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Analysis of the Design and Technological Parameters of the Designed Solar Dryer with a Heat Pump

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ABSTRACT

The article describes design of a solar dryer with a heat pump, which is used to increase heat output of the unit twice to solve the problem of using environmentally clean sources of thermal energy for fruit drying. The authors of the research have developed methodology of substantiation of the solar dryer parameters, which is applied to design and justify the optimal technological modes and parameters of the heat carrier in the unit, to describe the heat transfer characteristics of the unit operation, to assess the impact of physical environmental parameters on technological indicators of the process. The novelty described in the article is developed design of a solar dryer with a heat pump and to substantiate its constructive and technological structure. The work supplies scientifically substantiated methodic recommendations on composing and forecasting a parametric series of solar dryers for the conditions of private households and farms, and grounds their design parameters. Designed solar dryer with a heat pump affects solving environmental problems of power engineering due to substitution of the electric and thermal energy by the one obtained from solar energy, and mitigation of social problems by creating new job places needed when producing, installing and exploiting such units. The obtained results can be used for designing and improving technical aspects of fruit drying, to increase technological and energy efficiency of the process.

Keywords: solar energy, solar fruit drying, heat pump, heat accumulator, drying chamber.

INTRODUCTION

One of the elements of a country's food security is to create reserves of dry foodstuff, for instance dried fruit and dried vegetables as they are components of everyday meals and used in production of semi-finished products because of their high nutritional value and resistance to spoilage. They are produced by using dryers of different types with traditional sources of heat in the form of electric energy, or heat obtained from fuel combustion. By using solar air collectors or heat pumps, which operate together with dryers, it is possible to solve the problem of dependence on central power supply, reduce the level of energy resource consumption, etc.

The mentioned issue is studied in the works of Ozarkiv M.I., which deal with peculiarities of designing a solar drying equipment for timber, including by applying computer modeling [1]. These researches are fundamentals for development of the methodology of designing solar dryers, but are not relevant for the current climatology of Ukraine.

A principal share of researches in the field is conducted by foreign scientists. In particular, Abubakar et al. [2] presented new developed designs of mixed-mode solar dryers for drying foodstuff without heat accumulator and investigates the process of convective drying. Mehta et al. [3] substantiated design and analyzes efficiency of a mixed mode-tent type solar dryer for fish drying in coastal areas and suggests a numerical method of solving the mathematic model of stress calculation and moisture distribution during the drying as well as shapes a problem of the model optimization. Moreover, it proposes engineering scientific and methodic principles of determining an optimal volume of a drying chamber, increase of the heat carrier temperature in the dryer due to additional radiation of the drying chamber that is based on the methodology of designing infra-red dryers. Yagnesh et al. [4] supplied a substantial review of solar dryers and analyzes the process of convective fruit drying for different heat and moisture modes of drying.

Berville et al. [5] presented a short analysis of the recent findings in the technologies of solar drying, substantiates rational constructions, an optimal design and technological parameters of dryers. Raj et al. [6] justify the cost-effective method of improving the efficiency of solar air heater due to discrete macro-encapsulated PCM capsules for drying foodstuff. Bandara et al. [7], developed the engineering methodology of calculating the required size of a solar collector and assessed the possibility of the energy efficient use of unglazed solar collector for drying purposes as compared to the glazed type of collector.

Seerangurayar et al. [8] analyzed peculiarities of the process of drying dates, proposes a new design of a solar dryer and substantiates reasonability of using the simulating modeling of the process and its program implementation. Vengsungnle at al. [9] supplied results of the research of thermal performance of a photovoltaic–ventilated mixed mode greenhouse solar dryer with automatic closed loop control for Ganoderma drying. Sharma et al. [10] made review of the effect of natural and forced convection solar dryers in retention of proximate nutrients in tomato and quality of the dried materials during the drying process. The authors [6, 7, 8, 9, 10] conducted scientific researches aimed to develop and solve the applied scientific problem of substantiating the parameters and operation modes of a solar thermal unit to increase the autonomous energy efficiency of the technological process of drying foodstuff in the conditions of private households and farms as well as wet clothes and shoes of military men on the remote objects of various use. The researchers developed prototypes and experimental models of the units of transforming solar energy, particularly solar combined mobile energy module of the “source (generator) of heat energy – intermediate environment - object to be dried” scheme. The experimental models are developed to study efficiency of the units of transforming the energy of renewable sources, particularly a model of air collector; a model of solar thermal unit; a model of heat accumulator using gravel. The researchers are currently in the process of developing an intelligence system of control for operation modes of the solar thermal units on the basis of fuzzy logic algebra and control for moisture allocation, moisture removal and ventilation in the dryer both with the autonomous and mains power supply [11, 12].

The above-described solar thermal units (solar dryers) for drying foodstuff were, however, designed for the countries with hot subtropical, tropical, subequatorial, equatorial climate, like Azerbaijan, Kazakhstan, India, Taiwan, the USA, etc. Those prototypes of dryers do not need additional sources of thermal energy like a heat collector or a heat pump.

Analysis of the literary sources on the studied topic proves that the main works deal with the solar thermal units equipped with additional unit, like a solar air collector and accumulating system using gravel or a heat pump. In some publications, such units are supplemented with external power grid or a photovoltaic panel. Those works do not consider application of solar energy and heat energy units, particularly air collectors and heat accumulators using gravel which are characterized by a high level of primary solar energy transformation, for the moderate continental climate of Ukraine.

Fitting a solar thermal unit with air and photovoltaic elements and accumulation system for transformation of the radiant energy into thermal and electric one enables development of an energy unit of the complex energy supply to the remote and autonomous objects characterized by high efficiency of primary energy transformation and energy supply reliability [13].

