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RESULTS OF RESEARCH INTO THERMAL-TECHNICAL CHARACTERISTICS OF SOLAR COLLECTOR

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Розроблено нову конструкцію повітряного геліоколектора, виготовленого у вигляді нероздільного енергетичного блока, що включає в себе каркас з теплоізольованими стінками, одинарним засклінням і селективною поверхнею на його днищі. Встановлено ряд узагальнювальних залежностей для знаходження теплової ефективності повітряного геліоколоектора, а саме впливу масової витрати повітря $q_{тн}$ на перепад температур теплоносія $t_{вих}$ та інсоляції E, на теплопродуктивність q і ККД η геліоколектора.

На підставі експериментальних даних отримано лінійно регресійні залежності середньої денної температури навколишнього середовища $t_{нсср}$ від енергетичної освітленості E та середньої температури теплоносія $t_{тнср}$ від середньої денної температури навколишнього середовища $t_{нсср}$. Отримані регресійні залежності мають різні коефіцієнти регресії, а саме $t_{нсср}$ дозволяє аналітично прогнозувати значення залежнозмінної середньо денної температури навколишнього середовища за допомогою незалежно змінної енергетичної освітленості E, що є хаотичною величиною. Крім цього, дає можливість виявляти і пояснити, на скільки та як змінюються середньо денні температури навколишнього середовища $t_{нсср}$ та теплоносія $t_{тнср}$ при зміні енергетичної освітленості E впродовж доби. З цього випливає, що найбільш суттєвим фактором, що впливає на роботу геліоколектора, є енергетична освітленість E. Здійснено перевірку адекватності результатів теоретичних і експериментальних досліджень.

З'ясовано, що максимальні значення ККД геліоколектора η – від 65 до 80,6 %, досягаються за температури вихідного потоку теплоносія $t_{вих}$ від 10 до 60 °C та масовій витраті повітря, $q_{тн}$ від 170 до 190 м³/год. Визначено, що зростання рівня інсоляції E від 100 до 1000 Вт/м² дає змогу збільшити теплопродуктивність колектора q від 320 до 1260 Вт та температуру теплоносія на виході з колектора $t_{вих}$ від 10 до 60 °C.

Отримані результати можна використати під час розробки та вдосконалення технічних засобів сушіння фруктів, для підвищення технологічної та енергетичної ефективності процесу

Ключові слова: сонячний тепловий повітряний геліоколектор, селективне покриття, повітряна сонячна система опалення

1. Introduction

An air solar collector (ASC) is a device designed to collect energy of radiation of the Sun in the visible and infrared spectra and to convert it into thermal energy. ASC serves mostly as an additional heating element of a low-temperature source of heat, in particular at solar dry kilns during a low-temperature mode of fruits drying. ASC uses air as a heat-transfer agent. The basic heat engineering parameters of air are temperature, saturation φ, and psychrometric difference Δt between temperatures of dry t and wet t_w thermometers of a psychrometer. The advantage of ASC is that it does not freeze and there is no boiling of a heat-transfer agent, unlike in liquid systems.

Currently ASCs are not commonly used in the energy market. Researchers developed most of ASC for countries with different types of subtropical climate, and they conducted research in laboratory or with a use of computer simulations. Thus, the fundamental problem of such devices is lack of methods for selection of elements and materials for construction and methods for calculation of heat engineering characteristics, which require additional research and development of scientifically substantiated engineering and methodological foundations. In addition, known types of ASC constructions require modifications, refinements and field tests to increase thermal efficiency of operation and heat engineering characteristics under conditions of temperate continental climate in Ukraine. In this regard, the topic of the study devoted to investigation of optimal heat engineering characteristics of ASC is relevant.

2. Literature review and problem statement

There are many variants and varieties of structural designs of flat ASC used in agricultural-and-industrial or housing-and-communal services. The main element of them, is a heat-absorbing plate (an absorber), which may have a flat, wavy, and ribbed-rimmed shape of a surface. In particular, authors of study [1] developed a flat solar air heater. They presented ways of reducing of heat loss in flat solar collectors and specified the method for determination of thermal characteristics of a passive solar air heater. They also investigated thermal power and energy efficiency of the proposed thermal-syphon solar collector. However, they did not take into consideration a flow of heat loss through enclosing elements during calculation of a collector's area. This does not make it possible to calculate transitional modes of collector's operation and heat output.

Authors of work [2] developed a solar collector of a thermal-siphon type with a ribbed-rim absorber. They conducted field tests of the thermal-siphon solar collector and presented results of field experiments on determination of thermal efficiency of a solar collector in dependence on a value of Rayleigh number Re. They obtained dependences for determination of an amount of heat received. However, they did not investigate an influence of temperature of a heat-transfer agent at the outlet from a collector, t_o, °C and intensity of solar radiation, E, W/m^2 on heat output of an air solar collector q, W.

Authors of paper [3] proposed to calculate heat engineering parameters of a developed flat ASC with a wavy absorber by the ratio of an amount of heat utilized by a collector to density of a heat flow and intensity of solar radiation. However, the proposed method did not take into consideration mass flow of a heated heat-transfer agent, components of convective and radiation heat exchange.

Work [4] presented a substantiation of construction-and-technological parameters of a developed ASC with flat and wavy absorbers. The methodology analyzes features of a construction and a principle of operation of the developed solar collectors. It also presented a simulation model and dependencies, which make it possible to simulate changes in heat engineering characteristics in the process of receiving of solar energy on a perceptual surface of the device. However, there were no field tests of ASC with flat and wavy absorbers conducted. Also, authors neglected correlation of results of tests performed in different ranges of heat output and an increase of insulation level on a perceptual surface of the collector. In addition, authors overestimated a contact area of an absorbing panel with thermal insulation, which occurs through the intermediate layer of a heat-transfer agent in a heating channel, in calculations of heat loss. Taking into consideration the above circumstances, an overall heat loss factor will decrease significantly due to the insulation component, which will eventually lead to a decrease in thermal efficiency of the collector.

