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Raising the Drying Unit for Fruits and Vegetables Energy Efficiency by Application of Thermoelectric Heat Pump

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Abstract: Drying food stuffs and other materials belongs to one of the most commonly used feedstock processing techniques, featuring rather high energy consumption. The major disadvantage of conventional electric convective-type household dryers is substantial thermal energy emission into the environment with a wet exhaust, worked-out drying agent. Among other principal disadvantages common to all dryers of this type, the following have to be mentioned: spatial inhomogeneity of heating a product under processing and that of drying agent distribution due to its temperature reduction and relative humidity growth as it moves upwards. A block diagram and a breadboard model of a convective-type thermoelectric dryer employing a thermoelectric heat pump have been designed. In our approach, a product is treated with the help of a drying agent (normally, heated air) with partial exhaust-air recirculation and heat recovery. Laboratory studies of the drying process have been carried out using apple fruits as a test material in order to evaluate the power consumed for evaporation of 1 kg of water in the newly developed convective-type thermoelectric drying unit. Physical parameters of apple fruits before and after drying both in the thermoelectric drying unit and in a conventional series-produced convective-type domestic dryer have been reported. The energy efficiency of the newly designed drying unit has been compared with that of some series-produced samples. It has been found out that, unlike conventional convective-type dryers, the breadboard model of the developed thermoelectric drying unit features a smoother product drying process owing to the presence of side air channels and more effective drying agent path organization in the processing chamber. This conclusion was supported by the results of the carried out tests. Application of thermoelectric heat pumps with the function of the exhaust drying agent heat recovery will make it possible to reduce the drying agent heater installed capacity and the power consumed by the newly designed convective-type thermoelectric drying unit by up to 20% in the course of the drying process, compared to series-produced household convective-type dryers.

Keywords: convective-type drying; thermoelectric heat pump; thermoelectric Peltier unit; drying food stuffs; heat energy recuperation; air recirculation; energy saving

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1. Introduction

Thermal drying of products and materials is an integral stage of many agricultural production technologies and belongs to the most energy-demanding processes [1]. Dried vegetables, fruits, and berries can be stored in rather good condition and easily transported. Their storage does not require large area sizes, and they are widely used in the food industry as feed stocks for manufacturing various products [2]. Theoretical studies have been made to define the time and heating rate for cut-up apples in the radiant-heat drying method. Experimental studies of the drying process in a radiant-heat dryer for wild fruits and vegetables have been carried out as well [3,4].

In paper [5], an algorithm of energy-saving optimization for grain-cleaning-drying technological complex was described, making it possible to reduce both the direct energy intensity and power consumption of flow processing lines, and a method was proposed for calculating the potential of direct energy savings in the frames of existing technical-technological limits and financial stringencies. An energy-saving operation mode of a conductive-convective mini-grain dryer was suggested that partially uses the heat of solar radiation captured with the help of a solar collector, as well as that stored in a heat accumulator, with the addition of thermal energy received by recirculation of the exhaust drying agent [6]. An automated agricultural feedstock drying system was designed to ensure biotechnological heating conditions for particular product types, making it possible to retain the properties of its essential biological components [7]. Theoretical and experimental studies were performed in order to evaluate the effectiveness of thermoelectric cooler use for collecting water in the process of outdoor air dehumidification, including the case when they are applied in combination with solar photoelectric panels [8,9]. Jaber et al. [10] carried out theoretical studies of a thermoelectric cogeneration drying unit for heating air and water and electric energy production equipped with a multistep flue gas heat utilization system. Thermoelectric units for air cooling and heating were proposed to be integrated in the air duct system on premises, and the corresponding theoretical and experimental studies were made [11,12]. Design options of air dehumidification thermoelectric units were proposed and studied [13–15]. Malik et al. [16] carried out theoretical studies of the efficiency of solar photoelectric panels for the case of their application in combination with thermoelectric units in order to reduce the temperature of the solar cells, thus reducing the power loss of the entire photoelectric system. The results of studies on raising the efficiency of heat utilization by thermoelectric converters aimed at electric power production growth were reported [17–19].

The possibility of heat pumps being used in the drying process was considered in papers [20,21]. Junior et al. [22] proposed a design of a convective-type dryer with a thermoelectric heat pump equipped with a recuperative heat exchanger. A mathematical model describing the designed dryer operation was proposed in order to evaluate its productivity. A design for the vacuum freeze dryer was developed by employing thermoelectric units whose parameters and operation modes were substantiated [23].

Theoretical and experimental studies were performed in order to evaluate the operation efficiency of the designed thermoelectric dryer for clothes [24,25]. The experimental study results for thermoelectric heat pumps designed for drying green crops [26] and for a solar thermoelectric generator for drying grapes [27] were reported.

In household dryers, as well as in industrial dryer units of small productivity, the convective method for heating a product under processing is normally applied. In the majority of series-produced convective-type household electric units designed for drying vegetables and fruits, in the process of drying food stuffs, thermal energy generated by an electric heater, in the course of heating the air, is released into the environment along with the wet air. On the other hand, even partial wet air recirculation around a closed path without dehumidification will make the drying process go on for a long time, thus degrading the quality of a product under processing and not bringing a significant energy-saving effect [28].