When designing local autonomous systems of heat supply to the remote objects, there is a problem of their low efficiency due to a relatively low efficiency of the accumulation system that can be minimized by optimizing the amount of supplied energy. Using gravel heat collector in the solar thermal units reduces, however, quality of the dried products because of the excessive production of moisture in the early period due to dew point on the surface of accumulating material [14–16]. Therefore, the authors of the present research propose to use a heat pump instead of a heat accumulator using gravel.

Thus, to solve the outlined problems, it is necessary to develop a series of solar energy units for drying fruit and vegetables, which have the appropriate structure of technical and energy tools including solar collectors, drying chambers, systems of heat distribution, means of thermal energy accumulation, heat pumps, which can reduce consumption of energy resources in the process of dry food production to improve their autonomy.

Implementation of the main idea necessitates development, substantiation and optimization of the structure and parameters of solar energy units equipped with heat pumps. It will cease dependence on the central energy supply as well as increase the energy supply level. To conduct the research, it was needed to design an experimental stand of a solar thermal unit with a heat pump, which would be used to carry out experimental studies on optimization of its structure and parameters. Thus, the goal of the present research is to increase efficiency of the technological process of drying fruit and vegetables based on the developed design and to substantiate the constructive and technological parameters of a solar dryer that will reduce consumption of energy resources.

According to the set goal, the following tasks should be fulfilled:

- to develop design of a solar dryer with a heat pump and to substantiate its constructive and technological structure;
- to develop methodology of substantiating the rational constructive and technological parameters of the solar dryer;
- to justify the parametric series of solar dryers with a heat pump.

MATERIALS AND METHODS

Materials and methods of substantiating the constructive and technological parameters of a solar dryer with a heat pump

Description of a production solar dryer with a heat pump

General view of the proposed solar drying unit with a heat pump is shown at the Figure 1. The main elements of the unit are: body (2), drying chamber (1), framework made of optically transparent material, compressor (3), air condenser (4), air cooler (5), blackened flat evaporator (6), installed in a solar collector (7), directed southward at the angle 25−40° to the horizon. The evaporator (5, 6) is connected in parallel to the evaporator (8) and the heat evaporator-accumulator (8 evaporator-accumulator). A duct (9)

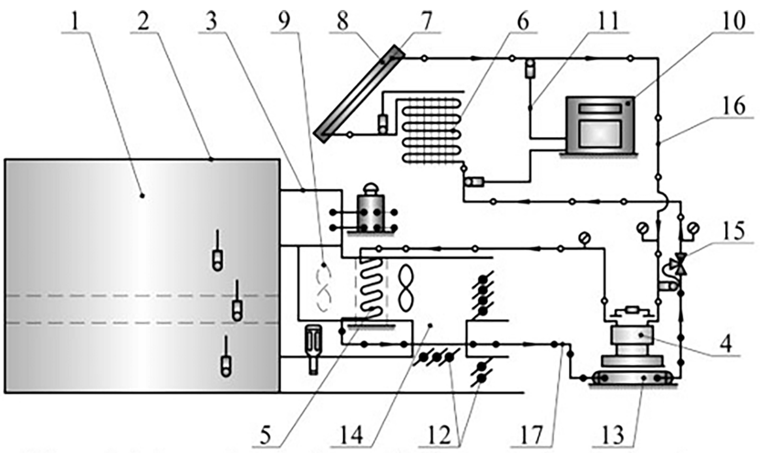

Figure 1. Scheme of a solar dryer with a heat pump: 1 – chamber; 2 – transparent body; 3 – compressor; 4 – condenser; 5, 6 – evaporator; 7 – solar collector; 8 – evaporator and evaporator-accumulator; 9, 12, 21, 26 – ducts; 10, 13, 27 – gates; 11 – ventilator; 14, 15, 16, 17 – pipelines

has a gate (10), ventilator (11) and serves for air inlet to the chamber (1). After the evaporator-accumulator (8), a circulating duct (12) is set with a gate (13). The compressor (3) is connected with the air condenser (4) by the pipeline (14), with the evaporator (5) by the pipeline (15) and with the evaporator-accumulator (8) by the pipeline (16).

The condenser (4) is connected with the evaporator (5) by the pipeline (17) through the valve (18) and with the evaporator-accumulator (8) by the pipeline (19) through the valve (20). The air condenser (4) is connected with the chamber (1) by the duct (21) through the vent (22) in the end wall (23) of the chamber (1). In the chamber (1), a mesh conveyor (24) is installed to move products during the drying process. At the end of the unloading part of the conveyor (24), a baffle (25), which periodically adjoins its upper part, is installed.

The main advantages of a solar dryer with a heat pump are their high efficiency coefficient and the ability to operate the heat pump at lower temperatures. This dryer is also characterized by lower energy consumption, which is achieved thanks to the high coefficient of performance of the heat pump and the high thermal efficiency of the dryer. In such dryers, the drying air passes through the drying chamber and extracts moisture from the product being dried. The moist air from the dryer is transported through the heat pump evaporator, which acts as a dehumidifier. The evaporator-accumulator (8) is a container with boxes to support nozzles with a gravel layer. In the nozzle layer, an evaporator in the form of a tube heat exchanger is installed.

The drying unit operates in the following way. Products are put into the chamber (1) during daytime and dried under the effect of solar radiation coming through the transparent body (2) and air with the temperature of 55−60 °C penetrating through a thick layer of products. At the same time, the ventilator (11) supplies the outside air to the duct (9) through the condenser (4) of the heat pump (3), where it is heated to the temperature of 55−60 °C and enters the chamber (1) from the duct (21). The heat from the exhaust air is accumulated in the gravel evaporator-accumulator (8), and the cooled air is transferred by the duct (12) into the atmosphere. If the air temperature is higher than the temperature of the environment $t_h > t_e$, it is partially or completely recirculated by means of the gate (13) depending on the air humidity.

