of dryer is used for recirculation). Different drying methods can be used for grape drying, in one of these methods [9], dried the

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Nomenclature			
$MR_{exp,i}$	Experimental dimensionless moisture ratio	H_0	dead state relative humidity (%)
$MR_{pre,i}$	Predicted dimensionless moisture ratios	T_A	condenser outlet air temperature / dryer inlet air temperature (K)
N	Number of observations	T_B	dryer outlet air temperature (K)
DR	Drying rate	T_C	evaporator inlet air temperature (K)
COP	Coefficient of performance	T_D	evaporator outlet air temperature (K)
$\dot{W}_{comp}$	Power consumption of compressor, kW	T_E	condenser inlet air temperature (K)
$\dot{Q}_{cd}$	heat delivered rate in internal condenser, kW	η	efficiency
$\dot{Q}_{hhr2}$	condensation section of heat recovery device in capacity, kW	Subscripts	
$\dot{W}_f$	Power consumption of fan, kW	a	Air
$\dot{m}_a$	mass flow rate (air, kg/s)	r	refrigerant
$\dot{m}_r$	mass flow rate (refrigerant, kg/s)	comp	Compressor
A	surface area (m ²)	f	Fan
ρ	density of air (kgm ⁻³)	hp	Heat pump
v	air speed (m/s)	sys	System
h_A	specific enthalpy (internal condenser air outlet, kJkg ⁻¹)	HPD	Heat pump drying
h_B	specific enthalpy (dryer air outlet, kJkg ⁻¹)	total	total
h_C	Specific enthalpy (evaporator air inlet, kJkg ⁻¹)	loss	loss
h_D	Specific enthalpy (evaporator air outlet, kJkg ⁻¹)	f	fresh
h_E	Specific enthalpy (internal condenser air inlet, kJkg ⁻¹)	dry	dryer
C_{pam}	specific heat of moist air (kJ/kg ⁻¹ K ⁻¹)	in	inlet
C_{pa}	specific heat of dry air (kJ/kg ⁻¹ K ⁻¹)	out	outlet
C_{pv}	specific heat of water vapor air (kJ/kg ⁻¹ K ⁻¹)	hhr1	evaporation section of heat recovery
ω_a	specific humidity of air (kJkg ⁻¹)	cd	condenser
T_0	dead state temperature (K)	ev	evaporator

grapes in a drying greenhouse with a solar collector. It was determined that when the DAT was between 28.8 °C and 55.94 °C, the moisture contents of the grapes was dropped to 18 % (based on wet basis) in 128 h. Achariyaviriya et al. [10] suggested a mathematical model for fruit HPD system and they determined that the ambient conditions significantly influence the performance HDP system which operated open or close loop. Hu et al. [11] designed a bi-evaporator HDP with a condensation temperature of 60 °C. The results revealed that drying time drops with the raise of tray spacing, air velocity and DAT. Li et al. [12] investigated the influence of compressor speeds on the COP, MER and SMER. The results show that as the compressor speeds increased, COP and SMER decreased, while the DAT and MER increased. A novel air cycle HDP system is proposed by Zhan et al. [13]. The proposed system saved approximately 15–27 % of energy compared to an electric heater dryer. In another study, Ren et al. [14] designed and installed a high temperature closed-loop HPD system. It was found that DAT increased with decreasing cooling water flow rate and increasing inlet temperature of cooling water.

Exergy analysis can be used to determine the irreversibility's of different points of thermal systems. There are studies on exergy analysis of HPD systems in the literature [15,16]. One of these studies, exergy analysis was carried out by Kuzgunkaya & Hepbasli [17] to a ground-source (geothermal) HPD system which was drying of laurel leaves. They determined that the exergy efficiency values vary between 81.35 % and 87.48 % depending on the airflow. Colak et al. [18] performed the exergy analysis of HPD system which was designed for drying of mint leaves. They found that the exergy efficiency values ranged from 76.03 % to 97.24 % at 40–50 °C DAT when the mass flow rates the air were between the 0.01–0.05 kgs⁻¹. In their studies, Karaaslan et al. [19] dried sultan grapes by using a microwave dryer. As a result, they determined that the drying time dropped significantly with increment of the microwave power and temperature. Aktaş et al. [20] performed on the grated carrot drying used a HPD system and infrared supported HPD system. Compared to HPD, infrared supported HPD provided 48.8 %

drying time savings however; the increase in energy consumption was 68 %. The maximum and minimum exergy efficiencies were between 66.8 % and 31.6 %. Atalay [21] performed exergoeconomic analysis both of solar dryer and HPD. It was determined that the condenser should be improved in terms of exergoeconomic factors in the HPD. Singh et al. [22] conducted the energy and exergy analysis of potato and banana chips in an open and closed loop HPD. As a result, the COP was calculated in the range of 3.89 to 2.85. The exergy efficiency values of compressor, condenser, TXV, evaporator and drying cabinet were found to be 69.9 %, 82 %, 86.5 %, 52.6 % and 33.5 % respectively, in closed loop mode. The exergy losses were calculated as 0.233 kW, 0.099 kW, 0.049 kW, 0.1645 kW and 0.0452, respectively.

In this study, nine different grape cultivars were dried in a closed circuit HPD system with digital thermostat control with a constant velocity of air (3 ms⁻¹) and temperature (45 °C). Energy-exergy analyses of different grape cultivars were made with the help of HPD system utilizing horseshoe heat recovery heat pipes (external condenser). The exergy losses/efficiency of the main components of this system used in the study was calculated. Comparison was made with or without external condenser. Additionally, drying kinetics of the products were determined using six different models and the most suitable model was determined.

Material and methods

In this study, nine different grapes cultivars were dried. Grapes grown in Yalova Atatürk Horticultural Central Research Institute were dried in a closed-circuit HPD system. The gravines are six years old and they were grown in the same experimental vineyard in Yalova, Turkey. The grapes were analyzed during the 2021 growing season in this study. The grape cultivars and abbreviations used in the study are as follows: Muscat Blue (MB), Mars Seedless (MS), Kay Gray (KG), Glenora (G), Rhea (R), Campbell Early (CE), Kishmish Vatkana (KV), Philipp (P), and Özer Beyazı (OB). The main characteristics of the grape cultivars used

Table 1
The main properties of the grape cultivars [23].