Therefore, the major disadvantage of convective-type electric dryers is thermal energy loss with moisture-laden air released into the environment. One more drawback of dryers having similar designs is the nonuniformity of the heating product under processing and the decrease in the temperature of the drying agent as it moves up through the pallets in the dryer housing. As a result, products placed on the bottom pallets dried faster compared to those placed on the upper ones. As a result, to ensure uniform drying of food products, it is necessary to periodically swap the bottom and lower pallets of the dryer. From the literature review, it can be concluded that the main areas of thermoelectric heat pump and Peltier elements application are ventilation systems for buildings where supply air is processed (dehumidification, heating, and cooling), as well as units using renewable energy sources. At the same time, very few works are devoted to applied research on increasing the drying process efficiency by improving the design of convection household dryers and assessing the energy-saving potential when drying food products using a thermoelectric heat pump.

The aim of our research is to carry out experimental studies of the apple fruit drying process for evaluating the specific amount of power consumed in the course of evaporation of 1 kg of water in the newly designed convective-type thermoelectric drying unit in order to compare its energy efficiency with that of the series-produced samples.

2. Materials and Methods

One of the effective methods to improve the energy efficiency of technological processes in agro-industrial complexes is the application of thermoelectric units and assemblies based on Peltier elements in various technological cycles, such as air dehumidification and heating [29,30], pasteurization and cooling milk [31], and young animals local heating [32]. In paper [33], the application of a thermoelectric heat pump designed to heat a drying agent in drying units was substantiated, providing a considerable reduction of power consumption and maintaining high quality indicators of a final product in drying processes compared to conventional convective-type household dryers mainly employing a tubular electric heater as a drying agent heating device.

The operating principle of a thermoelectric heat pump is as follows: the cold circuit of the thermoelectric assembly absorbs some part of thermal energy from the exhaust air in the drying unit and transfers it to the hot circuit, thus increasing thermal energy production on the hot side of the thermoelectric unit. This thermal energy is used for heating air to reduce the power consumed from the electric grid.

The newly designed thermoelectric assembly represents a thermoelectric heat pump on the basis of Peltier elements TEC1-22818 performing the drying agent heating function (see Figure 1). Specifications of Peltier elements are presented in Table 1.

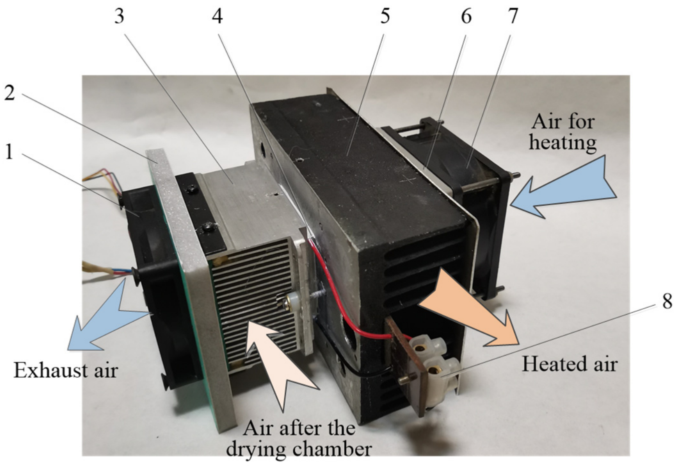


Figure 1. Overall view of the thermoelectric assembly based on Peltier elements: 1—cold-circuit fan; 2, 6—heat-insulating liner; 3—cold-circuit air radiator; 4—Peltier element; 5—hot-circuit air radiator; 7—hot-circuit fan; and 8—Peltier element power terminal block.

Table 1. Thermoelectric module specifications.

Module Type	Input Voltage, V	Input Current, A	Max. Temperature Difference of the Two Sides, °C	Max. Power Consumption, W	Dimensions, mm	Weight, g
TEC1-22818	24	16	70	360	50 × 50 × 4	35

Based on the carried out theoretical and experimental studies, the block diagram and a laboratory sample of the convective-type thermoelectric dryer comprising a thermoelectric assembly have been designed.

The function principle of the newly designed thermoelectric drying unit operating in the heat pump mode is shown in Figure 2 [34].