Authors of paper [5] formulated the actual task of an increase in thermal efficiency and reduce in overall dimensions of the developed ASC. They presented a methodology for calculation of the main heat engineering and technical-economical parameters. However, the described methodology is very general and does not correspond to the modern achievements of solar energy. In particular, the methodology calculates direct and scattered components of heat flows of solar radiation, which falls on one square meter of horizontal surface in the cloudless sky, by the recommendations of construction climatology [5]. There are tables of standard averaged flows of energy from direct and scattered radiation in the clear sky, and through them – an average energy illumination of a perceptual surface, taking into consideration the cloudiness factor. For most tasks, we can replace such calculations by computational results for typical orientation surfaces provided on NASA's website [6].

Paper [7] proposed construction of a corrugated solar ASC with a hermetically sealed and insulated body, which can serve as a part of a roof. There are regularities of an influence of a heat-transfer agent flow rate, temperature difference and intensity of solar radiation on ASC thermal power determined. Based on processed experimental data, there is efficiency coefficients for the subtropical climatic conditions of Ankara (Turkey) calculated and a number of dependencies obtained for finding thermal efficiency of the solar collector. However, a paper did not take into consideration a mass flow rate of the air and temperature of an incoming heat-transfer agent, which is close to the ambient temperature, along the direction of the air flow between an absorbing plate and a glass coating for increasing of thermal efficiency of the collector. This does not make it possible to solve the corresponding task during diagnosis and forecasting of heat engineering parameters of a heat-transfer agent between an absorbing plate and a glass coating in the process of convective-radiation heat transfer.

Work [8] presented a substantiation of construction-and-technological parameters of ASC for a grain solar dryer of a drum type. In particular, it proposed a method for determination of an optimal area of a metal substrate for an absorption surface, and increasing of temperature of a heat-transfer agent in the collector due to additional irradiation of solar dryer. However, it neglects correlation of results of tests performed in different ranges of a heat-transfer agent flow and energy illumination of the collector. The latter, in turn, does not make possible to obtain reliable information on thermal power and efficiency of installation associated with changes in physical parameters of the environment. This does not make it possible to extrapolate numerical values of thermal power and efficiency of ASC to a zero value for determination of thermal efficiency of a drying plant in accordance with the standard AM 1 [9] and AM 1.5 methodology [10].

Researchers applied the developed methods for selection and calculation of heat engineering characteristics of ASC under laboratory conditions or with a use of simulation models during computer simulation. Therefore, known types of ASC designs require further studies and field tests to increase heat output and efficiency of a device based on justification of optimal parameters and operating modes in the temperate continental climate in Ukraine. Thus, the decisive aspect for deciding when using ASC is the study of heat engineering characteristics and substantiation of its thermal efficiency.

The given arguments made possible to determine the main directions of the investigation of heat engineering characteristics of the proposed air solar collector during its study in the field for the temperate climate of Ukraine, in particular for the city of Korts, Rivne region (Ukraine).

3. The aim and objectives of the study

The objective of the study is to increase efficiency of heat engineering characteristics of an air solar collector based on substantiation of its operation modes, which will reduce a cost of energy resources.

It was necessary to perform the following tasks to achieve the objective:

– evaluation of operational heat engineering characteristics of an air solar collector under standard solar illumination and typical (seasonal) meteorological conditions;

– investigation of heat engineering characteristics of an air solar collector in dependence on a change of physical parameters of the environment;

– field tests of a solar collector.

4. Materials and methods to study the heat engineering characteristics of an air solar collector

4. 1. Substantiation of structural-and-technological parameters and a structure of an air solar collector

In the agroindustry, particularly, at individual farms, ASC often serves for drying of internal moist walls of a house, grain storage facilities, sheds under conditions of temperate continental climate of Ukraine. For example, it is possible to use ASC fully in the zone of western Polissya [11]. Specifically, there are many different designs of ASCs, which are active systems of solar energy use, developed at the Department of Energy of the Lviv National Agrarian University (Lviv, Ukraine).

ASC has the form of an indivisible energy block, which includes a box with insulated walls, a single glazing and a selective surface on its bottom. ASC frame has a size of 1×1.5 m. It represents the classic slit version. It consists of a wooden frame made of 25×150×4,000 cut-wood pine board. There is a quarter of a depth of 16 mm and a thickness of 10 mm picked out for better sealing of walls in one side of the board. We used foam plasters of 985×1,485 mm with a thickness of 12 mm and with a thermal conductivity of $\lambda=0.040$ W/m·K for heat insulation. We covered the rear wall of ASC with 985×1,485 mm wood-fiber plate and coated it with an aluminum foil thermal insulation film with a thermal conductivity coefficient of $\lambda=0.020$ W/m·K. We mounted angle corners of 140 mm length made of ordinary steel profile angular of 60×60×5 mm along the angled sides of the wooden frame of the solar collector for strength and geometricity of the body. We drilled 18 holes with a diameter of $d=55$ mm in the lower and the upper parts of the solar collector. We introduced a round metal-polyethylene air duct with a diameter of $d=54$ mm and a length of 10,000 mm into the holes for better heating and circulation of air in the middle of the collector.

At the inlet to the air duct, we mounted a compressor small-sized fan of YM1217ANB1 type with capacity of 12.6 W and productivity of 348.3 m^3/h. A 12 V DC power supply from a solar powered Perlight PLM-020P/12 battery with a power of 20 W and an autonomous power supply system powers the fan. This makes it possible to reduce hydraulic resistance of circulating air and heat losses in the middle of the collector in the air duct twice. A centrifugal fan of TORNADO DE 100 1F type with capacity of 45 W and productivity of 240 m^3/h performs removal of a heated heat-transfer agent from the collector's duct, for example, for needs of a solar dryer.

It is difficult to provide stable air exchange in the collector. Because it depends on changes in alternating, sharp and sudden modes of sunlight illumination and typical (seasonal) meteorological conditions and we can't take them into consideration during modeling. Particularly, a level of insulation that affects operation of a solar cell battery W and autonomous power supply system and those in turn, on performance of fans.

The upper part of ASC is made of a single-layer transparent coating, namely, of glass with a heat-reflective coating of solid 0.02 % Fe_2O_3 type with an emission coefficient of $\varepsilon=0.1...0.15$. The absorber is made of a thin copper sheet with a thickness of 0.9 mm coated with a selective paint of «Tinox» type of 4.5 μm thickness on the front side. The air goes through the air duct through the holes between the back part of the copper substrate of ASC and the aluminum foil thermal insulation film, which simultaneously serves as the undercoat for ASC. The protuberances of the profile located across an air flow for its turbulization serve to increase efficiency of the heat transfer absorber.