Cultivar	Species	Berry Colour	Seed Status	Special Flavour
Mars Seedless	Interspecies	Black	Seedless	Foxy
Campbell Early		Black	Seeded	Foxy
Glenora		Rose	Seedless	No
Muscat Bleu		Black	Seeded	No
Philipp		Black	Seeded	No
Kay Gray	<i>V. vinifera</i>	Yellow/Green	Seeded	Foxy
Kishmish		Black	Seedless	No
Vatkana		Rose	Seedless	No
Rhea		Yellow/Green	Seedless	No
Özer Beyazı		Yellow/Green	Seedless	No

are shown in Table 1.

Experimental system

Experimental system mainly consists of heat pump, fan, air channels and a drying chamber. The interior volume of the drying chamber is $53 \times 73 \times 154 \text{ cm}^3$, polystyrene foam was used as insulation material. In the drying system, ten stainless steel chamber trays sized $40 \times 60 \times 2 \text{ cm}^3$ were used. A metal frame was manufactured to carry the trays and it was attached to the hook of the load cell. Series-connected an internal and an external condenser were used in the experiment system. The control of the solenoid valves were made with a digital thermostat (which can be adjusted to the desired drying temperature). The external condenser that was connected in series with the inner one was activated at desired times. Thus, it was ensured that the drying chamber was kept at a constant temperature within a differential range ($\pm 1^\circ\text{C}$). Schematic illustration and a photograph of the manufactured experimental HPD system was given in Fig. 1a and b.

In case of need, the external condenser was activated and it was aimed to increase the performance of the system by reducing the load of the internal condenser. With the increase of the condenser capacity, the evaporation capacity also increased and thus more moisture was taken from the air leaving the drying chamber. Horseshoe heat recovery circuit with heat pipe is given in Fig. 2.

The HHR system consists of two heat exchangers (HHR1 and HHR2) and copper tubes connecting them to each other. The HHR system is three-tubes and contains R410A as the refrigerant and there is no energy input to activate the refrigerant. The refrigerant inside the HHR system evaporates or condenses depending on the velocity of the air passing over the heat exchanger. The heat exchanger between B-C is used for evaporation, between C-D for evaporator and between D-E for condensation. Table 2 represents the technical properties of the equipment used in the HPD system.

Experimental procedure

For stabilization, the drying temperature was brought to 45°C before each experiment and the drying process was started by placing the trays. In each trial, grape berries weighing 6000 g were placed on trays and dried. Weight was measured every 10 min using the load cell weight indicator. With the timer connected to the energy input of the system, the system was operated for 12 h and turned off for 1 h. Humidity and temperature measurements were carried out at one minute intervals from five different points. Erbay et al. [24] determined that HPD systems demonstrated the best performance in the temperature range of 35 to 45°C . In studies with HPD systems, the drying air velocities are taken in the range of 0.5 to 4.5 m/s [25–28]. In this study, the air velocity was kept constant at 3 m/s . Nine different types of grapes were dried at 3 ms^{-1} air velocity and 45°C drying temperature. Testo 405-V1 and Testo 410-2 brand air velocity meters were used to measure the air flow

velocity. Drying process is given in Fig. 3.

The wax components on the surface of grapes were removed before they were dried. For this, an emulsified solution was prepared by adding 1 % olive oil to 5 % K_2CO_3 . The products were immersed in this solution and left for two minutes. It was then filtered and placed in plastic drying crates and taken to the dryer. Large bunches were divided into medium-sized pieces. The main purpose of applying this mixture is to raise the water-permeability of the grape skin by disrupting the monomolecular-structure of the wax-layer on the surface of the grapes [29]. After placing the products on the drying tray in a homogeneous way, we started the process of drying. The mass loss measurement was carried out using the load cell, and the indicator was used to record the data to computer every ten minutes. Enthalpy (h) values were found from the psychometric diagram by using the relative humidity (%) and temperature (T) values measured from points A, B, C, D and E (Fig. 1). The drying procedure was stopped once a decrease of 16 % (wet basis) was observed in the moisture contents of the samples. All the experiments were conducted in 3 replicates. A digital manifold was used to determine the low/high pressures of the refrigerant, and a 4-channel digital thermometer via pipes was used to measure the temperature of the circulating fluid in the system. The measuring devices used in different stages of the study and their technical details are given in Table 3.

Mathematical modelling

Moisture ratios (MRs) were calculated using the equation (1):

$$MR = \frac{M_t - M_e}{M_0 - M_e} \quad (1)$$

where M_t is the moisture-content (ct), and M_e is the equilibrium-ct, M_0 is the initial-ct, at any time. Drying times are given in minutes. Considering the value of M_t or M_0 , M_e is so small that it can be negligible. So, the equation (1) can be simplified as follows: $MR = M_t/M_0$ [30].

As seen in Table 4, the six (thin-layer-drying) models were placed to the drying curves (time versus moisture ratio). Levenberg-Marquardt-algorithm was employed in the nonlinear analysis to determine the parameters of the models. The goodness-of-fit between the predicted moisture values and the experimental ones was evaluated using the following indicators: χ^2 and R^2 . The lower the value of χ^2 , the better the fit. The RMSE value (ideally 0) gives insight into how the experimental values deviate from the predicted ones. If R^2 values are close to 1, this can be interpreted as a good fit [31]. The R^2 , χ^2 , and RMSE values can be computed using the formulas (2), (3), and (4), respectively:

$$R^2 = 1 - \frac{\sum_{i=1}^N (MR_{pre,i} - MR_{exp,i})^2}{\sum_{i=1}^N (MR_{pre} - MR_{exp,i})^2} \quad (2)$$

$$\chi^2 = \frac{\sum_{i=1}^N (MR_{exp,i} - MR_{pre,i})^2}{N - z} \quad (3)$$

$$RMSE = \left[\frac{1}{2} \sum_{i=1}^N (MR_{pre,i} - MR_{exp,i})^2 \right]^{\frac{1}{2}} \quad (4)$$

Determining of effective moisture diffusion

Drying takes place mostly during the falling-rate-period and internal diffusion controls moisture (ω) transfer. The MR in Eq. [6] is determined using Fick's second law of unsteady-state diffusion [Equation (5)]. The following equations show the solution of the diffusion Eq. for infinite-slab reported by Crank [37].