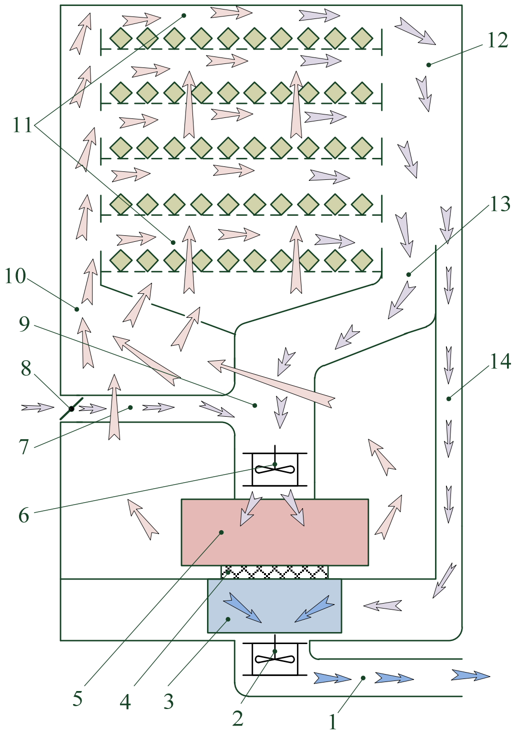


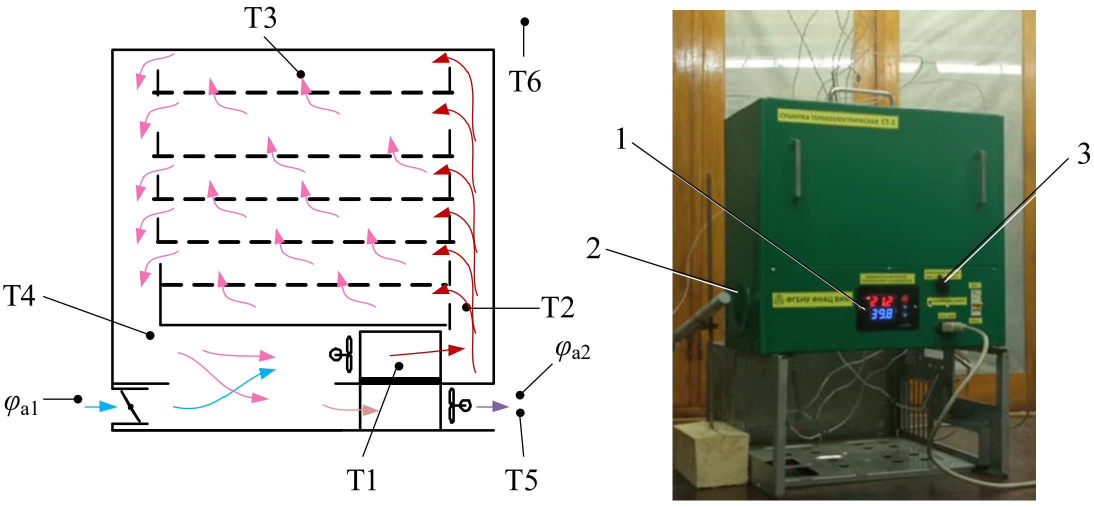
Figure 2. Block diagram of the drying unit using a thermoelectric assembly operating in the heat pump mode: 1—outlet duct; 2—exhaust-air fan; 3—cold-junction radiator; 4—Peltier elements; 5—hot-junction radiator; 6—circulation fan; 7—air supply channel; 8—air intake damper; 9—mixing chamber; 10—air supply duct; 11—drying chamber; 12—exhaust-air chamber; 13—circulation air duct; and 14—exhaust-air channel.

A product under processing is heated by the hot air with partial recirculation and exhaust-air heat recovery. The drying agent is additionally heated as it passes through thermoelectric assembly 5, and then it is transported into drying chamber 11 via air supply duct 10. Then, the drying agent flow coming from air exhaust chamber 12 gets divided. One of its parts is transported into air exhaust channel 14, while the other is directed into circulation air duct 13. Outdoor air enters mixing chamber 9 through air intake damper 8 and air supply channel 7, wherefrom it is directed to hot-junction radiator 5 of the thermoelectric heat pump with the use of circulation fan 6. From exhaust-air channel 14, the drying agent is transported onto cold-junction radiator 3 of the thermoelectric heat pump, where exhaust agent heat recovery takes place. Then, the air is put out into the environment with the help of exhaust-air fan 2 via exhaust-air channel 1. Outdoor air is captured via intake damper 8. Outdoor air passes the casing of the main power supply unit (220VAC/24VDC), where it is previously heated by cooling internal electric circuit components.

3. Results and Discussion

Based on the newly designed block diagram, a physical model of the thermoelectric drying unit was manufactured for studying the drying process parameters.

Figure 3 shows the block diagram of the thermoelectric drying unit with air temperature and humidity sensors positions along with the laboratory sample layout.



a

b

Figure 3. Air temperature T1–T6 and humidity sensor (φ_{a1} and φ_{a2}) positions in the drying chamber (**a**): T1—hot radiator surface; T2—air supplied into the drying chamber after heating; T3—air in the drying chamber; T4—air leaving the drying chamber; T5—exhaust air removed from the drying unit; T6—ambient air; φ_{a1} is air relative humidity in the intake of drying unit; and φ_{a2} is air relative humidity in the output from the drying unit; layout of the thermoelectric drying unit with the air temperature and humidity sensors (**b**): 1—temperature and humidity control meter; 2—outlet for exhaust drying agent; and 3—air intake damper control.

3.1. Laboratory Tests of the Newly Designed Thermoelectric Drying Unit

Laboratory tests of the drying process with apple fruits grade "Gala" subject to processing were performed on five pallets in both the thermoelectric drying unit and household convective-type dryer ROTOR SH-002 with the following parameters: initial values of moisture content were 80%, ambient air temperature was 20 ± 1 °C, and relative humidity was 45 ± 2%. Temperature and humidity values met the requirements of [35]. Temperature measurements were carried out with Chromel–Kopel thermocouples connected to temperature metering and control unit TRM138 at several control points inside the body of the dryers under study. Relative air humidity was recorded by sensors at the supply opening for the intake of outside air and at the outlet for exhaust spent drying agent. The total energy consumption during the drying process was recorded using a single-phase electric energy meter with accuracy class 1.