During daytime, the heat pump (3) operates in a close circuit (Figure 2, a): the air condenser (4) is connected with the air cooler (5) and the evaporator installed in the solar collector (6) by the pipeline (17) through the valve (18). The temperature of absorption of the working mass *R−410a* is equal to t_a = 45−50 °C, whilst the temperature of condensation is t_k = 70 °C, t_o = 16 ÷ 20 °C.

During night-time (Fig. 2, b) or overcast weather, the air in the condenser (4) is heated by means of the heat accumulated in gravel and the heat of exhaust air. In case the temperature after throttling (of the valve) is lower than the temperature of the environment $t_o < t_e$, both valves (18, 20) are open, and if $t_o > t_e$, the valve (18) is closed and the valve (20) is open. The exhaust air gives a share of its heat away when passing the evaporator. Afterwards its temperature reaches the temperature of dew and the air is getting dry. Then, it moves to the duct (9) through the recirculating duct (12) by means of the gate (13).

In the process of fruit drying, the volume of products is getting 4–5 times reduced, and because the drying process takes 2–3 days, after the first day, it is necessary to mix and compact the products at the half length of the conveyor (24) when the baffle (25) is in tight contact with the net. At that time, new portions of products are being loaded into the appropriate part of the dryer.

METHODOLOGY OF THE ENGINEERING CALCULATIONS OF A SOLAR DRYER WITH A HEAT PUMP

Basing on the conventional methodology of designing a thermo-radiation drying unit, the authors of the research has composed the energy balance equation for a time period, c:

- daytime:

$$dQ_a + dQ_c = dQ_w + dQ_m + dQ_{aa} + dQ_e + \sum dQ_h \tag{1}$$

- night-time:

$$dQ_c = \pm dQ_w + dQ_m + dQ_e + \sum dQ_h \tag{2}$$

where: dQ_a – energy of solar radiation absorbed by the product amount, kJ; dQ_c – energy (heat) which enters the chamber with the heated air, kJ; dQ_w – energy (heat) spent for product warming, kJ; dQ_m – heat spent for moisture evaporation, kJ; dQ_{aa} – heat accumulated in the accumulator, kJ; dQ_e – heat which comes out of the drying chamber with the exhaust air, kJ; $\sum dQ_h$ – heat lost in the environment, kJ.

Calculations of separate terms of the balance equation are done. Energy of solar radiation absorbed by the product amount:

$$dQ_a = K_p \cdot K_r \cdot \bar{A}_a \cdot q_{s,\tau} \cdot F_a \cdot d\tau \tag{3}$$

where: K_r – coefficient of multiple reflection of the solar dryer chamber, $K_r = \dfrac{\bar{T}_{кор}}{1 - \bar{R}_{пр} \cdot \bar{R}_{кор}}$;

$\bar{R}_p, \bar{R}_s$ – average passing capacity of the product and the solar dryer body; $\bar{T}_a$ – average passing capacity of the body; $\bar{A}_a$ – average absorption capacity of the product; K_p – coefficient of pollution of the drying unit body, $K_p = 0.95$; $q_{s,\tau}$ – solar radiation energy with consideration of air transparence during a time period, kW/m^2 кВт/м^2, F_a – radiated surface, m^2.

Energy obtained from the heated air in the chamber:

$$dQ_c = F \cdot \upsilon_a \cdot \rho_a \cdot c_a (t_{a2} - t_{a1}) \cdot d\tau \tag{4}$$

Energy spent for product heating:

$$dQ_w = h_{сл} \cdot \rho_p \cdot F \cdot C_p (\Theta_2 - \Theta_1) \tag{5}$$

Energy spent for moisture evaporation:

$$dQ_m = q_m \cdot r' \cdot F \cdot d\tau \tag{6}$$

Energy discharged (or supplied) from the accumulator:

$$dQ_{aa} = \pm V_{aa} \cdot \rho_{aa} \cdot C_{aa} (t_{aa2} - t_{aa1}) d\tau \tag{7}$$

Energy disposed by the exhaust heat carrier (air) into the environment:

$$dQ_e = (1 - K_r) \cdot \rho_{ya} \cdot C_{ya} \cdot \upsilon_{ya} \cdot F (t_e - t_a) d\tau \tag{8}$$

where: υ_a – air velocity, m/sec; $\rho_a, \rho_p, \rho_{aa}$ – density of air, product, accumulating substance respectively, kg/m^3; C_a, C_p, C_{aa} – specific heat capacity of the product and accumulating material, kJ/kg·K; q_m – average density of the moisture flow, kg/m^2·sec ; r' – average specific heat of moisture evaporation, kJ/kg.ms; K_r – coefficient of air recirculation.

Heat losses through the transparent double-layer body of the drying unit are calculated by the known formula:

$$dQ_{втр.1} = K \cdot F (\Delta t_d - t_e) d\tau \tag{9}$$

where: K – general coefficient of heat transfer through the transparent body of the dryer, W/m·K. The dryer body is divided into two parts depending on their wall position (vertical or horizontal) and for each part K [17] is calculated:

$$K = \frac{1}{\frac{1}{\alpha_1} + \sum \frac{\delta_b}{\lambda_t} + \frac{1}{\alpha_2}} \tag{10}$$

where: α_1 − coefficient of heat transfer from the drying chamber to the internal surface of the transparent body, W/m^2·K; α_2 − coefficient of heat transfer from the external body to the environment, W/m^2·K; δ_l, δ_b − thickness of some layers composing the unit body, mm; δ_w − thickness of a wall of the internal transparent body of the unit, δ_w = 0.1 mm; δ_{we} − thickness of a wall of the external transparent body of the unit, δ_{we} = 0.07 mm; δ_a − thickness of air between the double-layer transparent body, δ_a = 25 mm; λ_a, λ_t, − corresponding heat transfer factors; for PE film, λ_t = 0.266 W/m·K; for air when t_a = 50 °C, λ_a = 0.027 W/m·K.