$$\frac{\partial M}{\partial t} = D_{eff} \nabla^2 M \quad (5)$$

$$MR = \frac{8}{\pi^2} \sum_{n=0}^{\infty} \frac{1}{(2n+1)^2} \exp\left(- (2n+1)^2 \pi^2 \frac{D_{eff} t}{4L^2}\right) \quad (6)$$

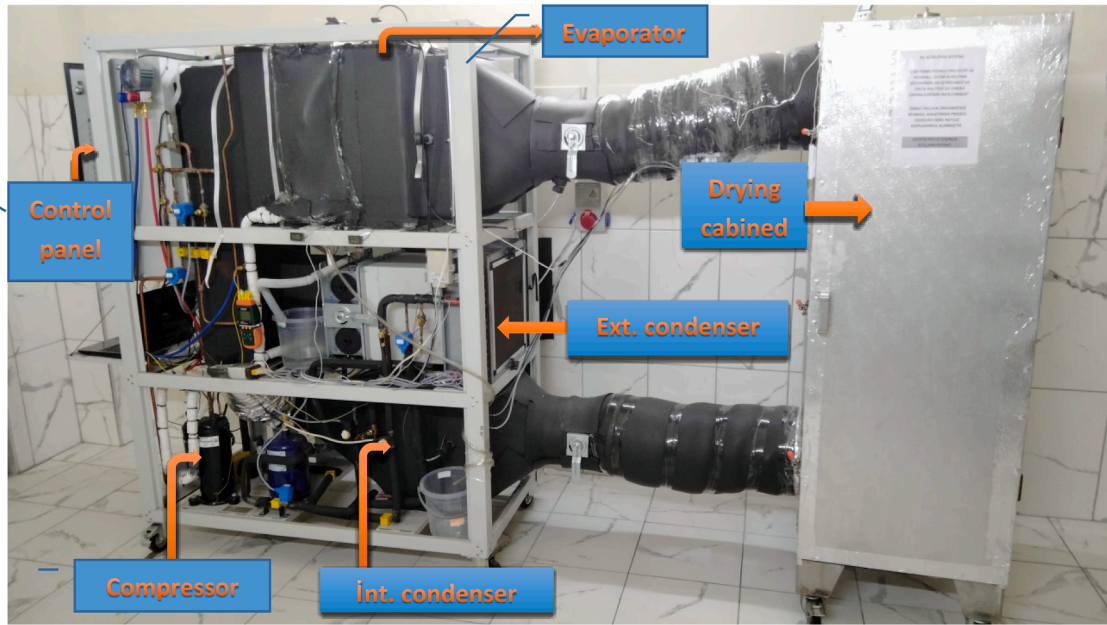
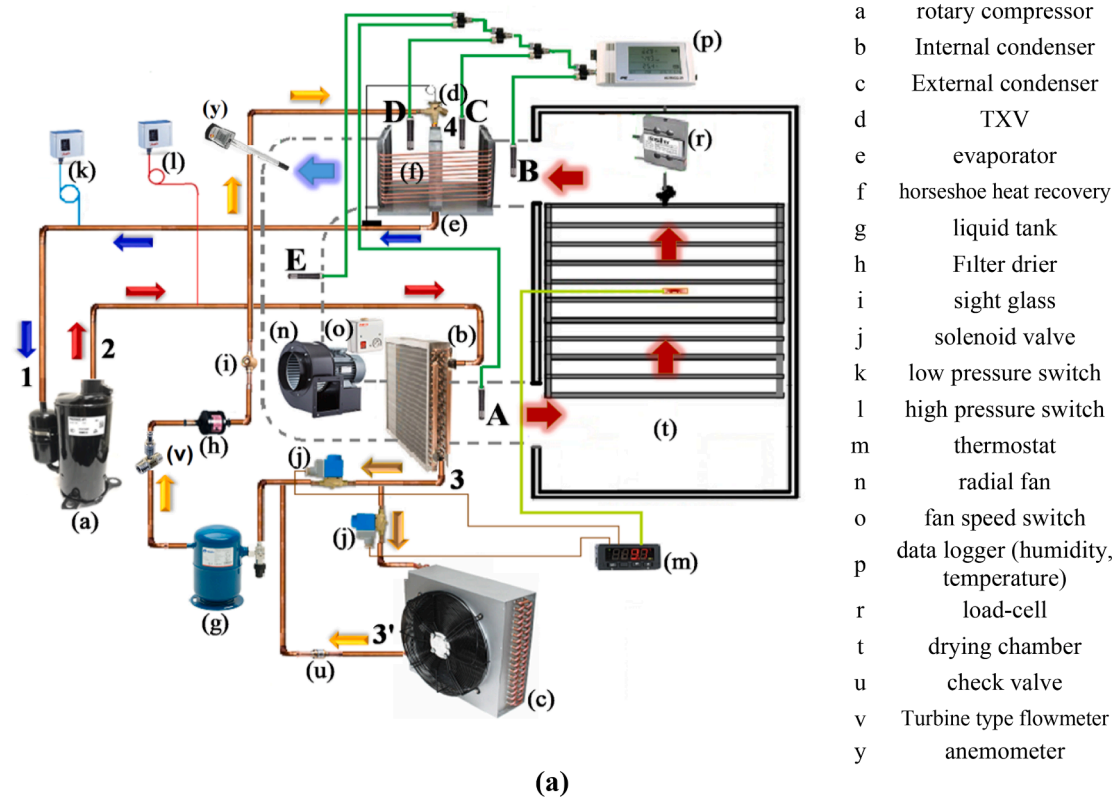


Fig. 1. a) Closed circuit HPD system b) photograph of HDP.

where L to half thickness of samples (m), n refers to a positive-integer, D_{eff} refers to effective diffusivity (m^2s^{-1}), and t to time (s). If the drying period is long, the Eq. [6] can be simplified as follows.

$$MR = \frac{8}{\pi^2} \exp\left(-\pi^2 \frac{D_{eff} t}{4L^2}\right) \quad (7)$$

$$\ln(MR) = \ln\left(\frac{8}{\pi^2}\right) - \left(\pi^2 \frac{D_{eff} t}{4L^2}\right) \quad (8)$$

$$K = \frac{\pi^2 D_{eff}}{4L^2} \quad (9)$$

Applying natural-logarithmic at both sides linearizes the Eq. [7] [see the Eq. [8]], where D_{eff} was found using the slope (K) of the graph ($\ln MR$ vs time).