The residual moisture content determination in the dried product (using the example of apples) was carried out by an indirect method, according to the recommendations given in [36].

Averaged parameters of apples before and after drying in the thermoelectric drying unit obtained in the course of laboratory tests are presented in Table 2.

Table 2. Parameters of apples before and after drying in the thermoelectric drying unit.

	Before Drying			After Drying			
Pallet No.	**Weight of Pallet and Apples, g**	**Pallet Weight, g**	**Weight of Apples, g**	**Weight of Pallet and Apples, g**	**Weight of Apples, g**	**Weight of Evaporated Water, g**	**Amount of Evaporated Water, %**
1	446.4	171.6	274.8	235.7	64.1	210.7	76.67
2	438.5	170.4	268.1	233.3	62.9	205.2	76.54
3	420.1	173.4	246.7	225.3	51.9	194.8	78.96
4	462.3	172.4	289.9	236.4	64	225.9	77.92
5	444.6	175.4	269.2	233.8	58.4	210.8	78.31
Total:	2211.9	863.2	1348.7	1167.2	304	1044.7	

Averaged parameters of the apple fruit drying process in convective-type dryer ROTOR SH-002 obtained in the course of laboratory tests are presented in Table 3.

Table 3. Parameters of apples before and after drying in convective-type dryer ROTOR SH-002.

Before Drying				After Drying			
Pallet No.	**Weight of Pallet and Apples, g**	**Pallet Weight, g**	**Weight of Apples, g**	**Weight of Pallet and Apples, g**	**Weight of Apples, g**	**Weight of Evaporated Water, g**	**Amount of Evaporated Water, %**
1	623.1	252.6	370.4	351	98.4	272	73.43
2	593.2	252.4	340.6	353	100.6	240	70.46
3	634.0	252.0	382.0	377	125	257	67.28
4	618.9	252.3	366.7	387	134.7	232	63.27
5	583.1	251.9	331.1	381	129.1	202	61.01
Total:	3052	1261.2	1790.8	1849	587.8	1203	

Figure 4 shows the layout of convective-type dryer ROTOR SH-002 with the air temperature and humidity sensors in the course of laboratory tests.

Figure 4. Overall view of the convective-type dryer ROTOR SH-002 with temperature and humidity sensors.

Table 4 presents the estimation results of power consumption for 1 kg of evaporated water in a newly designed thermoelectric dryer in comparison with some of its series-produced analogues.

Table 4. Comparison of household convective-type dryers in terms of energy consumption per 1 kg of evaporated water for the apple fruit drying process.

Household Dryer		**Power Consumed from the Electric Grid, W**	**Number of Pallets**	**Energy Consumption, kW·h per 1 kg Evaporated Moisture (Calculated Data)**
1	Veterok-2	600	5	2.6
2	Hyundai HYFD-S1202	500	12	2.5
3	Vitek VT 5054	300	5	2.6
4	Redmond RFD-0159	250	5	2.5
5	Kitfort KT-1903	250	5	2.6
6	Masterica	150	5	2.8
7	ROTOR SH-002	520	5	2.8
8	Laboratory sample	240	5	2.2

The specific energy consumption calculation for evaporation of 1 kg from the dried product is carried out according to the following formula W_i:

$$W_\mathrm{i} = \frac{W_\mathrm{agg}}{(m_1 - m_2)}, \tag{1}$$

where W_agg is the aggregate electricity consumption for the entire drying period, kW·h/kg; m_1 is the initial mass of the wet product, kg; and m_2 is the final mass of the dried product, kg.

Calculated data were reported for series-produced samples No. 2–6 (see Table 4), while for samples No. 1 and No. 7, experimental data obtained in conditions identical to those of laboratory sample No. 8 are presented.

The readings of the studied values were taken at least five times. A sample of experimental data was used as the resulting values for further statistical processing, and their arithmetic averages were taken. Also, as a result of statistical processing, random variables, lower and upper confidence limits were found, forming a confidence interval.

In the final drying mode, after 5 h of basic drying (see Figure 5), pallets with apples were weighed every 20–30 min on laboratory analytical balances. When the moisture content normative in apples was reached (20–22%) the drying process was completed [35].