In the drying chamber, air moves under the effect of a ventilator, and thus, heat transfer from the air of the drying chamber to the internal body is supplied mainly due to forced convection. The heat transfer between two transparent bodies is provided by free convection that is caused by the difference in density of the corresponding layers of air. The coefficient of convective heat transfer from the top of the external transparent body to the environment is measured by wind speed above the drying unit. Therefore, calculation of the coefficient of losses through the transparent isolation is rather time-consuming. To simplify the calculation, the authors of the present research used the data provided in the work [11], particularly the heat transfer coefficient α_1 =17.1 W/m^2·K at the drying agent speed of 4.5 m/s in the chamber, the heat transfer coefficient α_2 = 12.2 W/m^2·K at the wind speed of 3.2 m/s (wind speed of 3.2 m/sec is a characteristic of the area of Korets town in Rivne region in the period of fruit drying). In solar dryers, the value Q_c is maximum in the period from 2^{00} to 6^{00}, when the heat of the condensation is used, i.e.:

$$dQ_c = dQ_{hp} \tag{11}$$

$$dQ_{hp} = Q_{pc} = L_a \cdot C_a (t_{a2} - t_{a1}) \tag{12}$$

where: L_a − massive air loss, kg/sec; t_{a1} − air temperature after partial circulation of the exhaust air before the air condenser of the heat pump; Q_{hp} − heat output of the heat pump, kW; Q_{pc} − heat taken by air from the heat pump condenser, kW; t_{a2} − temperature of the air coming into the chamber after the condenser.

Specific heat output of the heat pump $q_{hp}^{/}$ referred to 1m^2 of the solar drying unit can be expressed by the formula [18].

$$q_{hp}^{/} = \frac{Q_{hp}}{F} = \frac{M(q_o + l)}{F} \tag{13}$$

where: M_{wm} − losses of the working mass, kg/sec; q_o − specific energy absorbed by the working mass during evaporation, kJ/kg; l − specific energy obtained at compression of the working mass in the compressor, kJ/kg; F − area of the solar dryer, m.

Measuring the specific energy absorbed in the evaporator by the working mass depends on many factors making the task rather complicated. Therefore, the thermal technical calculations for the chosen working mass R−410a are made according to the experimental data in the conditions of Ukraine.
By using the accumulated heat in night-time and solar energy during daytime, temperature of the working mass R−410a can be maintained, under its similar consumption, at the exit from evaporator t_a = 43 − 48 °C, while at the entry to evaporator, the temperature of R-410a is equal to t_o = 18 − 20 °C and the condensation temperature is t_k = 70 °C.

The Table 13 presents thermodynamic parameters of the working mass R-410a under the set temperature. The specific amount of heat supplied to the working mass in the evaporator is calculated by the formula [19]:

$$q_0 = i_1 - i_4 \tag{14}$$

where: i_1 − entropy of the working mass at the evaporator exit, kJ/kg; at t_a = 45 °C; i_4 − entropy of the working mass after throttling (at the evaporator entry) at t_0 = 20 °C.

The specific energy obtained when compressing the working mass in the compressor is calculated:

$$l = i_2 - i_1 \tag{15}$$

where: i_2 − entropy of the working mass after compressing, kJ/kg

Thus:

$$q_k = q_0 + l = i_2 - i_4, \tag{16}$$

Consumption of the working mass R−410a per 1m^2 of the solar dryer is calculated:

$$M = \frac{q'_{hp}}{q_k} = \frac{q'_{hp}}{q_0 + l} \text{ kg/m}^2\cdot\text{s} \tag{17}$$

The theoretical specific capacity of the heat pump is calculated by the formula [20]:

$$N_t = M \cdot l \text{ kJ/m}^2\cdot\text{s}. \tag{18}$$

The theoretical coefficient of energy transformation is determined by the formula:

$$\mu_t = \frac{q_k}{l} = \frac{l + q_0}{l} = 1 + \frac{q_0}{l} \tag{19}$$

The heat pump efficiency is calculated:

$$\eta_{hp} = \frac{\mu_a}{\mu_t}$$

where: μ_a − actual COP.

The actual capacity of the heat pump is estimated:

$$N_a = \frac{N_t}{\eta_{hp}} = \frac{q'_{hp} \cdot l}{q_k \cdot \eta_{hp}} \text{ kW/m}^2 \tag{20}$$

The area of heat transfer of the condenser surface is calculated by the formula:

$$F_к = \frac{Q_{pc}}{K_{ht} \cdot \Theta_k} \text{ m}^2 \tag{21}$$

where: K_{ht} − coefficient of heat transfer, W/m^2·K; (K_{ht} = 43÷51 W/m^2·K); Θ_k − average logarithmic difference in temperature, °C.

$$\Theta_k = \frac{t_{a2} - t_{a1}}{\ln \dfrac{t_k - t_{a1}}{t_k - t_{a2}}} \text{ °C} \tag{22}$$