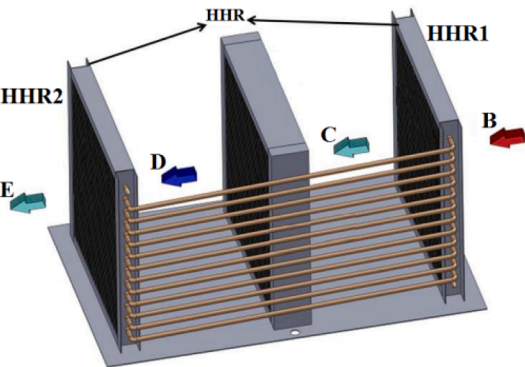


Fig. 2. Horseshoe (wrap around) heat recovery circuit with heat pipe.

Table 2
Properties of experimental HPD system components.

System Elements	Technical Properties
Refrigeration compressor	Power 5200 W, rotary type.
Internal evaporator	Aluminium lamella, heat transfer surface area, 8.70 m ²
Fan of the drying system	Radial type (350 W)
Condenser (external)	heat transfer surface area, 5.4 m ² , Aluminium lamella,
Internal condenser	heat transfer surface area, 12.35 m ² , Aluminium lamella
Condenser fan (external)	Axial type (75 W)
Thermostatic expansion valve	externally balanced, orifice type, evaporation temperature range (+10/−40 °C)
Refrigerant	Freon – 410A
Horseshoe (wrap around) heat pipes heat recovery circuit	three-pipe, heat transfer surface area, 9.75 m ² , Aluminium lamella,

Energy and exergy analysis of the HPD system

Energy analysis

Air side;

The drying air inlet temperatures to the condenser and the air velocities are important factors determining the HPD system' energy performance.

The condenser capacity at the air side can be found from the

Table 3
Properties of the measurement devices used in the experimental system.

Part Measured	Device Used	Measurement Range	Accuracy
Product Weight	Load cell (ESIT STCS-50)	0 ~ 50 kg	± 0.05 g
Speed of the drying air	Thermal anemometer (TESTO 405)	0.1 ~ 10 ms ⁻¹	± 0.1 ms ⁻¹
Temperature of drying chamber	Digital thermostat NTC sensor (EVCO evkb21)	−50 ~ +130 °C	±0.1 °C
Power consumed by fans and compressors	Digital electric meter (MAKEL)	150 V ~ 300 V	0.5 sday ⁻¹
Humidity and temperature of the ambient air	Vane anemometer (TESTO 410-1)	−10 ~ +50 °C 0.4 ~ 20 ms ⁻¹	±0.5 °C ± 0.2 ms ⁻¹
Working pressure of the condenser/evaporator of the heat pump	Digital Manifold (TESTO 570-2)	0 ~ 50 bar	± 0.1 bar
Working outer pipe temperatures of the heat pump	4-channel datalogger thermometer (Extech)	−40 ~ +250 °C	±0.5 °C
DAT and relative humidity	5-channel (A-B-C-D-E) datalogger (Humlog20 E)	0 ~ 100 % RH −40 ~ +80 °C	± 2 % RH ± 0.4 °C
Turbine type flowmeter	stainless steel, liquid	0.0065 ~ 0.325 kgs ⁻¹	±0,2



Fig. 3. Drying process.

Table 4

Mathematical models used for describing the grape cultivars' drying kinetics.

Model name	Model ¹⁾	References
Vega – Galvez I	$MR = \exp(n + kt)$	[32]
Lewis	$MR = \exp(-kt)$	[33]
Page	$MR = \exp(-kt^n)$	[34]
Midilli & Kucuk	$MR = a \exp(-kt^n) + bt$	[35]
Logarithmic	$MR = a \exp(-kt) + c$	[36]

¹⁾ a , b , c , k , and n are the empirical constants and coefficients.**Table 5**The equations for the η_{ex} and exergy loss of the HPD system.

Components	Exergy loss (kW)	Exergy efficiency (%)
Compressor	$Ex_{loss,comp} = W_{in} + Ex_{r1} - Ex_{r2}$	$\eta_{ex,comp} = \frac{Ex_{r2} - Ex_{r1}}{W_{in}}$
Condenser	External condenser off $Ex_{loss,cd} = Ex_{r2} - Ex_{r3} + Ex_{aE} - Ex_{aA}$	$\eta_{ex,cd} = \frac{Ex_{aA} - Ex_{aE}}{Ex_{r2} - Ex_{r3}}$
		$\eta_{ex,cd+hhr2} = \frac{Ex_{aA} - Ex_{aD}}{Ex_{r2} - Ex_{r3}}$
	External condenser on $Ex_{loss,cd} = Ex_{r2} - Ex_{r3'} + Ex_{aE} - Ex_{aA}$	$\eta_{ex,cd} = \frac{Ex_{aA} - Ex_{aE}}{Ex_{r2} - Ex_{r3'}}$
		$\eta_{ex,cd+hhr2} = \frac{Ex_{aA} - Ex_{aD}}{Ex_{r2} - Ex_{r3'}}$
Thermostatic expansion valve	$Ex_{loss,exp} = Ex_{r3} - Ex_{r4}$	$\eta_{ex,exp} = \frac{Ex_{r4}}{Ex_{r3}}$
Evaporator	$Ex_{loss,ev} = Ex_{r4} - Ex_{r1} + Ex_{aC} - Ex_{aD}$	$\eta_{ex,ev} = \frac{Ex_{aD} + Ex_{r1}}{Ex_{aC} + Ex_{r4}}$
	$Ex_{loss,ev+hhr1} = Ex_{r4} - Ex_{r1} + Ex_{aB} - Ex_{aD}$	$\eta_{ex,ev+hhr1} = \frac{Ex_{aD} + Ex_{r1}}{Ex_{aB} + Ex_{r4}}$
	$Ex_{loss,dry} = Ex_{aA} - Ex_{aB}$	$\eta_{ex,dry} = \frac{Ex_{aB}}{Ex_{aA}}$

Table 6

Measuring range and uncertainty values of measuring instruments.