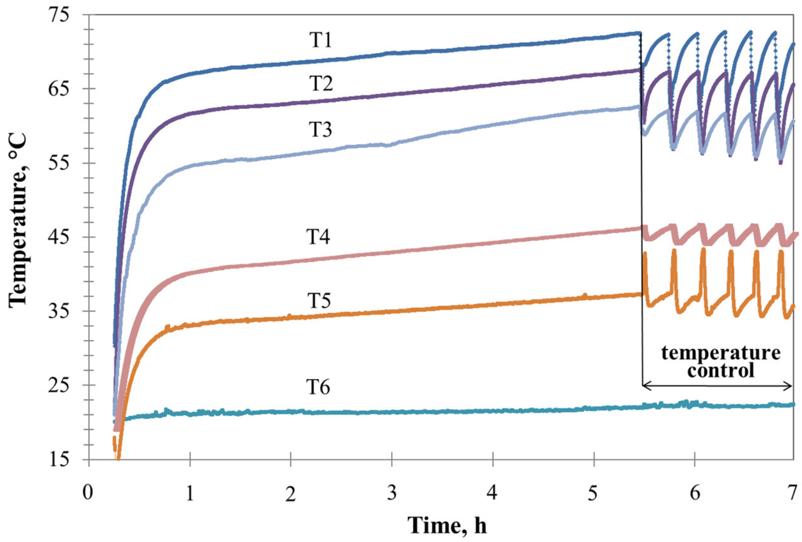


Figure 5. Drying process temperature diagram for characteristic points of the thermoelectric drying unit: T1—hot radiator surface; T2—air supplied into the drying chamber after heating; T3—air in the drying chamber; T4—air leaving the drying chamber; T5—exhaust air removed from the drying unit; and T6—ambient air.

Figure 5 shows drying process temperature diagrams for measuring points of the newly designed laboratory sample of the thermoelectric drying unit.

Figure 6 shows the change in drying agent relative humidity depending on time, in the intake and in the output of the drying unit. Sharp growth of the air relative humidity in the output from the drying unit ($\varphi_{\mathrm{a}2}$) in the initial drying process period can be attributed to intensive water evaporation from the surface of the product under processing.

In Figure 5, cyclic change in the air temperature (checkpoints T1–T6) and that in the air relative humidity ($\varphi_{\mathrm{a}2}$), in the output from the drying unit, has to be associated with the operation of the air temperature and relative humidity metering and control module integrated into the drying unit (see Figure 3b).

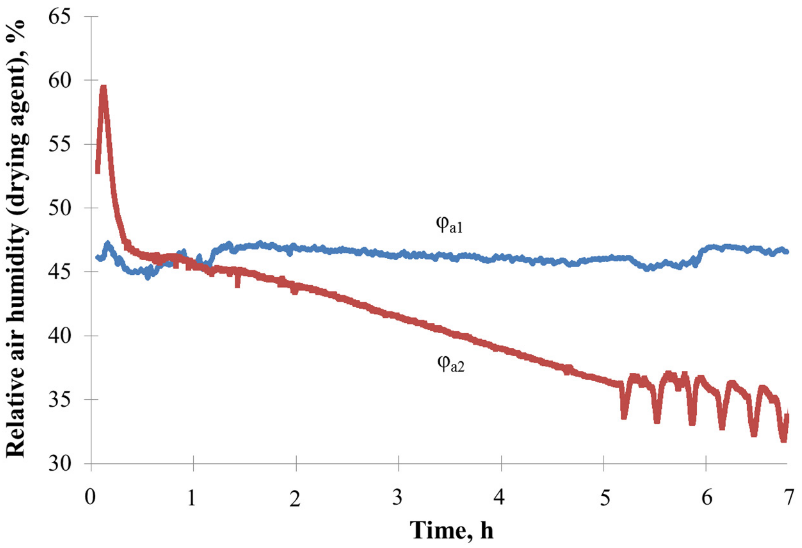


Figure 6. Change in the drying agent's (air) relative humidity during the product drying process, in the intake (φ_{a1}) and in the output (φ_{a2}) of the drying unit.

3.2. Calculating the Thermal Flow Q_c Supplied to the Cold-Junction Radiator of the Thermoelectric Heat Pump

In order to calculate the energy efficiency of the thermoelectric heat pump application, let us define the amount of heat Q_c absorbed by the cold-junction radiator of the thermoelectric heat pump. This heat is extracted from the exhaust drying agent in the drying unit. The amount of heat was estimated by both theoretical calculations and in experiments with the application of statistical repeating procedures and data reduction processes.

Theoretical calculations were based on determining the thermal EMF of the thermoelectric module cold circuit.

The thermal EMF value ($\varepsilon_{T,c}$) for the thermoelectric module cold junctions can be calculated from the following equation [37]:

$$\varepsilon_{T,c} = E t_c N_j, \tag{2}$$

where $\varepsilon_{T,c}$ is the thermal EMF of the module cold side (V); E is the thermal EMF coefficient (Seebeck coefficient), for the thermoelectric junction, equal to 96 ($\mu V \cdot K^{-1}$); N_j is the number of junctions in a module (one 50 × 50 mm Peltier module comprises 228 junctions); and t_c is the average temperature value of the cold-junction radiator of the thermoelectric heat pump, equal to 38 °C (this value was determined experimentally on the drying unit laboratory sample).

By substituting numerical values, we obtain $\varepsilon_{T,c}$ = 6.8 V. Based on the results of the performed experiments on the product subject to drying, it was found out that the average current value of the thermoelectric module was 8.2 ± 0.4 A. Wherefrom the theoretical thermal energy flow Q_c absorbed by the cold circuit of the thermoelectric pump from the exhaust drying agent is 55.8 W.