By knowing area of the condenser surface in the heat pump F_k, one can determine the number of sections, the section height, area of the open cross-section of the condenser with the plate fins and air loss from the condenser heat balance by applying the methods described in [21, 22].
The actual heat transferring area of the fin-tubular condenser is calculated:

$$F_{\kappa} = N\left\{2\pi\left[\left(h\cdot в - n_1\frac{\pi d^2}{4}\right) + (h + в)\cdot\delta\right] + n_1\pi d(l - n\delta)\right\} \tag{23}$$

where: N − number of sections in the condenser; n_1 − number of fins and tubes in a section, m; h, $в$, δ − height, width and thickness of a fin, m; l − length of the condenser in the frontal cross-section, m.
Area of the heat transferring surface of the evaporator fitted in the solar collector for conditioning the drying chamber as well as in the accumulator is calculated by the formula:

$$f_i = \frac{Q_z}{K_z(t_{ob} - t_o)}; \quad f_2 = \frac{Q_6}{F_k\cdot Q^*\cdot\eta_{kil}} \tag{24}$$

where: Q_z − heat flow of the evaporator on the corresponding zone, W; K_z − coefficient of heat transfer of the evaporator in the corresponding zone, W/m^2·K. The index i = 1, 2, 3 for the air cooler, evaporator in the solar collector and evaporator in the gravel accumulator; for the air coolers K_z = 12 ÷ 14 W/m^2·K; $\Theta_n = t_{ob} - t_o$ – difference in temperature of the boiling object: for the air cooler Θ_{n1} =10−12 °C, for the solar collector Θ_{n2} = 15 − 20 °C, for the evaporator-accumulator Θ_{n2} = 12 − 17 °C.

Amount of the accumulating agent in the evaporator-accumulator is calculated by using the data provided in the works (6, 11, 114, 121). The calculation is made by using the heat balance equation:

$$q^{/}_{aa} = \left[M_{wm}(i_1 - i_4) + \sum q_l\right]\Delta\tau_2 \tag{25}$$

$$q^{/}_{aa} = \left[\alpha_{ef}\cdot V_{aa}\cdot a(t_{ea} - t_n) - K_{nd}\cdot F_{aw}\left(\bar{t}_{aa} - \bar{t}_k\right)\right]\Delta\tau_1 \tag{26}$$

$$q^{/}_{aa} = \rho_{aa}\cdot C_{aa}\cdot V_{aa}(t_{n2} - t_{n1}) \tag{27}$$

where: $\Delta\tau_1$ − time for charging the evaporator-accumulator; $\Delta\tau_2$ − time for discharging the evaporator-accumulator; i_1, i_4 − entropy of the working mass R-410a at the exit of the evaporator-accumulator and at the entry; M_{wm} − weight of the working mass; V_{aa} − volume of the heat accumulator; α_{ef} − effective coefficient of heat exchange between the nozzle and the heat carrier; a − heat exchanging surface of the nozzle referring to a volume unit; F_{aw} − external surface of the accumulator wall; K_{nd} − coefficient of heat transfer from the nozzle to the drying chamber; t_{n1}, t_{n2} − temperature of the accumulating mass at the beginning and in the end of the charging process; $\bar{t}_{aa}, \bar{t}_k$ − average temperature of the accumulating mass and the drying chamber.

By solving the Equations (25) and (27), the amount of the accumulating agent for the solar dryer is calculated:

$$V_{aa} = \frac{\left[M_{wm}(i_1 - i_4) + \sum q_l\right]\Delta\tau_2}{\rho_{aa}\cdot C_{aa}(t_{n2} - t_{n1})} \tag{28}$$

It is necessary to check duration of the evaporator-accumulator charge during daytime. At the same time, it is assumed that the average temperature of the exhaust air is t_{ea} = 53 °C during the charging, and the heat loss through the walls is ignored. The heat transfer coefficient is calculated by the formula [1]:

- for the range $Re > 200$; $Nu = 0.61\ Re^{0.67}$:

$$\alpha_{ef} = \frac{0{,}61\lambda(\upsilon\cdot d_{\text{к}})^{0{,}67}}{\mu^{0{,}67}\cdot d_2} = 0.109,\ \text{kJ/m}^2\cdot\text{K} \tag{29}$$

where: μ, λ − thickness and heat transfer of the heat carrier; d_2 − gravel diameter.

Then, the formula for calculating the charge duration is:

$$\Delta\tau_1 = \frac{\rho_{aa} \cdot C_{aa}(t_{n2} - t_{n1})}{\alpha_{ef} \cdot a(t_{ea} - t_n)} \quad (30)$$

The formula (1) is used to calculate the needed specific heat output of the heat pump during daytime. If the calculated heat output of the heat pump is higher during night-time than the calculated heat output of the heat pump $Q_{dt} - Q_{nt}$, the formula (30) is used to calculate shift index of the heat pump during a day.

$$B = \frac{Q_{nt} + Q_{dt}}{Q_{hp} + Q_s} \quad (31)$$

Methodology of substantiation of a parametric series of solar dryers

Parametric standardization means that parameters and sizes of units are not set randomly but in compliance with the established, substantiated series of preferable numbers, which are recommended to be chosen first of all to define values of the parameters when designing items, construction, calculation, for standardization and unification, etc. Mass values of the material which is going to be dried in the chamber are chosen according to the set series of preferable numbers which are accepted according to the results of calculation of the geometric progression determined by the formula [14]

$$U_n = a \cdot Q^{n-1} \quad (32)$$

where: a – first member of the progression; Q – denominator of the progression; n – number of the progression member, n = 1, 2, 3, 4, 5.