To increase efficiency of heat transfer from the absorber, we turbulized an air flow by selection of a slit cross section, flow velocity or additional structural elements. Thus, we exclude a need for double glazing, and an increase in heat output of the collector compensates a small decrease in temperature of an outflow. Therefore, in order to increase the efficiency of heat transfer of the rear side of the absorber, we attach additional supports in the form of bent sheet copper channels of 20...30 mm height. They increase rigidity of the absorber and prevent its oscillation during interaction with a turbulized air flow. Fig. 1 shows the schematic structure of ASC. Fig. 2 exhibits mounting stages.

The maximum sensitivity of the growth of heat output from a change in the angle of inclination β_{opt} to the horizon is 20 % for stationary ASC in two summer months. And in two autumn months, heat output of the collector does not depend on the angle of inclination significantly. Therefore, we choose the optimal angle of inclination of ASC to the horizon for Korts city of Rivne region (Ukraine) β_{opt} as an angle close to the average annual optimal one, which is – 40.4° according to NASA [6].

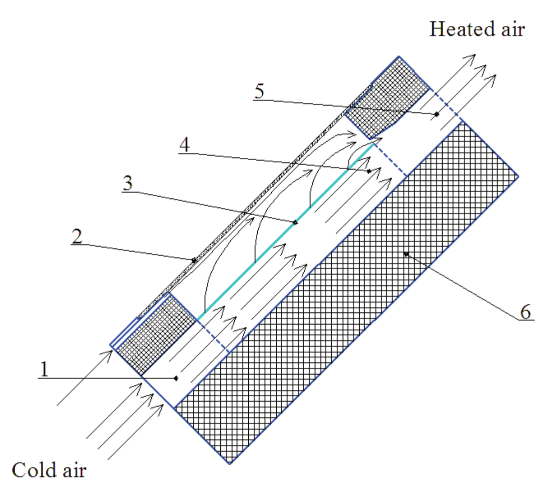

Fig. 1. Structural diagram of the design of the air solar collector: 1 — an input channel; 2 — single-layer transparent coating; 3 — an absorber; 4 — an air duct; 5 — an output channel; 6 — a heat-insulating wall

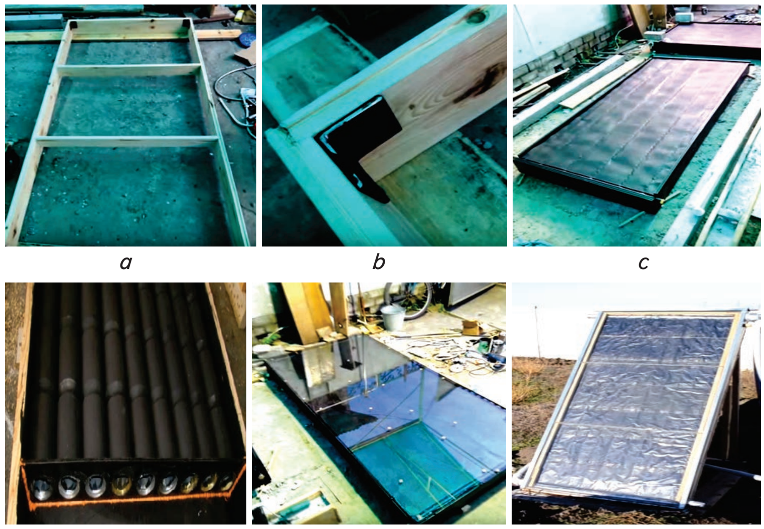

Fig. 2. Stages of assembly of design elements of the air solar collector sample: *a* — assembly of a collector's frame; *b* — verification of the geometry of the collector's body; *c* — fixing of the absorber; *d* — placement of air channels; *e* — installation of single-layer glass with a heat-reflective coating of solid 0.02 % Fe_2O_3 type; *f* — installation and commissioning of the sample ASC

The flat mirror concentrator rotates around the axis, which is parallel to the longitudinal side of ASC. Reflected rays illuminate a perceptive surface of ASC additionally in the morning until noon by changing of the angle of inclination manually in the range from 0 to 120°. It is necessary to reinstall the turning device on the opposite side of the collector after noon.

Thus, the developed design of ASC corresponds to the concept of an active solar power plant. At the same time, the mirror concentrator and the air collector combined in one energy block do not correspond structurally to the classic models of solar thermal installations. There is no theoretically determined correlation between energy parameters for the developed installation. For example, it is impossible to test ASC and the mirror concentrator separately according to the standard method, or to calculate heat engineering parameters of the collector or to investigate the process of its operation. Specifically, quality of the heat energy received depends on changes in physical parameters of the environment. Therefore, we define parameters, which determine effectiveness of decisions, during implementation of experimental studies based on the analysis of ASC operation.

4. 2. Theoretical prerequisites for substantiation of heat engineering characteristics of the air solar collector

An air solar collector operates in the mode of maximum heat output. An energy efficiency coefficient characterizes heat output numerically.

In view of the variability of parameters of incoming solar energy, a solar collector operates in the near-stationary mode during a light day. Therefore, its operation parameters correspond to the steady state mode. The equilibrium between input and output energy flows determines such mode. An input heat flow is the absorbed by the absorber solar energy q_c. We can determine it from formula:

$$q_c = K_p \cdot K_r \cdot A_a \cdot E \cdot S_{sk} \cdot \tau, \text{ kJ}, \tag{1}$$

where K_p is the coefficient of pollution of an ASC body, $K_p = 0.95$; K_r is a coefficient of repeated reflection of solar radiation from the absorber to the light-penetrating material of ASC, $K_r = 0.23$; A_a is the average absorption capacity of the absorber; E is the energy illumination, W/m^2, S_{sk} is the irradiated surface, m^2; $d\tau$ is the duration of drying, s.

Output air flows consist of a useful heated heat-transfer agent carried by mass q_a, and a flow of losses to the environment q_l. Here q_l is equal to the sum of convective q_c [12] and radiation q_r [13] heat exchange.

$$q_a = G_m \cdot c_p \cdot t_o; \tag{2}$$

$$q_l = q_c + q_r, \tag{3}$$

where c_p is the specific heat of air at constant pressure, kJ/(kg·°C); G_m is the mass flow rate of air, kg/m^2·s; t_i is the temperature of an incoming air flow, °C.