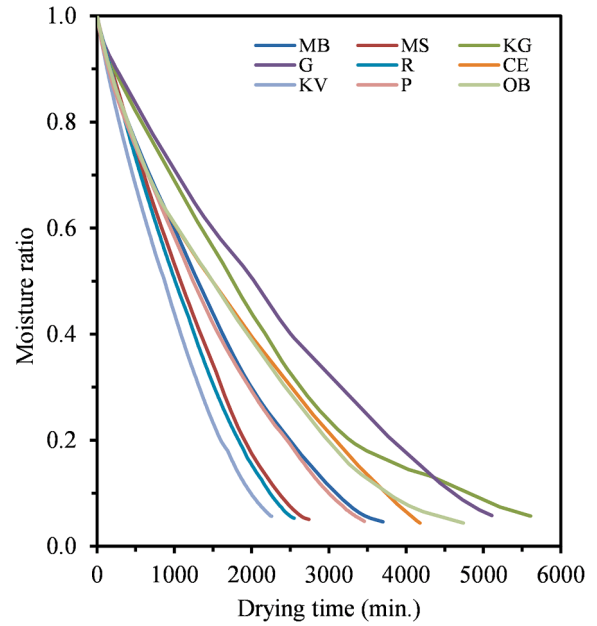
Part Measured	Measurement Range	Uncertainty
Product Weight	0 ~ 50 kg	±0.055 %
Drying air speed	0.1 ~ 10 ms ⁻¹	± 0.101 ms ⁻¹
Humidity and temperature of the ambient air	-10 ~ +50 °C, 0.4 ~ 20 ms ⁻¹	± 0.201 ms ⁻¹ , ± 0.502 °C
Power consumed by fans and compressors	150 V ~ 300 V	±1.31 %
Working pressure of the condenser and evaporator of the heat pump	0 ~ 50 bar	± 0.52 %
Working outer pipe temperatures of the heat pump	-40 ~ +250 °C	± 0.505 °C
Temperature of the drying chamber	-50 ~ +130 °C	± 0.103 °C
Relative humidity and temperature of the drying air	0 ~ 99 % RH, -40 ~ +80 °C	±0.217 %, ±0.403 °C
Turbine type flowmeter	0.0065 ~ 0.32 kgs ⁻¹	±0.202

following equation;

$$\dot{Q}_{cd} = \dot{m}_a \cdot C_{pam} \cdot (T_A - T_E) \quad (10)$$

 $\dot{m}_a$ is calculated as follows.

$$\dot{m}_a = A \cdot \rho \cdot \theta \quad (11)$$

**Fig. 4.** Drying curves of grape cultivars at a temperature of 45 °C. C_{pam} is calculated from,

$$C_{pam} = C_{pa} + \omega_a \cdot C_{pv} \quad (12)$$

The heat dissipated by the condensation section of the horseshoe (wrap around) heat pipes heat recovery device (Fig. 2, C-D section).

$$\dot{Q}_{hhr2} = \dot{m}_a \cdot C_{pam} \cdot (T_E - T_D) \quad (13)$$

The ratio of the total amount of heat transferred to the drying air from the condenser and heat recovery devices to the compressor power is defined as Coefficient of Performance (COP).

$$COP_{total, hp} = \frac{\dot{Q}_{cd} + \dot{Q}_{hhr2}}{W_{comp}} \quad (14)$$

The total COP.

$$COP_{total, sys} = \frac{\dot{Q}_{cd} + \dot{Q}_{hhr2}}{W_{comp} + W_f} \quad (15)$$

For refrigeration side;

The analyses were performed in two different ways, with or without the use of an external condenser.

Condensing capacity (external condenser off).

$$\dot{Q}_{cd} = \dot{m}_r (h_{r2} - h_{r3}) \quad (16)$$

Condensing capacity (external condenser on).

$$\dot{Q}_{cd} = \dot{m}_r (h_{r2} - h_{r3'}) \quad (17)$$

Exergy analysis

In this part of the study, the effects of $\dot{m}_a$ and ambient temperature on the η_{ex} of the HPD system are investigated. In addition, exergy efficiencies and exergy losses were determined for each system component. The flow was assumed to be in a steady state in the analysis and air was considered as an ideal gas. Friction and pressure drops were neglected and the isentropic efficiency of the compressor was taken 100 %. Exergy balance equations were used to determine the exergy efficiencies.

The equilibrium equations are given below:

$$\sum_{in} \dot{m} = \sum_{out} \dot{m} \quad (18)$$

Table 7

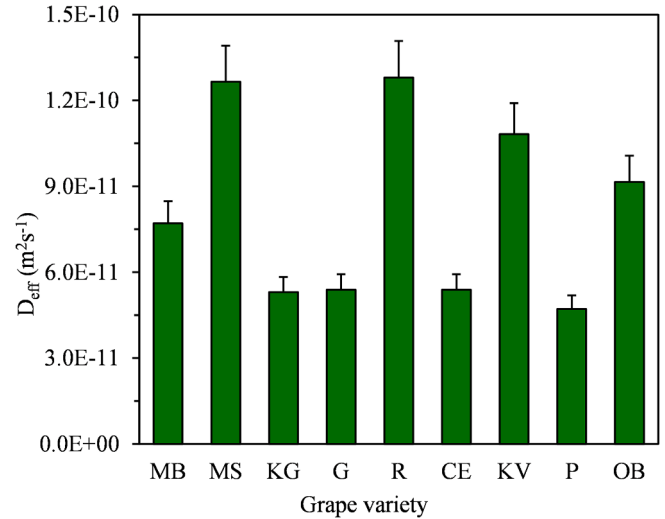
Moisture contents and drying times of grape cultivars.

Cultivar	MB	MS	KG	G	R	CE	KV	P	OB
Moisture content (g-water/g-dry matter)	4.097	3.757	3.318	3.264	3.570	4.302	3.352	4.033	4.359
Drying times (minute)	3700	2740	5610	5110	2550	4180	2260	3460	4740

Table 8

Statistical analysis results of various models for drying of grape cultivars.