Mass values of the material to be dried can be used to calculate constructive parameters of a solar dryer in terms of series. The geometric progression with a denominator should be considered. It is calculated by the formula

$$Q = \sqrt[R]{10} \quad (33)$$

where: R – identification mark of a series; according to the DSTU 8032 [14], the main parametric series is determined: R = 10, which meets the requirements while choosing nomenclature of the numerical values of mass of the material dried in the drying chamber; Q – denominator of the progression,

$$Q = \sqrt[10]{10} = 1.25 \quad (34)$$

Denominator of the progression Q = 1.25, the first member of the progression after rounding a = 1. A series of the numerical values of mass of the material to be dried in the drying chamber is obtained: 5.5; 6.3; 8; 10; 12.5.

RESULTS AND DISCUSSION

Results of substantiation of the constructive and technological parameters of a solar dryer with a heat pump

The detailed description of the results of substantiation of the constructive and technological parameters of a solar dryer for fruit with different configuration is supplied in [14, 15, 16]. In particular, the researchers propose design of a solar dryer with a heat accumulator and a flat mirror concentrator. The works substantiate the solar dryer structure which provides rational reduction of energy consumption in the process of fruit drying. Analytical dependences for substantiation of the constructive and technological parameters of the solar dryer are obtained, namely: area of the collector and concentrator, mass of the heat accumulator, volume of the drying chamber.

Basing on the design of a solar dryer with a heat accumulator and a flat mirror concentrator (Fig. 2), it is proposed to supplement the unit with one more structural element of the construction, i.e. a heat pump.

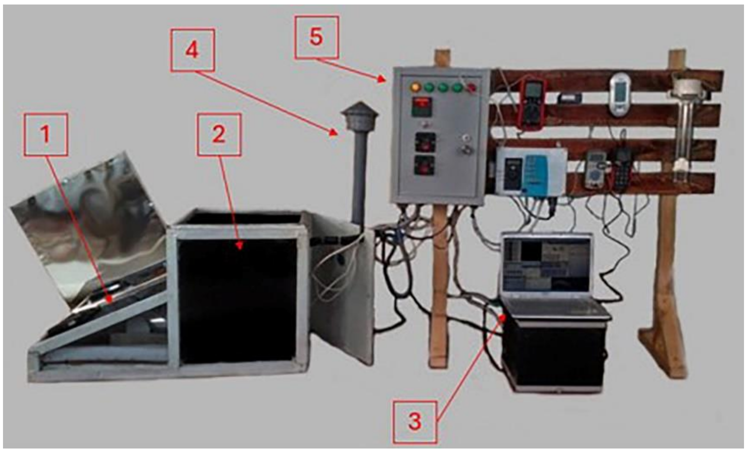

Figure 2. General view of a solar dryer with a heat pump (1 – solar collector, 2 – chamber, 3 – acquisition station, 4 – chimney, 5 – operating parameters controller)

To design the solar dryer, the following parameters are set:

1. Efficiency of the unit, which is assessed by the amount of agro-industrial production (fruit), particularly mass of the material to be dried in the drying chamber: 5.5; 6.3; 8; 10; 12.5.
2. Initial moisture content of fruit to be dried (67−78%).
3. Final moisture content of the fruit (18−24 %).
4. Thickness of the product.
5. Thermo-radiation, heat-, mass and volume characteristics of the product, elements of the solar dryer and the heat pump.
6. Speed of the heat carrier supplied to the drying chamber

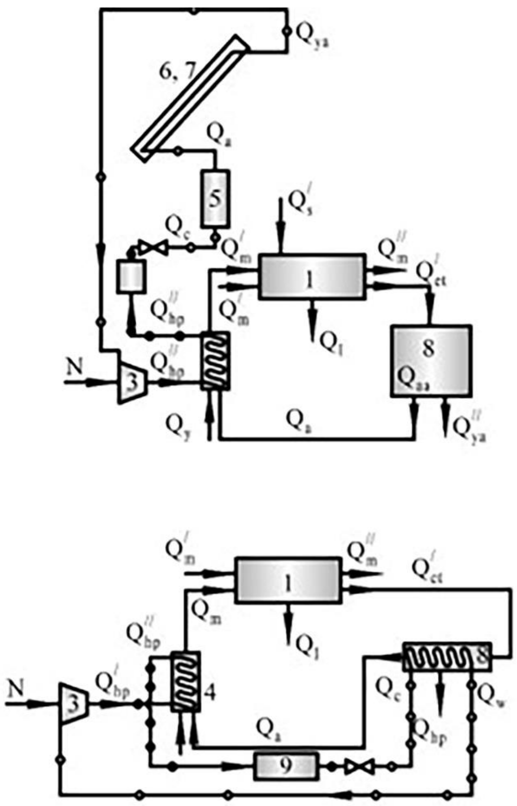

Figure 3. Principal scheme of a solar dryer with a heat pump and flows of thermal energy: 1 – chamber; 3 – compressor; 4 – condenser; 5, 6 – evaporators; 7 – solar collector; 8 – evaporator and evaporator-accumulator; 9 – receiver

The values which are calculated according to the set parameters are:
1. Heat output of the heat pump.
2. Capacity of the compressor.
3. Surface area of the evaporator and air condenser.
4. Consumption of working mass in the unit.
5. Mass of the accumulating agent.