Temperature distribution along the air duct is steady in time during the stationary mode of ASC operation. Therefore, the basic condition of equilibrium of input and output heat engineering parameters of the solar collector remains the same for each element of b width and dy length. Here, an increase in temperature Δt in each element is proportional to its gradient $\Delta t/\Delta y$, which goes into the derivative dt/dy under boundary conditions. For each element of the solar collector, heat loss in the environment is proportional to surface area bdy and a temperature difference inside the collector t and the environment t_e:

$$dq_l = K \cdot (t - t_e) \cdot bdy, \tag{4}$$

where K is the effective coefficient of heat transfer of a collector element, W/m^2·K; t, t_e are the temperature inside the collector and temperature of the environment, respectively, °C; b is the absorber's width, m; dy is the length of absorber, m.

We can describe the thermal equilibrium of each element of ASC cross-section with the following differential equation [1]:

$$\eta_0 \cdot E \cdot bdy = K \cdot (t - t_e) \cdot bdy + G_m \cdot c_p \frac{dt}{dy} dy, \tag{5}$$

where η_0 is the optical efficiency of the solar collector.

After the standard transformation we obtain:

$$\frac{dt}{dy}+\frac{K\cdot b}{G_m\cdot c_p}\cdot t-\frac{\eta_0\cdot E\cdot b+b\cdot K\cdot t_e}{G_m\cdot c_p}=0. \tag{6}$$

Record (6) is typical for linear differential equations of the first order:

$$\frac{dt}{dy}+P(y)t=Q(y), \tag{7}$$

the difference is that functional values take stable values:

$$P(y)=\frac{Kb}{G_m c_p}=\text{const and } Q(y)=\frac{\eta_0 Eb+bKt_e}{G_m c_p}=\text{const}. \tag{8}$$

First, we need to find a partial solution t^* for the case $Q(y)=0$, for which separation of variable temperatures of input and output air flows is possible in ASC:

$$\frac{dt}{dy}+\frac{Kb}{G_m c_p}t=0 \text{ or } \frac{dt}{t}=-\frac{Kb}{G_m c_p}dy \text{ and}$$

$$\int\frac{\mathrm{d}t}{t}=-\frac{Kb}{G_m c_p}\int \mathrm{d}y, \tag{9}$$

$$\frac{dt}{dy}+\frac{Kb}{G_m c_p}t=0 \text{ or } \frac{dt}{t}=-\frac{Kb}{G_m c_p}dy \text{ and}$$

$$\int\frac{\mathrm{d}t}{t}=-\frac{Kb}{G_m c_p}\int \mathrm{d}y, \tag{10}$$

$$\ln t+C=-\frac{K\cdot b}{G_m\cdot c_p}y \text{ or } t^*=C_1\exp\left(-\frac{K\cdot b}{G_m\cdot c_p}y\right), \tag{11}$$

where $C_1=e^C$.

Thus, at the inlet to the collector, when $y=0$, and temperature is equal to $C_1=t_i$ then the value of the temperature of an outflow of a heat-transfer agent in other ASC elements is a function of distance $u(y)$. Therefore, we take the general solution of the differential equation (6) in the form of the product of the partial solution (11) and a change in the constant at the input of each of the next collector element $u(y)$, and we obtain:

$$t=u(y)\exp\left(-\frac{K\cdot b}{G_m\cdot c_p}y\right). \tag{12}$$

We substitute the obtained equation (12) in the original equation (6) and obtain:

$$\frac{d}{dy}\left[u\exp\left(-\frac{K\cdot b}{G_m\cdot c_p}y\right)\right]+\frac{K\cdot b}{G_m\cdot c_p}\times$$

$$\times u\exp\left(-\frac{Kb}{G_m\cdot c_p}y\right)=\frac{\eta_0\cdot E\cdot b+b\cdot K\cdot t_e}{G_m\cdot c_p}. \tag{13}$$

Here, for the first summand, we perform differentiation according to the rule of the product of two functions:

$$\exp\left(-\frac{K\cdot b}{G_m\cdot c_p}y\right)\frac{du}{dy}-u\frac{K\cdot b}{G_m\cdot c_p}\exp\left(-\frac{Kb}{G_m\cdot c_p}y\right)+$$

$$+\frac{Kb}{G_m c_p}\cdot u\exp\left(-\frac{K\cdot b}{G_m\cdot c_p}y\right)=\frac{\eta_0\cdot E\cdot b+b\cdot K\cdot t_e}{G_m\cdot c_p}. \tag{14}$$

After a mutual contraction of the second and third summands of the equation, we reduce it to the form with separated variables:

$$du=\frac{\eta_0\cdot E\cdot b+b\cdot K\cdot t_e}{G_m\cdot c_p}\exp\left(\frac{K\cdot b}{G_m\cdot c_p}y\right)dy. \tag{15}$$

We integrate the left part with a new unknown constant C_2, and the right part – in the range from 0 to y:

$$\int \mathrm{d}u=\frac{\eta_0\cdot E\cdot b+b\cdot K\cdot t_e}{G_m\cdot c_p}\int_0^y\exp\left(\frac{K\cdot b}{G_m\cdot c_p}y\right)\mathrm{d}y \tag{16}$$

or

$$u+C_2=\frac{\eta_0\cdot E\cdot b+b\cdot K\cdot t_e)}{G_m\cdot c_p}\cdot\frac{G_m\cdot c_p}{K\cdot b}\left[\exp\left(\frac{K\cdot b}{G_m\cdot c_p}y\right)-1\right]. \tag{17}$$

Here the boundary condition for determination of C_2 constant is temperature at the inlet to the collector, that is, if $y=0$, the sum $u+C_2=t_i$, hence:

$$u+C_2=\left(\frac{\eta_0 E}{K}+t_e\right)\left[\exp\left(\frac{Kb}{G_m c_p}y\right)-1\right]-t_i. \tag{18}$$

Therefore, the general solution to equation (8) taking into consideration expressions (12) and (18) takes the form:

$$t=\left\{\left(\frac{\eta_0\cdot E}{K}+t_e\right)\left[\exp\left(\frac{K\cdot b}{G_m\cdot c_p}y\right)-1\right]-t_i\right\}\exp\left(-\frac{K\cdot b}{G_m\cdot c_p}y\right). \tag{19}$$