The following equation was applied to determine the experimental value of the heat flow rate Q_c supplied to the cold-junction radiator of the thermoelectric heat pump with the account of the above-mentioned parameter values:

$$Q_c = c_a L_a \Delta t, \tag{3}$$

where c_a is air specific heat capacity (assumed to be equal to 1.3 $kJ \cdot m^{-3} \cdot {}^\circ C^{-1}$); L_a is air flow rate (m^3/s); and Δt is drying agent (air) temperature difference in the input and in the output of the cold-circuit radiator of the thermoelectric assembly (°C).

The airflow rate of the exhaust drying agent in the outlet duct (F_{oc} = 0.00232 m^2) was measured in the course of the drying process with the use of the thermal anemometer TKA-RKM60 equipped with an external probe ($w_{a,oc}$ = 2.2–2.5 m/s).

For wet air specific heat capacity value 1.3 $kJ \cdot m^{-3} \cdot {}^\circ C^{-1}$, average heat flow rate Q_c absorbed from the exhaust drying agent and supplied to the cold-junction radiator of the thermoelectric heat pump is 52.8 W, for the average value of temperature drop between points T4 and T5 equal to 7.3 °C in the drying process (see Figure 5), which corresponds to ~20% of the aggregate power (~240 W) consumed by the thermoelectric drying unit from the electric grid (see Table 5).

Table 5. Specifications of the thermoelectric drying unit.

Parameter	**Unit**	**Indicator Values**	
		Theoretical Values	**Test Data**
Max. power consumed from the electrical grid	W	240	237
Heat returned by the unit, at least	W	55.8	52.8
Input voltage	V	220	219
Input voltage of one thermoelectric Peltier element	V	20	19
Number of thermoelectric Peltier elements	pc	1	1
Material (product) volume to be dried	l	10	9.6
Drying agent (air) max. temperature	°C	70	67
Average product (apple fruits) processing time for initial and final moisture content of 85% and ~22%, respectively	h	8	7.5
Overall unit dimensions	m	1.0 × 0.7 × 0.06	1.0 × 0.7 × 0.06
Max. unit weight	kg	7	6.7

The technical parameters of the thermoelectric drying unit laboratory sample are presented in Table 5.

Values of heat absorbed by the cold junction of the thermoelectric heat pump obtained by calculations are in rather good correlation with those measured in experiments, which can be considered a proof of the heat pump operating mode of the thermoelectric assembly. A laboratory sample of the newly designed thermoelectric drying unit features a smoother product drying process owing to the presence of side air channels and effective drying agent path organization in the process chamber (see Figure 2). This conclusion is supported by the results of the carried out tests (see Table 3). In the case of drying various products in conventional series-produced household convective-type dryers (see Table 5), one has to swap periodically locations of pallets in the upper and lower parts of the unit in order to maintain a more homogeneous product drying process. Thermoelectric dryers for vegetables and fruits have to comply with the requirements of particular drying process technology, making it possible to set drying agent temperature modes within its optimal range.

3.3. Calculating the Heat Balance of the Thermoelectric Drying Unit

The equation for the internal heat balance of the drying unit under study has the following form:

$$q_c + q_e = q_m + q_h + kq_a - c_{liq} \cdot \theta_1, \tag{4}$$

where q_c is the specific amount of heat absorbed from the exhausted drying agent that is further supplied to the cold-junction radiator of the thermoelectric module (kJ per 1 kg water); q_e is the specific amount of heat produced by electric current in the thermoelectric module (kJ per 1 kg water); c_{liq} is the specific heat capacity of the fluid component within a wet product under processing for initial product temperature (supposed to be equal to that of environment) θ_1 ($kJ \cdot kg^{-1} \cdot {}^\circ C^{-1}$) in the dryer input; q_m is the specific heat consumption in the dryer, used for heating a product under processing (kJ per 1 kg water); q_h is the specific heat emission from the drying unit casing surface into the environment (kJ per 1 kg water);

q_a is the specific heat loss with the exhaust drying agent (air) (kJ per 1 kg water); and k is the coefficient taking account of the share of heat absorbed by the cold-junction radiator of the thermoelectric module from the exhaust drying agent (equals to 0.80–0.88).

The aggregate value of the specific heat q_c absorbed from the exhaust drying agent and supplied to the cold-junction radiator of the thermoelectric module, along with the specific amount of heat q_e produced by electric current in the thermoelectric module, gives the total value q_{hot} of specific heat absorbed by a drying agent from the hot-junction radiator of the thermoelectric heat pump.

Specific heat flow into the drying unit consumed for heating a product under processing can be defined from the following expression:

$$q_m = \frac{G_2 \cdot c_m \cdot (\theta_2 - \theta_1)}{W} + c_{liq} \cdot (\theta_2 - \theta_1), \quad (5)$$

where G_2 is drying unit productivity expressed in final (dried) product weight (kg/s); c_m is specific heat capacity of dried product (kJ·kg^{-1}·°C^{-1}), we assume that the average value c_m = 3.88 kJ·kg^{-1}·°C^{-1}, for apple fruits; W is amount of water extracted from a product under processing (kg/s); and θ_2 is product temperature in the output from the dryer (°C).