Cultivar	Model	R ²	χ ²	RMSE
MB	Lewis	0.9802	0.001498	0.634637
	Henderson and Pabis	0.9835	0.001253	0.591061
	Logarithmic	0.9991	0.000068	0.135176
	Page	0.9937	0.000472	0.356641
	Midilli and Kucuk	0.9993	0.000053	0.110084
MS	Vega-Galvez et al.	0.9835	0.001253	0.591061
	Lewis	0.9647	0.002882	0.804219
	Henderson and Pabis	0.9751	0.002040	0.670037
	Logarithmic	0.9986	0.000114	0.144811
	Page	0.9944	0.000454	0.315705
KG	Midilli and Kucuk	0.9990	0.000076	0.114435
	Vega-Galvez et al.	0.9751	0.002040	0.670037
	Lewis	0.9887	0.000835	0.616487
	Henderson and Pabis	0.9912	0.000653	0.531026
	Logarithmic	0.9953	0.000342	0.388048
G	Page	0.9887	0.000838	0.635142
	Midilli and Kucuk	0.9981	0.000137	0.234244
	Vega-Galvez et al.	0.9912	0.000653	0.531013
	Lewis	0.9790	0.001478	0.727998
	Henderson and Pabis	0.9815	0.000683	0.691613
R	Logarithmic	0.9996	0.000024	0.092673
	Page	0.9913	0.000612	0.488319
	Midilli and Kucuk	0.9997	0.000014	0.064302
	Vega-Galvez et al.	0.9815	0.001301	0.691623
	Lewis	0.9690	0.002464	0.706897
CE	Henderson and Pabis	0.9783	0.001725	0.581532
	Logarithmic	0.9993	0.000053	0.099697
	Page	0.9961	0.000391	0.276869
	Midilli and Kucuk	0.9996	0.000027	0.070902
	Vega-Galvez et al.	0.9783	0.001725	0.581532
KV	Lewis	0.9850	0.000965	0.528601
	Henderson and Pabis	0.9850	0.000966	0.524825
	Logarithmic	0.9974	0.000166	0.202959
	Page	0.9865	0.000869	0.526582
	Midilli and Kucuk	0.9997	0.000016	0.067221
P	Vega-Galvez et al.	0.9850	0.000966	0.524827
	Lewis	0.9769	0.001751	0.553159
	Henderson and Pabis	0.9822	0.001351	0.487366
	Logarithmic	0.9993	0.000047	0.086452
	Page	0.9942	0.000436	0.273226
OB	Midilli and Kucuk	0.9994	0.000043	0.084078
	Vega-Galvez et al.	0.9822	0.001351	0.487368
	Lewis	0.9823	0.001280	0.556170
	Henderson and Pabis	0.9842	0.001142	0.539903
	Logarithmic	0.9995	0.000036	0.095499
	Page	0.9925	0.000537	0.364973
	Midilli and Kucuk	0.9995	0.000029	0.079024
	Vega-Galvez et al.	0.9842	0.001143	0.539902
	Lewis	0.9867	0.000935	0.590068
	Henderson and Pabis	0.9869	0.000925	0.600242
	Logarithmic	0.9979	0.000148	0.232518
	Page	0.9903	0.000685	0.522636
	Midilli and Kucuk	0.9979	0.000143	0.226491
	Vega-Galvez et al.	0.9869	0.000925	0.600245

**Fig. 5.** Calculated effective moisture diffusivity values of grape cultivars.

$$E_{xa} = \dot{m}_a \cdot C_{pam} \cdot \left[(T_a - T_0) - T_0 \cdot \ln\left(\frac{T_a}{T_0}\right) \right] \quad (21)$$

The equations of exergy efficiency and loss for HPD system main components are given in Table 5 [22,24].

The basic assumptions for HPD:

- All the processes were operated at steady state,
- Dead state circumstances were chosen as $P_0 = 101.325$ kPa for the air. Since the drying times are long in the drying of grape cultivars, the ambient T_0 and H_0 values differed. These differences were given in Tables 7 and 8 for each grape cultivar.
- The specific heat and gas constant values of vapor were as follows: $C_{pa} = 1.005$ kJkg⁻¹K⁻¹ and $R_a = 0.287$ kJkg⁻¹K⁻¹.
- The software package “CoolPack” and “EES” was used to determine the thermodynamic characteristics of the air and refrigerant (R410A).

In the equations, the “a” index refers to the drying air and the “r” index to the refrigerant.

The uncertainty analysis applied to the system in this study is an influential method when it is used in the evaluation of experiments. Uncertainty in the experimental examinations is explained by [38]. R refers to the function of the independent variables $y_1, \dots, y_n$. $u_1, \dots, u_n$ refer to the uncertainty in the independent variables and U_R is the resulting uncertainty. Thus, the uncertainty is calculated by,

$$U_R = \left[\left(\frac{\partial R}{\partial y_1} u_1 \right)^2 + \left(\frac{\partial R}{\partial y_2} u_2 \right)^2 + \dots + \left(\frac{\partial R}{\partial y_n} u_n \right)^2 \right]^{1/2} \quad (22)$$

Table 6 shows the measuring range and uncertainty values for the measurement instruments.

$$\sum_{in} (\dot{m}_i h + \dot{Q} + \dot{W}) = \sum_{out} (\dot{m}_i h + \dot{Q} + \dot{W}) \quad (19)$$

The exergies of the refrigerant and moist air, respectively, are calculated for each case.

$$E_{xr} = \dot{m}_r \cdot [(h - h_0) - T_0 \cdot (S - S_0)] \quad (20)$$

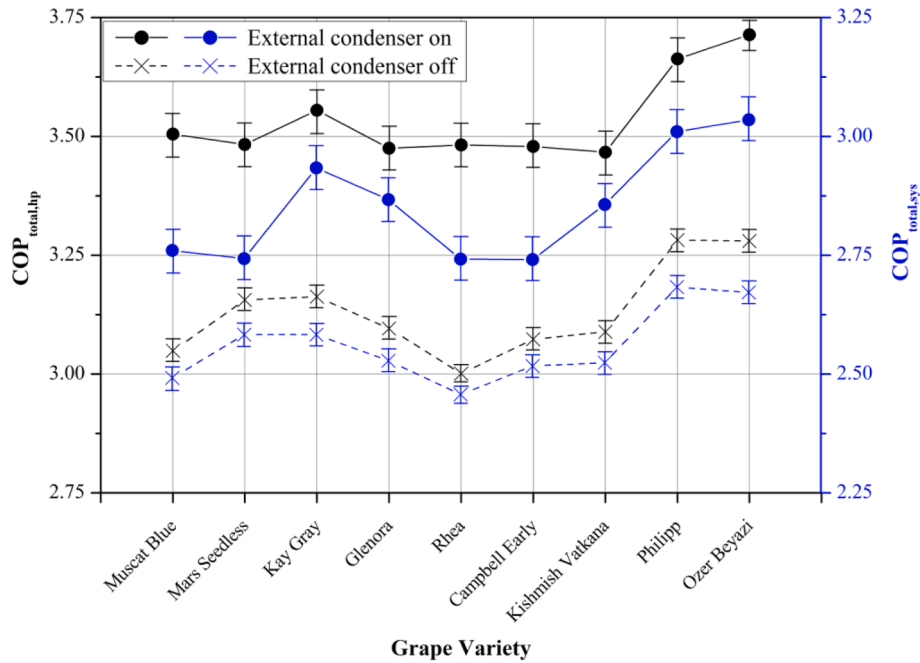


Fig. 6. The changing of $COP_{total,hp}$ and $COP_{total,sys}$.