Thus, after elementary operations of multiplication with the subsequent reduction of exponents and disclosure of curly brackets from expression (19), we obtain the initial temperature of a heat-transfer agent from ASC:

$$t_o=\frac{\eta_0\cdot E}{K}+t_e-\left(\frac{\eta_0\cdot E}{K}-t_e+t_i\right)\exp\left(-\frac{K\cdot b}{G_m\cdot c_p}y\right). \tag{20}$$

We determine heat output of ASC from formula:

$$q=S_{sk}\cdot F_R\cdot\left[\begin{array}{l}\left(k(\tau)\cdot R_\beta\cdot E^{\max}\cos\pi\frac{\tau}{\tau_c}\right)\times\\ \times(\alpha\cdot\tau)-U_L\cdot(t_o-t_e)\end{array}\right], \text{ kJ/h or kW}, \tag{21}$$

where S_{sk} is the ASC area, m^2; F_R is the ASC heat emission coefficient from ASC; k is a coefficient of amplification of solar energy by a mirror concentrator; R_β is a coefficient of average monthly income of solar radiation; E^{max} is the maximum energy illumination of a horizontal surface of ASC, W/m^2; U_L is a coefficient of ASC thermal losses, W/(m^2·°C); τ_c is the duration of receiving solar energy, s; α, τ are the coefficients of absorption and transmission of solar radiation; t_e, t_o are, respectively, the air temperature at the inlet to the solar collector and the outlet from it, °C.

The area of ASC S_{sk} takes the form:

$$S_{sk} \geq \frac{q/2}{3600 \cdot \eta \cdot H_\beta^d}, \text{ m}^2, \quad (22)$$

where q is the daily demand for thermal energy, kJ; η is the efficiency of the solar collector; H_β^d is the solar energy arrival, kW/m^2.

Thus, the efficiency of the collector takes the form:

$$\eta = \frac{q}{\eta_0 \cdot E} = \frac{G_m \cdot c_p}{K_{sk}} \cdot \left(1 - e^{-N_{sk} \cdot \xi}\right) - \\ - \frac{G_m \cdot c_p}{E} \cdot \left(1 - e^{-N_{sk} \cdot \xi}\right) \cdot (t_{at} - t_e), \quad (23)$$

where N_{sk} is the thermal power of the air solar collector, W, ξ is a coefficient of heat transfer, W/m^2·K, t_{at} is the average temperature of circulating air in the solar collector, °C.

Thus, the obtained dependences make it possible to calculate heat engineering characteristics of ASC, namely, t_o is the initial heat-transfer agent temperature, (20), q is the heat output, (21), η is efficiency of the solar collector (23).

4. 3. Substantiation of the methodology of investigation of heat engineering characteristics of the air solar collector

We selected the spring-summer season from May to June, namely, from May 15 to June 15, 2018, to study ASC operation. There is usually stable sunlight at this time of a year, the weather is close to the optimum one for testing of solar power plants.

During experimental research, we equipped the experimental plant with measuring tools and sensors. Fig. 3 shows the set-up.

We measured the energy illumination amplified by a flat mirror concentrator with PELENG SF-06 pyranometer (OJSC Safonovsky Plant «Hydrometriprilad», Russia). We determined the velocity of a heat-transfer agent with Testo 425 heat-loss anemometer (Testo, Germany).

We performed temperature measurement of the environment, a heat-transfer agent, heat-accumulating material simultaneously with PT-0102-8 regulator-meter and Pt 100 thermistors (PAT NVO «Termoprylad» named after V. Lakh, Ukraine).

We determined the relative humidity of air with WCM-1 air humidifier with EE HC 200 humidity sensor (TROTEC, Germany), and of a heat-transfer agent in the collector – with PT-0102 digital thermal hygrometer with Pt 1000 thermistors (PAT NVO «Termoprylad» named after V. Lakh, Ukraine).

Therefore, it is more appropriate to carry out natural tests on a special stand equipped with low-inertial temperature sensors and small-sized flow velocity sensors that do not distort the nature of an air flow in the air duct. Moreover, it is necessary to carry out registration in the continuous mode with visualization of real-time results for their archiving and subsequent analysis.

5. Results of the investigation of heat engineering characteristics of the air solar collector

We performed the natural tests of ASC at «Zorya» private farm located in the city of Korts, Rivne Oblast (Ukraine), in the spring-summer period from May 15 to June 15, 2018. We used the results of weather monitoring of the Koretsky meteorological station of the first level of Rivne region (Ukraine) for refinement of the standard modes of solar illumination and typical (seasonal) meteorological conditions.

The weather was clear without precipitation in the city of Korts, Rivne region (Ukraine) during ASC testing period from May 15, 2018 to May 16, 2018. The degree of transparency of the atmosphere varied from 0.72 to 0.86. The air mass flow (wind) ranged from 1.3 m/s to 2.8 m/s.

During our study, the average daily physical parameters of the environment were as follows:

1. Air temperature t_e – 12...32 °C.
2. Relative humidity of air φ_e – 84.5...62 %.
3. Energy illumination E – 100...988 W/m^2 for the area of the absorbing surface S_{sk}=1.5 m^2.
4. Atmospheric pressure of air masses fluctuated within 740–744 mm of mercury.

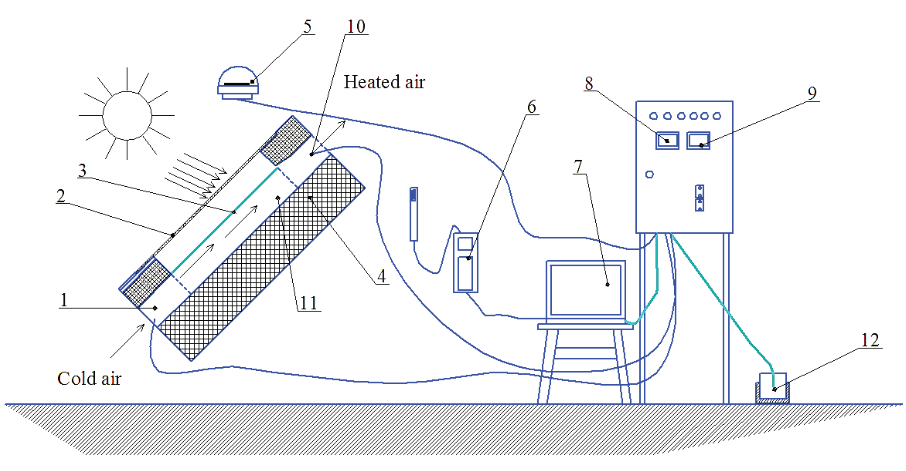

Fig. 3. General view of the experimental plant:
1 – an input channel; 2 – single-layer transparent coating; 3 – an absorber; 4 – an insulating wall; 5 – PELENG SF-06 – digital pyranometer; 6 – Testo 425 heat-loss anemometer; 7 – a computer; 8 – PT-0102-8 regulator-meter; 9 – PT-0102 thermal hygrometer; 10 – an output channel; 11 – an air duct; 12 – a thermostat