Therefore, the drying unit performance in terms of dried product outcome equals to the following:

$$G_2 = G_1 - W = G_1 - \frac{\varphi_1 - \varphi_2}{1 - \varphi_2} = G_1 \frac{1 - \varphi_1}{1 - \varphi_2}, \quad (6)$$

where G_1 is drying unit performance measured in initial (wet) product weight (kg/s), and φ_1 and φ_2 are values of moisture content (%) in the initial product and in that that has passed the dying procedure, respectively.

The amount of water W extracted from the material under processing can be defined from the following expression:

$$W = G_1 \frac{\varphi_1 - \varphi_2}{1 - \varphi_2} = G_2 \frac{\varphi_1 - \varphi_2}{1 - \varphi_1}. \quad (7)$$

We assume the product temperature in the output from the dryer θ_2 to be equal to that of the wet-bulb thermometer t_{2w}. In a theoretical approach, the drying process is assumed to comply with the adiabatic relationship, in which case t_{2w} can be determined from the *i-d* diagram for wet air conditions and for specified initial drying agent parameters (temperature t_1 and moisture content d_1).

Dried product-specific heat capacity can be calculated using the following formula:

$$c_m = \frac{c_d \cdot 100 + c_{liq} \cdot \varphi_2}{100 + \varphi_2}, \quad (8)$$

where c_d is specific heat capacity of moisture-free material (KJ·kg^{-1}·°C^{-1}), and φ_2 is final moisture content in product after the drying process completion (%).

Specific heat loss q_h from the dryer casing surface into the environment is assumed to be in the range from 125 kJ to 420 kJ per 1 kg of evaporated water, depending on the moisture content of the particular material (for apple fruits, the lower value has to be selected). Alternatively, the value of this parameter can be selected in the range from 10% to 12% of specific heat q_m absorbed in the drying unit for heating a product under processing [38]. Specific heat emission from the dryer casing surface into the environment can also be determined with the help of expression, provided that the value of coefficient K_h for heat transfer through the casing walls has been defined [28].

Specific heat loss with the exhaust drying agent (air) q_B is determined from the following formula:

$$q_a = \frac{G_a c_a (t_2 - t_0)}{W}, \quad (9)$$

where G_{a} is the drying agent flow rate during the drying process (kg/s); c_{a} is the drying agent's (air) specific heat capacity (kJ·kg^{-1}·°C^{-1}); t_2 is the exhaust drying agent temperature (°C); and t_0 is the ambient air temperature (°C).

The drying agent's flow rate in the process of drying equals to the following:

$$G_{\mathrm{a}} = \frac{W}{d_2 - d_1}, \tag{10}$$

where $d_1 = d_0$ is air moisture content (kg/kg) in the dryer input (air is heated in the dryer as it passes through the hot-junction radiator of the thermoelectric assembly in conditions of constant moisture content d = const), and d_2 is moisture content of air in the dryer output (kg/kg).

3.4. Calculating the Hot-Junction Radiator Design Parameters of the Thermoelectric Drying Unit Heat Pump

Calculations were carried out with the purpose of defining the design parameters of an air radiator (see Figure 7), making it possible to transfer heat from the hot side of the thermoelectric module to the drying agent (air) with minimal loss.

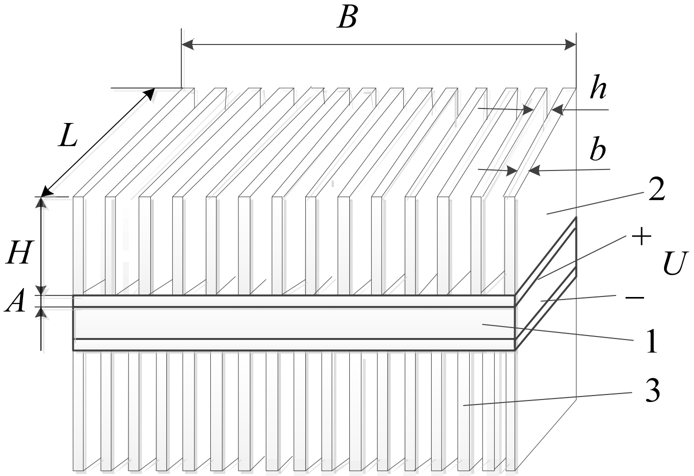


Figure 7. Diagram of an air-to-air thermoelectric heat pump: 1—Peltier thermoelectric element; 2—hot-circuit air radiator fins; and 3—cold-circuit air radiator fins; A is radiator base thickness; B is air radiator size in the direction perpendicular to the fin plane; b is the fin thickness; H is the fin height; h is the distance between the adjacent fins; L is air radiator size in the line of the fin plane.

The following initial design parameters were defined while calculating air radiator (see Figure 7) design parameters: ΣQ is the thermal performance of the thermoelectric assembly (W); λ_{r} is the radiator material thermal conductivity coefficient (W·m^{-1}·°C^{-1}); λ_{a} is the air thermal conductivity coefficient (W·m^{-1}·°C^{-1}) for specified temperature values.

We considered an option with forced radiator air cooling. The value of drying agent (air) velocity w_{a} in the space between the air radiator fins was assumed to be from 2 m/s to 4 m/s, based on the typical parameters of similar unit design options [39].