The described methodology is used to calculate the technical and economic indicators of the proposed solar dryer equipped with a heat pump. The average specific heat output of the heat pump is calculated by the energy balance equation under the following conditions:

a) specific load made by fresh product per 1 m^2 of the chamber area is G_p = 17−15 kg/m;
b) initial moisture content of fruit W_i^c = 67 − 78%;
c) final moisture content of fruit W_f^c = 28.0%;
d) total recirculation of the exhaust air during night-time and partial recirculation during daytime (K_r = 0.65);
e) temperature of the air entering the chamber: $t_{e1} = 50 - 55$ °C; $t_{e2} = 55 - 62$ °C;
f) temperature of the exhaust air: $t_{ea} = 42 - 46$ °C; $t_{e3} = 55 - 56$ °C;
g) temperature of the recirculating air during night-time: $t_r = 33 - 35$ °C;
h) average thickness of the air flow under a rational mode of drying in night-time Q_{nt} = 0.41 kg.ms/m^2·hour, during daytime q_{dt} = 0.352 kg.ms./m^2·hour.

The formulas (5) and (6) are used to calculate the energy spent during night-time to heat the product and evaporate moisture:

$$Q_h^{/} = h_p \cdot \rho_p \cdot C_p (Q_2 - Q_1) \cdot \tau_h = 0.03 \text{kW/m}^2$$

$$Q_{et}^{/} = q_{m.} \cdot r = 0.296 \text{kW/m}^2.$$

The formula (9) can be used to calculate the loss of heat through the double-layer transparent body of the drying unit.

$$Q_{l1} = K(t_k - t_e) = 0.961 \cdot 35 = 34 \text{ kW/m}^2$$

or

$$Q_{l1}^{/} = 0.034 \text{ kW/m}^2$$

The heat loss due to air outflow of the system is measured as:

$$Q_{l2}^{/} = 0.089 \text{ kW/m}^2.$$

The total energy spent during night-time per 1 m^2 of the solar dryer:

$$Q_{nt}^{/} = Q_y^{/} + Q_{et}^{/} + Q_l^{/} = 0.45 \text{ kW/m}^2$$

Parameters of the working mass R−410a are calculated in the specific points of the system (Fig. 2b) according to the thermo-dynamic table [22, 23].

1. Temperature after throttling t_0 = 20 °C; pressure P_0 = 90.91 kPa; entropy i_0 = 444.14 kJ/kg.
2. Temperature of the working mass after the air dryer (evaporator 8, Fig. 1) t_{01} = 20 °C; entropy i_{01} = 596.6 kJ/kg.
3. Temperature of the working mass after the evaporator-accumulator t_a = 45 °C; entropy i_1 = 634.69 kJ/kg.
4. Temperature of the working mass after compression t_{co} =105 °C; temperature of condensation t_k = 70 °C; pressure P_k = 420 kPa; entropy i_2 = 664.02 kJ/kg.
5. Temperature of R−410a in the receiver during night-time t_3 = 28 °C; pressure P_3 = 420 kPa; entropy i_3 = 444.14 kJ/kg.
6. Temperature of the working mass R−410a (Fig. 3) after the air cooler (evaporator, 5) t_{01} = 20 °C, entropy i = 610.39 kJ/kg.

The formulas (14; 15) are used to calculate specific loss of heat per a unit of consumed working mass in the apparatus of the heat-pump unit:

$$Q_0 = \mathrm{i}_1 - \mathrm{i}_3 = 190.61 \text{ kJ/kg}$$

$$l = \mathrm{i}_2 - \mathrm{i}_1 = 29.33 \text{ kJ/kg}$$

$$Q_k = \mathrm{i}_3 - \mathrm{i}_2 = 219.86 \text{ kJ/kg}$$

The necessary amount of R‒410a circulating per a time unit is calculated:

$$\mathrm{M}_{wm} = \frac{Q_z^{/}}{Q_k} = \frac{0.45}{219.86} = 0.0021 \text{ kg/m}^2\text{·sec}$$

The theoretical and actual capacity of the heat pump is calculated by the formulas (18, 19):

$$N_m = \mathrm{M}_{wm.} \cdot l = 0.0021 \cdot 29.33 = 0.064 \text{ kW/m}^2$$

$$\eta = \frac{\mu_a}{\mu_t} = \frac{\mu_a}{q_k/l} = \frac{5.1 \cdot 29.33}{219.86} = 0.68 \text{ kW/m}^2$$

$$N_a = \frac{\mathrm{M}}{\eta} = \frac{0.064}{0.68} = 0.094 \text{ kW/m}^2$$

The amount of accumulated heat required during night-time (regardless of the heat loss through the accumulator walls) are calculated [24]:

$$Q_{aa}^{/} = \mathrm{M}_{wm}(\mathrm{i}_1 - \mathrm{i}_{01}) = 0.0021 \cdot (634.69 - 605.6) = 0.061 \text{ kJ/m}^2\text{·s}$$

$$Q_{aa} = \tau_{nt} \cdot 3600 \cdot q_{aa} = 12 \cdot 3600 \cdot 0.061 = 2639.4 \text{ kW/m}^2$$

Thus, it is proposed to fit the construction of a solar dryer with a prototype of the Raymer RAY-07MN air-water heat pump improved and redesigned to less capacity of 1 kW and powered by 12 W. The specific amount of the accumulating substance incoming per 1 m^2 of the solar dryer is calculated by the formula (25):

$$V_{aa} = \frac{2639.4}{1562 \cdot 0.83 \cdot 15} = 0.135 \text{ kW/m}^2$$

If the accumulator height is H = 0.15 m, its specific area is calculated:

$$S_{aa} = 0.67 \text{kW/m}^2$$

The specific heat-transferring surfaces of evaporators installed in the solar collectors and air coolers are determined by the formula (24):

$$f = \frac{Q}{K(t_x - t_0)} = \frac{0.258}{0.015 \cdot 10} = 1.72 \text{ kW/m}^2$$

$$f_{sc} = \frac{Q_{sc}}{F \cdot Q^* \cdot \eta_c} = \frac{3282}{1 \cdot 21452 \cdot 0.54} = 0.283 \text{ kW/m}^2$$