5. Heat engineering parameters of a heat-transfer agent (air) entering the output channel were: daytime temperature (from 8:00 to 21:00 hours) t_o – 15...71 °C, night temperature (from 22:00 to 7:00 hours) t_o – 50...14 °C.

6. Velocity of circulation of a heat-transfer agent (air) v_o – 1...3 m/s.

7. Relative humidity of a heat-transfer agent (air) φ_o – 10.8...82.3 %.

We performed temperature measurement of heat engineering parameters of air (a heat-transfer agent) in Celsius degrees (°C) in the course of experimental studies. We converted them into units of thermodynamic temperature (K) during calculations. We carried out correlation of adequacy of the results of theoretical and experimental studies for purity of the test in Celsius degrees (°C).

We translated numerical values of the mass flow rate of air in the solar collector q_a in m^3/h in accordance with technical characteristics of the fan for convenience, accuracy of identity and purity of the study. In particular, in accordance with curves of fan's productivity and its passport data.

The efficiency of ASC depends on the orientation of ASC, an angle of inclination of ASC to the horizon β_{opt}=40.4°, the latitude (for Korets, Rivne region – 50.61°), as well as on air pollution. In the period of experimental studies, the average value of energy illumination was within the range of E – 100...988 W/m^2. We recorded the maximum (1,345.5 W/^{2}) on May 16 at 13:00 hours using the flat mirror concentrator.

We determined values of heat engineering characteristics during execution of calculation-quantitative experiments for the analysis of ASC operation. Table 1 shows summarized results of the production tests of the developed ASC.

Thus, the distribution structure of solar energy (Table 1), which entered ASC, showed that the absorber used solar energy in the range from 100 to 1000 W/m^2 during a day (from 8:00 to 21:00 hrs.). This enabled ASC to produce thermal energy in the range of 789.4 MJ on May 15 to May 16, 2018, and with the use of the flat mirror concentrator on May 16 to May 17, 2018, we obtained it within the limits of 1,223.1 MJ. The heat output of ASC was on average 568–889 W. The operating ASC absorption surface is S_{sk}=1.5 m^2, so it can receive on average 600–1,460 W of thermal energy. However, according to the calculation of thermal productivity, ASC produced energy in the range of 610.5–1,223.1 W.

We obtained linear regression dependence based on experimental data: the average daily temperature of the environment t_{eat} on energy illumination E in the period of May 15 to May 20, 2018 (Fig. 4):

$$t_{eat} = 1.0893 \cdot E + 16.571. \quad (24)$$

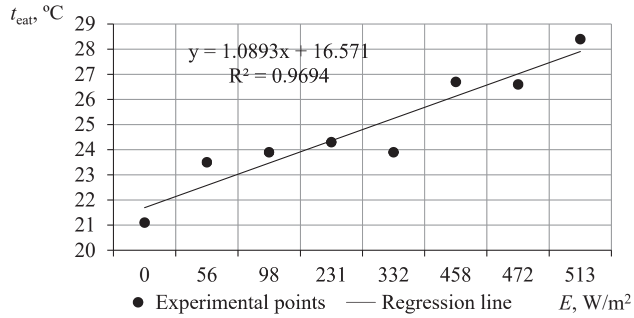

Fig. 4. Dependence of the average ambient temperature on energy illumination in the period of May 15 to May 20, 2018

The correlation coefficient of the output data was R=0.994, the determination coefficient was R^2=0.9694. For this regression model, the criterion F_p=5.56 was at the level of n=0.05 with the tabular value of the criterion F_t=5.59, which indicates the adequacy of the model. The verification by t-criterion confirmed the reliability of confidence intervals of regression coefficients.

Consequently, the thermal efficiency of ASC ranged from 610.5 to 1,223.1 W with the active absorbing surface area S_{sk}=1.5 m^2.

The main heat engineering parameter of ASC is the dynamics of a change in the temperature of air (a heat-transfer agent) at the inlet and the outlet of the air duct (Fig. 5). For example, the temperature of the heat-transfer agent was 38.6 °C at the outlet from the air duct of the collector on May 15, at about 10:00 hours, and it reached the maximum upper peak of 49.4 °C over 13:00 to 15:00 hours. And from 15:00 to 19:00 hours, the temperature of the heat-transfer agent dropped to 45.3 °C. In the night period from 15 to 16 May from 20:00 to 5:00 hours, the temperature of the heat-transfer agent ranged from 30.6 °C to 25.1 °C, and it reached the lower peak of 15 °C at about 6:00 hours. The velocity of circulation of the heat-transfer agent (air) in the air duct of the solar collector was v_a – 1.8 m/s.

The ambient temperature reached the maximum upper peak from 23.3 to 25.6 °C on May 15 from 10:00 to 18:00 h. The ambient temperature varied from 22.8 to 25.1 °C on the night of May 15 to 16 from 22:00 to 9:00 hours, and it had a lower peak at 15 °C at about 6:00 hours.

Based on experimental data, we obtained linear regressive dependences of the average temperature of the heat-transfer agent t_{aat} on the average daily temperature of the environment t_{eat} in the period of May 15 to May 20, 2018 (Fig. 6):

$$t_{aat} = 4.1821 \cdot t_{eat} + 20.386. \quad (25)$$

The correlation coefficient of the output data was R=0.96, the determination coefficient was R^2=0.9501. For this regression model, the criterion F_p=3.46 was at the level of n=0.056 with a tabular value of the criterion F_t=4.23, which indicates the adequacy of the model. The verification of t-criterion confirmed reliability of confidence intervals of regression coefficients.