Table 9

Exergy loss and exergy efficiency values external condenser “off”.

Grape cultivar	Ambient conditions		Exergy loss (W)						
	T_0 (°C)	H_0 (%)	$Ex_{loss,dry}$	$Ex_{loss,comp}$	condenser		$Ex_{loss,exp}$	evaporator	
					$Ex_{loss,cd}$	$Ex_{loss,cd+hhr2}$		$Ex_{loss,ev}$	$Ex_{loss,ev+hhr1}$
Muscat Blue	29	50	46	538	383	375	284	615	684
Mars Seedless	30	48	47	593	278	261	241	675	736
Kay Gray	28	52	43	593	333	325	240	656	732
Glenora	28	50	42	614	334	326	236	631	707
Rhea	29	56	41	677	278	266	214	633	701
Campb. Early	30	55	42	664	292	274	186	640	703
Kish Vatkana	26	54	35	614	268	265	295	546	632
Philipp	25	40	34	531	261	264	231	624	718
Özer Beyazı	24	61	31	527	259	261	234	602	701

Grape cultivar	Ambient conditions		Exergy efficiency (%)						
	T_0 (°C)	H_0 (%)	$\eta_{ex,dry}$	$\eta_{ex,comp}$	Condenser		$\eta_{ex,exp}$	Evaporator	
					$\eta_{ex,cd}$	$\eta_{ex,cd+hhr2}$		$\eta_{ex,ev}$	$\eta_{ex,ev+hhr1}$
Muscat Blue	29	50	63.35	68.00	49.42	52.57	89.49	72.66	74.72
Mars Seedless	30	48	62.58	64.93	53.92	63.68	91.87	73.67	75.32
Kay Gray	28	52	66.52	64.56	68.59	72.93	91.79	73.68	75.75
Glenora	28	50	68.97	63.21	68.47	72.33	91.81	74.16	76.28
Rhea	29	56	69.62	59.92	62.45	69.44	92.43	73.91	75.85
Campb. Early	30	55	68.85	60.81	55.38	67.49	92.16	74.50	76.25
Kish Vatkana	26	54	76.08	62.32	72.91	74.17	88.98	74.68	77.35
Philipp	25	40	76.00	66.59	73.15	75.25	91.64	73.00	75.66
Özer Beyazı	24	61	78.28	66.22	74.12	75.41	91.32	73.04	75.93

Results and discussion

Drying curves

Fig. 4 represents the impact of temperature on the grape cultivars' drying curves during the drying process. The drying curves have typical features in similar fruits. The moisture contents were found to decrease steadily with drying time. Moisture contents (before drying) and drying times of the grape cultivars can be seen in Table 7. The grape cultivars were left for drying in a HDP at a drying temperature of 45 °C. The drying process was continued until the grapes reached a MC of $0.19 \pm$

0.01 g-water/g-dry matter. We are of the opinion that the differences in drying times are due to the structure of grape cultivars, water and dry matter content, berry size and length.

Assessment of the models

Moisture content values of the grape cultivars were used to calculate the moisture ratios (MRs). The model yielding the lowest RMSE and χ^2 values and the highest R^2 value was identified as the best fit. Table 6 shows the results for the statistical computation. R^2 of all the models were found to be more than 0.96. Based on results of the statistical

Table 10

Exergy loss and exergy efficiency values external condenser “on”.

Grape cultivar	Ambient conditions		Exergy loss (W)						
	T ₀ (°C)	H ₀ (%)	Ex _{loss,dry}	Ex _{loss,comp}	Condenser		Ex _{loss,exp}	Evaporator	
					Ex _{loss,cd}	Ex _{loss,cd+hhr2}		Ex _{loss,ev}	Ex _{loss,ev+hhr1}
Muscat Blue	29	50	32	379	319	305	426	505	572
Mars Seedless	30	48	33	398	248	228	468	489	549
Kay Gray	28	52	36	367	307	293	455	496	567
Glenora	28	50	32	372	293	280	443	464	537
Rhea	29	56	30	365	264	245	342	475	541
Campb. Early	30	55	29	383	231	208	399	419	478
Kish Vatkana	26	54	33	468	385	376	421	339	423
Philipp	25	40	31	472	408	405	414	330	420
Özer Beyazı	24	61	32	483	434	433	401	288	383

Grape cultivar	Ambient conditions		Exergy efficiency (%)						
	T ₀ (°C)	H ₀ (%)	η _{ex,dry}	η _{ex,comp}	condenser		η _{ex,exp}	evaporator	
					η _{ex,cd}	η _{ex,cd+hhr2}		η _{ex,ev}	η _{ex,ev+hhr1}
Muscat Blue	29	50	69.11	74.94	52.21	59.32	84.34	75.87	78.07
Mars Seedless	30	48	69.79	73.79	51.96	65.33	83.21	79.65	81.79
Kay Gray	28	52	72.68	75.57	58.62	66.24	83.52	76.29	78.64
Glenora	28	50	73.77	74.61	62.54	69.81	83.55	77.01	79.48
Rhea	29	56	70.98	74.48	52.81	64.39	87.50	78.09	80.22
Campb. Early	30	55	70.31	72.95	47.32	63.74	85.79	80.66	82.64
Kish Vatkana	26	54	76.39	66.86	53.83	57.52	83.91	81.63	84.71
Philipp	25	40	79.61	65.43	56.59	57.97	83.85	81.51	84.86
Özer Beyazı	24	61	80.46	64.72	59.74	59.61	84.09	82.98	86.64

computations, it can be asserted that Midilli and Kucuk' model was the best-fit drying model with the lowest χ^2 and $RMSE$ and the highest R^2 (Table 8). This model's R^2 was found to be between 0.9979 and 0.9997, its χ^2 between 0.000016 and 0.000143, and its $RMSE$ between 0.067221 and 0.226491. Doymaz [39], Coşkun et al. [40], and Polat & İzli [41] reported similar results for various agro industrial products.