When the maximum time of the evaporator-accumulator charging is assumed as 6 hours in daytime, then the required amount of heat during daytime is calculated in the following way:

$$Q_{aa}^{/} = \frac{Q_{aa}}{\Delta\tau_{dt} \cdot 3600} = \frac{639}{6 \cdot 3600} = 0.122 \text{ kW/m}^2$$

Time of the evaporator-accumulator changing is checked by the formula (27):

$$\Delta\tau = 4.5 \text{ hour}$$

The formulas (1, 8) are used to calculated the required specific heat output of the heat pump during daytime:

$$Q_{zd}^{/} = Q_y^{/} \cdot Q_{et}^{/} \cdot Q_{aa}^{/} \cdot \sum Q_l^{/}$$

$$Q_y^{/} = R_p \cdot \rho_p \cdot \mathrm{C}_p \cdot (\Theta_3 - \Theta_2) = 0.006 \text{ kW/m}^2$$

$$Q_{et}^{/} = Q_{dt} \cdot r = 0.264 \text{ kW/m}^2$$

$$Q_{aa}^{/} = 0.122 \text{ kW/m}^2$$

$$\sum Q_{l}^{/} = 0.034 + 0.089 = 0.123 \text{ kW/m}^2$$

$$Q_{ya}^{/} = 0.084 \text{ kW/m}^2$$

$$Q_{zd}^{/} = Q_{hp}^{/} + Q_s = 0.599 \text{ kW/m}^2$$

$$Q_{hp}^{/} = Q_{z}^{/} - Q_s = 0.599 - 0.212 = 0.387 \text{ kW/m}^2.$$

During a day, the average solar radiation reaches Q_s^* = 21452 kJ/m^2. Shift index of the heat pump per day is determined by the formula (30).

$$\text{B} = \frac{Q_{nt}+Q_{dt}}{Q_{hp}+Q_s} = 0.86.$$

According to the research results, an optimal parametric series of five solar dryers is obtained. It provides assessing and substantiating the constructive and technological parameters: heat output of the heat pump, compressor capacity, surface area of the evaporator and air condenser incoming per 1 m^2 of the solar dryer, consumption of working mass in the unit, mass of the accumulating substance for conditions of small households and farms. Hence, basing on the theoretical and experimental studies, the authors are able to substantiate optimal parameters of a solar dryer and propose a parametric series of five solar dryers for the conditions of private households and farms.

Referring to the analysis of the existing technical means of fruit drying, particularly home fruit and vegetable dryers of periodic action, it is determined that their use is unprofitable in case of small amounts of fresh fruit to be processed in the conditions of private households and farms [19, 25–27]. It is primarily related with large capital investments and high energy consumption. Therefore, it is reasonable to dry small amounts of fruit in a solar dryer which provides energy-saving mode of drying. It is proposed to apply engineering methods to design main structural elements of a solar dryer with a heat pump. Referring to the obtained results, a heat pump that can twice increase heat output of the unit to solve the problem of using non-traditional environmentally safe sources of thermal energy for fruit drying is chosen.

The authors have developed and scientifically substantiated recommendations on development and forecast of a parametric series of solar dryers by mass values of the material to be dried. A parametric series of solar dryers is obtained and their optimal constructive parameters are substantiated. The results are presented in the Table 1.

The conducted studies need further development in the direction of increase of the efficiency of the technological process of fruit drying based on substantiation of the operation modes and energy analysis of the solar dryer efficiency that ensures reduction of the consumed energy sources due to applied solar energy.

Thus, the proposed solar dryer is of the appropriate characteristics being not inferior to the existing solar dryers and traditional technical means of drying.

Table 1. Parametric series of solar dryers

Parameter	Index				
Mass of the material to be dried m_m, kg	5.5	6.3	8	10	12.5
Heat output of the heat pump, N_m, kW/m^2	0.064	0.113	0.160	0.190	0.220
Compressor capacity, N_a, kW/m^2	0.098	0.112	0.120	0.180	0.252
Surface area of the evaporator and air condenser per 1 m^2 of the solar dryer, V_{aa}, kW/m^2	0.135	0.157	0.182	0.280	0.320
Consumption of working mass in the unit, Q_k, kJ/kg	219.86	269.56	286.36	318.50	340.80
Mass of the accumulating substance, M_{wm}, kg/m^2·s	0.0021	0.0069	0.0086	0.0118	0.0140

CONCLUSIONS

A solar dryer with a heat pump is designed on the basis of renewable sources of energy and the substantiated parametric series of five solar dryers. It contributes to intensification of the drying process and its further development, as well as positively influences solving environmental problems of power engineering due to substitution of the electric and thermal energy by the one obtained from solar energy, and mitigation of social problems by creating new job places needed when producing, installing and exploiting such units.

Thus, the described findings are implemented in the form of engineering methods used to calculate and predict the processes of convective and radiation drying with the variable potential of the heat and mass transfer, which create a basis for predicting the intensity of drying, development and improvement of the rational and energy-efficient technologies and drying units. Practical value of the research is revealed in a new type of equipment which uses solar thermal energy in the drying process, developed methods to determine parameters of the convective and radiation heat exchange during the drying process, which are basic for designing new dryers and creation of the energy-efficient technologies of drying.

The designed solar dryer with a heat pump contributes to solving environmental problems in the energy sector by replacing electrical and thermal energy with that obtained from solar energy. In the next steps, it is planned to conduct an environmental analysis of the dryer using the life cycle analysis (LCA). Such an analysis will allow to determine the extent to which the dryer affects individual elements of the natural environment.

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