Table 1

Summarized results of study into heat engineering parameters of the air solar collector in the period from May 15 to June 15, 2018

No.	Duration of operation of the solar collector, hours	Duration of an interval of operation of the solar collector τ_d, hours (day)	Plant configuration		E, W/m^2	q_c, MJ
			ASC	+FMC*		
1	24	May 15, 2018	+	+	996	1,108.5
2	48	May 15 to May 16, 2018	+	–	835	789.4
3	72	May 16 to May 17, 2018	+	+	1,123	1,223.1
4	96	May 17 to May 18, 2018	+	–	768	610.5
5	120	May 19 to May 20, 2018	+	+	714	974.8

*Notes: * – + a flat mirror concentrator*

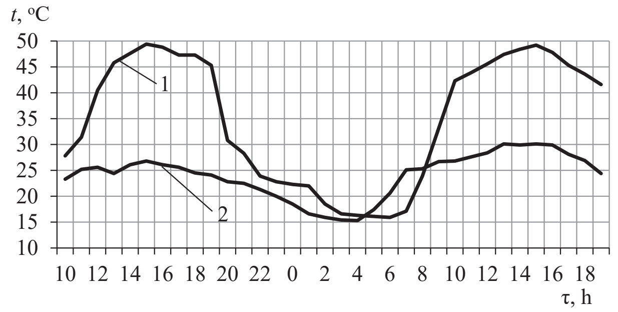

Fig. 5. Dynamics of a change in the temperature of air (a heat-transfer agent) in solar collector on May 15 to May 16, 2018: 1 – the temperature of the heat-transfer agent at the outlet from the air duct of the solar collector; 2 – the temperature at the inlet to the air duct of the solar collector (ambient temperature)

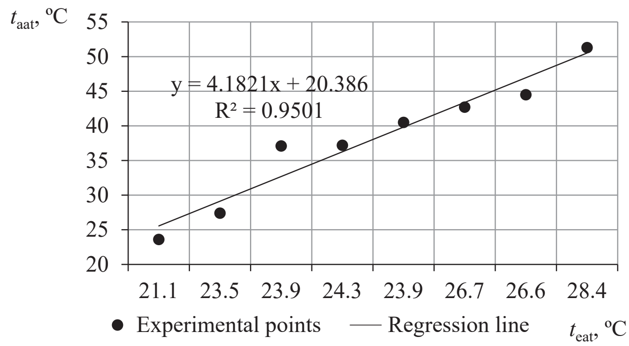

Fig. 6. Dependence of the average temperature of the heat-transfer agent on the average daily temperature of the environment in the period of May 15 to May 20, 2018

Based on calculation-quantitative experiments, we constructed graphs of heat engineering characteristics. In particular, Fig. 7 shows the dependence of the efficiency of the developed ASC, η, % on the temperature of the heat-transfer agent at the outlet from it, t_o, °C and the mass flow of air, q_a, m^3/h.

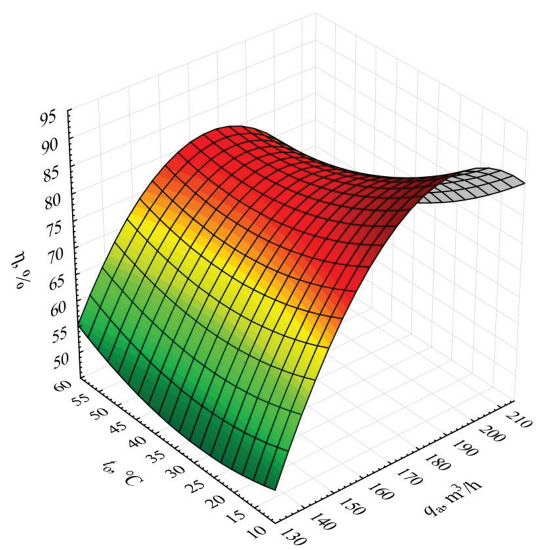

Fig. 7. Dependence of efficiency of the solar collector, η, % on the temperature of the heat-transfer agent at the outlet from the solar collector, t_o, °C, and its mass flow rate, q_a, m^3/h

In addition, we constructed a graph of the dependence of heat output of the developed ASC q, W on the temperature of the heat-transfer agent at the outlet from the collector, t_o, °C and intensity of solar radiation, E, W/m^2 (Fig. 8).

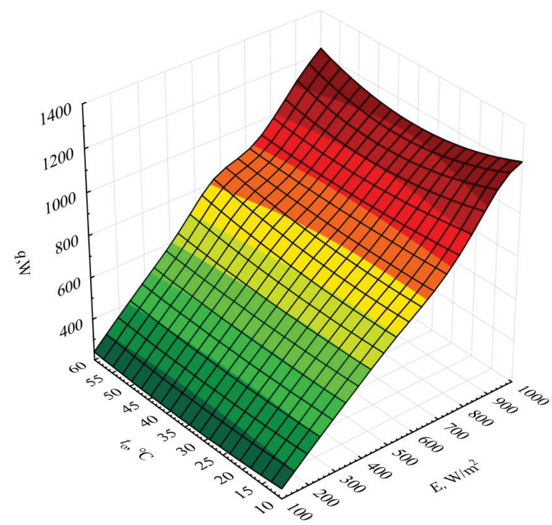

Fig. 8. Dependence of heat output of the air solar collector q, W on the temperature of the heat-transfer agent at the outlet from the collector, t_o, °C and intensity of solar radiation, E, W/m^2

The analysis of the obtained results shows that the maximum values of efficiency of the solar collector η (from 65 to 80.6 %) appear at the temperature of the output flow of the heat-transfer agent t_o from 30 to 60 °C and the mass flow of air, q_a from 170 to 190 m^3/h. In the case of an increase in the mass flow of air, q_a from 190 to 200 m^3/h, heat engineering characteristics, namely, the efficiency of the solar collector and the temperature of the initial flow of the heat-transfer agent t_o fall intensely and nonlinearly (Fig. 7). Therefore, to increase the temperature of an outflow of air, it is necessary to reduce the heat-transfer agent flow rate with the simultaneous intensification of the process of its turbulization to maintain heat output of ASC. Because during the simulation, numerical values of the efficiency of the solar collector η ranged from 68 to 82.4 %. The diagram represents this in the operation red zone (Fig. 7), and the experimental parameters ranged from 50 to 65.3 % in the operation green zone. This happens due to a sudden change in the mode of solar illumination and typical (seasonal) meteorological conditions, which we can't consider during simulation. In addition, we established that an increase in the insulation level E from 100 to 1,000 W/m^2 can increase heat output of the collector q from 320 to 1,260 W and the temperature of the heat-transfer agent at the outlet from the collector t_o from 10 to 60 °C and vice versa (Fig. 8). This makes it possible to explain how redistribution of ratios of the maximum heat output, the temperature of the heat-transfer agent at the outlet of the collector and its efficiency, despite a variety of physical parameters of the environment and weather conditions, takes place.