In order to define heat transfer coefficient α_{r}, one has to select appropriate values of air velocity w_{a} and of kinematic viscosity ν_{a} depending on the temperature of air flowing about the radiator fins. Reynolds number Re has to be determined as well. Depending on the Reynolds number calculated value Re (whether laminar or turbulent air flow mode takes place), Nusselt number Nu is calculated with the use of a formula (either 11 or 12) [40].

For a laminar (air) flow mode, we have the following:

$$\mathrm{Nu_L} = 0.57\mathrm{Re_L^{0.5}}. \tag{11}$$

For a turbulent (air) flow mode, the following expression is valid:

$$\mathrm{Nu_L} = 0.032\mathrm{Re}_\mathrm{L}^{0.8}. \tag{12}$$

For a given velocity of air flowing about the radiator fins, the convective heat transfer exchange coefficient α_r can be calculated with the use of Reynolds (Re) and Nusselt (Nu) numbers as follows:

$$\alpha_\mathrm{r} = \frac{\mathrm{Nu}\lambda_\mathrm{a}}{L}. \tag{13}$$

The air radiator overall dimensions can be defined from the following expression, insuring the most reasonable relationship between fin thickness b and its height H when the maximum heat exchange is achieved for an acceptable specific quantity of metal required for radiator fin fabrication [39]:

$$H = 1.419\frac{b}{2}\sqrt{\frac{2\lambda_\mathrm{r}}{\alpha_\mathrm{r} b}}. \tag{14}$$

The area of a single radiator fin $F_\mathrm{r}{}'$ supplying thermal energy to a drying agent is as follows:

$$F_\mathrm{r}{}' = 2HL. \tag{15}$$

Drying agent contacts with the radiator, whose total heat exchange surface area F_r can be deduced from the following heat flow rate equation:

$$\sum Q = K_\mathrm{t} F_\mathrm{r} \Delta t_\mathrm{a}, \tag{16}$$

where ΣQ is heat flow rate of the thermoelectric assembly (heat pump) hot circuit (W) (normally defined based on particular type of thermoelectric module specifications); F_r is the surface area of all radiator fins (m^2); $\Delta t_\mathrm{a} = t_\mathrm{a1} - t_\mathrm{a2}$ is the specified temperature difference for air in radiator input and output (°C); and $K_\mathrm{t} = \frac{1}{1/\alpha_\mathrm{r} + A/\lambda_\mathrm{r}}$ is heat exchange rate for heat transfer from the thermoelectric module surface to a drying agent (W·m^{-2}·°C^{-1}).

The number of radiator fins N is calculated as follows:

$$N = \frac{F_\mathrm{r}}{F_\mathrm{r}{}'}. \tag{17}$$

With the account of the selected value of drying agent velocity w_a, the cross-section f_c of the air channel in which the thermoelectric module air radiator has to be installed can be defined as follows:

$$f_\mathrm{c} = \frac{G_\mathrm{a}}{w_\mathrm{a}} \tag{18}$$

4. Conclusions

In accordance with the newly designed block diagram, a laboratory sample of a thermoelectric drying unit has been manufactured for studying the drying process parameters. Laboratory studies of the apple fruit drying process have been carried out in order to evaluate specific power consumption for evaporating 1 kg of water in the newly designed convective-type thermoelectric drying unit. The values of this indicator were compared with those of some series-produced household dryers.

The drying process was performed with partial recirculation and heat recovery of the exhaust drying agent heat. The diagrams for temperature change in the reference points and for the drying agent relative humidity in both the input and the output of the thermoelectric drying unit have been presented, making it possible to characterize the drying process in its dynamic development. The properties of apple fruits before and after processing in the thermoelectric drying unit have been reported along with those of ones dried with the use of conventional series-produced convective-type dryers.

It has been found out that application of a thermoelectric heat pump ensuring exhaust agent heat recovery makes it possible to reduce both the capacity of the drying agent heater and the electric power consumption in the course of drying in the newly designed convective-type thermoelectric drying unit by 20% compared to conventional series-produced household dryers.

A laboratory sample of the thermoelectric drying unit ensures more homogeneous drying product process conditions owing to the presence of side air channels and a more effective drying agent path structure in the processing chamber compared to conventional convective-type dryers. This conclusion was confirmed by the results of the carried out studies.

In the newly designed thermoelectric unit, the difference in the amount of evaporated water in different pallets did not exceed 2.5% by the end of the apple fruit drying process. For conventional convective-type dryers, the value of this parameter is more than 12.5%.

Mathematical expressions for calculating the dimensions of the hot-junction radiator of the thermoelectric heat pump (type “air-to-air”) have been presented that form the core of substantiating basic parameters of the thermoelectric assemblies of various performance for drying units of the proposed design.

Further research will be focused on optimizing the operating modes of convective-type thermoelectric drying units with the account of current values of moisture content in a product under processing, amount of recirculation air, and amount of exhaust air from the drying unit and sent for recovery to a thermoelectric heat pump, as well as the amount of mixed-in outdoor air in during the drying process to automatically regulate the supply air intake damper position.

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