Effective moisture diffusivity

Fig. 5 shows the D_{eff} calculated using the Eq. (9) for different grape cultivars. According to Fig. 5, the calculated D_{eff} values of MB, MS, KG, G, R, CE, KV, P and OB grape cultivars were found as 7.708×10^{-11} , 1.265×10^{-10} , 5.301×10^{-11} , 5.389×10^{-11} , 1.280×10^{-10} , 5.390×10^{-11} , 1.082×10^{-10} , 4.715×10^{-11} , and $9.152 \times 10^{-11} \text{ m}^2\text{s}^{-1}$, respectively. Among these values, the highest value was obtained in Rhea grapes, while the smallest D_{eff} value was obtained in Philipp grapes. These values were found to be between 10^{-12} and $10^{-8} \text{ m}^2\text{s}^{-1}$, which is similar to those reported in the studies in the literature for drying agro products [42]. These results are in line with those reported by Doymaz & Altınar [43], who found the effective diffusion coefficients of seedless grapes as 1.048×10^{-10} and $6.919 \times 10^{-10} \text{ m}^2\text{s}^{-1}$ in convective drying, and Bai et al. [44] who examined the convective drying of grapes and found the effective diffusion coefficients within the range of 1.08×10^{-10} – $1.65 \times 10^{-10} \text{ m}^2\text{s}^{-1}$. In another study, it was reported by Miraei Ashtiani [45] that the D_{eff} values for grapes ranged between 3.51×10^{-9} and 4.54×10^{-9} . These differences in D_{eff} may be due to the differences in type, size, and shape of cultivars, their initial/final moisture contents, compositions, and tissue characteristics of grapes.

Results of energy and exergy analysis

The COP values calculated during the drying of the grape cultivars are given in Fig. 6, according to the on-off status of the external condenser. As seen in Fig. 6, the highest COP values were obtained when the external condenser was on. In this case, the $COP_{total,hp}$ and $COP_{total,sys}$ values were found to be 3.714 and 3.035, respectively.

Analyses of energy and exergy were conducted based on the data at the sixth hour of the experiments for each grape cultivar. The η_{ex} and

loss values were calculated for the main components of the closed circuit HPD system, that is, expansion valve, compressor, condenser, drying chamber, and evaporator. Calculated values were given in Table 9 for the case where the external condenser was off, and in Table 8 for the case when it was on. These results are compatible with the literature [46].

The exergy efficiency and exergy loss values obtained as a result of the calculations were summarized in Table 10. The exergy losses were found 31–47 W in the drying chamber, 527–677 W in the compressor, 261–383 W in the condenser, 186–284 W in the TXV and 615–736 W in the evaporator in the range. As the condenser temperature increases, the exergy loss in the compressor increases. When the external condenser was activated, as seen in Table 9, the exergy loss in the compressor decreased between 10 % and 23 % on average. The highest exergy loss was found in the evaporator and compressor. The highest exergy efficiency was found for drying chamber as 78.28 %. These results are compatible with the studies by Kuzgunkaya & Hepbasli [17] and Singh et al. [22]. The exergy efficiency values in the compressor, condenser, TXV and evaporator were calculated as 68 %, 75.41 %, 92.43 % and 77.35 %, respectively.

The results obtained according to the exergy equations given in Table 4 were shown in Table 8. The highest exergy loss for were calculated for evaporator (between 288 and 572 W). The exergy losses for other equipment were found in the range of 365–483 W in the compressor, 231–434 W in the condenser, 342–468 W in the TXV, and 29–36 W in the drying chamber. The highest exergy loss was occurred in the evaporator and compressor. Additionally, drying chamber had the highest exergy efficiency among the components (80.46 %). Moreover, the exergy efficiency values in the compressor, condenser, TXV and evaporator were calculated as 75.57 %, 69.81 %, 87.5 % and 86.64 %, respectively. When the external condenser was activated, the compressor power decreased and the exergy loss decreased between 40 and 44 %. Similarly, the loss in the evaporator decreased by 20–40 %. Exergy efficiencies were determined according to whether the external condenser was on or off in the system. Accordingly, the exergy efficiency was increased by 3.65 % in the drying chamber, 7.42 % in the compressor, 5.79 % in the evaporator, and was decreased by 7.90 % in the condenser (when the external condenser was on).

Conclusion

Drying kinetics and energy-exergy analysis of an experimental HP dryer utilized horseshoe heat recovery heat pipes (external condenser) were performed in this study. In order to determine the performance parameters of the system, the HPD system was tested with and without external condenser and drying characteristics of nine different grape cultivars were assessed. Drying characteristics of different grape cultivars were investigated in a HPD system at a temperature of 45 °C and an air velocity of 3 ms⁻¹. When the external condenser was activated, the exergy efficiency of the evaporator increased between 5 and 15 %. The exergy efficiencies decreased between 5 and 8 % when the external condenser was “off” position. This showed that when the system reached the desired drying temperature, the longer the external condenser stayed in the “on” position, the more moisture can be drawn from the product. It was found that evaporator and compressor had the highest exergy loss among the components of the HPD system. Moreover, the compressor the exergy loss decreased between 40 and 44 % when the external condenser was on. In the light of the experimental results obtained from the grape drying experiments, it was concluded that the Midilli and Küçük’ model was the most suitable model. The D_{eff} were found to be within the range of 4.715×10^{-11} and 1.280×10^{-10} ms⁻¹. As future study, the heat pump drying system can be operated with solar assisted.

CRedit authorship contribution statement

Cüneyt Tunçkal: Validation, Writing – original draft, Writing – review & editing, Visualization, Supervision, Project administration. **Mehmet Direk:** Methodology, Validation, Conceptualization, Investigation, Formal analysis, Writing – original draft, Writing – review & editing. **İbrahim Doymaz:** Validation, Investigation, Writing – original draft, Writing – review & editing, Visualization. **Zekiye Göksel:** Methodology, Validation, Conceptualization, Writing – original draft, Writing – review & editing. **Arif Atak:** Methodology, Validation, Conceptualization, Writing – original draft, Writing – review & editing.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Data availability

The data supporting the findings of this study are available within the article

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