Thus, the proposed design of ASC showed its work capacity, reliability, and excellent thermal efficiency.

6. Discussion of results of studying the heat engineering characteristics of the air solar collector

The result of our study has proven that it is necessary to use ASC as an additional heating element of a low-temperature heat source. We can use ASC in solar dryers at low-temperature mode for drying of fruits, grain, or in ventilation systems for partial drying of air due to its cheap design elements, ease of installation and commissioning and operating conditions.

A collector in the form of a box with insulated walls, single glazing and a selective surface on its bottom is the simplest one. The proposed design solutions, namely single glazing, minimized a flow of convective heat loss from an absorbing surface through a large heat transfer coefficient across a turbulized air flow. In addition, we know that radiation heat loss flow is approximately twice greater than free convective flow at temperatures close to room temperature, therefore, it is necessary to use a glass of 0.02 % Fe_2O_3 type [11] to reduce it. The advantage of this coating is high mechanical stability, which makes it possible to use it on open surfaces, and the disadvantage is a slightly higher (than for «soft» coverage) radiation ($\varepsilon = 0.1...0.15$).

The heat transfer process from the perceptual surface to the useful flow of air occurs through an intermediate layer of a metal substrate with a high coefficient of thermal conductivity. To increase efficiency of heat transfer, we attached nine round metal-polyethylene air ducts with a diameter of $d = 54$ mm and a length of 10,000 mm to the rear of the receiving panel and the same quantity of bended sheet metal copper channels of 20...30 mm in height for better heating and circulation of air in the middle of the collector. Since the ratio of the length of the air duct to its diameter is greater than 10, contribution of boundary effects of a developed turbulent flow is insignificant and we can neglect heat transfer effects associated with it. At low flow velocities, the mode of developed turbulence is maintained along the collector, if $L/2l > 10$, where L is the length of L channel, and $2l$ is its double height. A height of the channel with turbulized flow should not exceed 7 cm for a collector of 1.5 m of length. In addition, this gives possibility to increase rigidness of the thin sheet and to prevent its oscillation at interaction with the turbulized air flow.

To increase the efficiency of heat transfer from the heated panel, we turbulized the air flow by selection of the slit cross section and two additional centrifugal fans. Instead, in the slit under the glass cover, it is foreseen to drain overheated air from its upper part through holes of small diameter directly under the roof glass, and the upper holes – near the absorbing panel. This prevents formation of condensation on a surface of the absorber, due to effective ventilation of a space under the glass and the absorbent plate. In this case, emerging vertical flows will be partially cut off by a tangent to the glass weak laminar flow of fresh air. Thus, we eliminate a need for double glazing, and an increase in heat output of the collector compensates a small decrease in the temperature of an outflow. Due to such structural design, we align a temperature field of the transparent coating and reduce a level of convective and radiation components of heat loss. Due to this design, we can adjust characteristics of the air collector within certain limits also by changing of lower and upper gaps between the glass cover and the body. We should limit the length of the collector to 1.5 m, otherwise local chaotic convective microflows, in the presence of traction, merge into a continuous steady flow along a transparent coating with the inevitable increase in convective heat losses. The combination of the above structural solutions contributes to an increase in the coefficient of heat transfer from the metal substrate at reduced flow velocities, initial flow temperature and energy efficiency of the collector in general.

The research carried out in this study is a final stage of a comprehensive study for improvement of the efficiency of the process of fruits drying based on development of design and justification of operating modes of a solar dryer, which will reduce a cost of energy resources due to solar energy. Because the developed air solar collector relates to the air solar heating system of a solar dryer. The solar dryer was developed at «Zorya» private farm located in the zone of western Polissya, namely in the city of Korts, Rivne region (Ukraine).

7. Conclusions

1. We developed a new design and described a phased assembly of elements of ASC sample with hermetically sealed and insulated body. We can use it as an additional heating element of a low-temperature source of heat. We defined a number of generalizing dependences for determination of ASC thermal efficiency, namely an effect of the mass flow of air q_a (2) on a temperature difference of a heat-transfer agent t_o (20) and insulation E, on heat production q, (21) and efficiency η of ASC (23).

2. We presented the program and described equipment, devices, experimental plant and methods for the study and field tests of ASC. We verified the adequacy of the results of theoretical and experimental studies. In particular, the average daily temperature of the environment t_{eat} on energy illumination E and the average temperature of a heat-transfer agent t_{aat} on the average daily temperature of the environment t_{eat} in the period of 15 to 20 May, 2018, by t-criterion of the adequacy of the model, which confirmed the reliability of confidence intervals and regression coefficients.

3. We calculated heat engineering characteristics of the air solar collector for the climatic conditions of the city of Korts, Rivne region (Ukraine) based on the experimental data processed. In particular, we determined the maximum values of the efficiency of the solar collector η, (from 65 to 80.6 %).

They are reachable at the temperature of an initial flow of the heat-transfer agent t_o of 30 to 60 °C and the mass flow of air, q_a from 170 to 190 m^3/h. We established that the efficiency of the solar collector and the temperature of an initial flow of the heat-transfer agent t_o fall intensely and nonlinearly at an increase in the mass flow of air, q_a from 190 to 200 m^3/h. Therefore, it is necessary to reduce a flow rate of the heat-transfer agent to q_a – 190 m^3/h. We established that an increase in the insulation level E from 100 to 1,000 W/m^2 increases heat output of the collector q from 320 to 1,260 W and the temperature of the heat-transfer agent at the outlet from the collector t_o from 10 to 60 °C.

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