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**Boris Kotov,**
**Roman Kalinichenko,**
**Anatolii Spirin,**
**Natalia Gudzenko,**
**Andrii Didyk,**
**Mykhailo Zamrii**

# DEVELOPMENT AND RESEARCH OF ENERGY-EFFICIENT HELIO-AIR COLLECTORS FOR DRYING AGRICULTURAL PRODUCTS

*The object of study is the drying of agricultural plant products. Artificial heat drying of agricultural plant products (seeds, fruits, nuts, grain, etc.), as a traditional method of canning and preparation for storage and further processing, usually takes place in dryers of seasonal use. To reduce the consumption of traditional types of organic fuel, it is proposed to use a lightweight portable film solar collector to heat the drying agent in seasonal dryers. A mathematical description of thermal processes in a solar collector is given. To increase the efficiency (degree of air heating), devices are proposed – heat exchange intensifiers. Effective methods of increasing the thermal power of the solar collector based on the use of ring and spiral turbolyzers of the coolant flow and sectional multi-pipe (multi-element) absorber have been theoretically substantiated, experimentally confirmed. The use of these structural elements of the solar collector will increase the thermal performance per unit area of the solar radiation absorber, which will increase the heating of the blunt carrier with 26 °C before 32 °C for tube absorbers and up to 36 °C for sectional absorbers. Maximum specific power of the solar collector with an area 240 m$^2$ – 0.2 kW/m$^2$ with a specific flow rate of the coolant of 23 m$^3$/h·m$^2$. For systems of active ventilation of seed material with specific air flow rates up to 100 m$^3$/h·m$^2$, it is possible to heat the atmospheric air up to 10 °C, which ensures round-the-clock drying of the product. A simplified mathematical model is formulated to intensify parameters according to the data of experiments. The heat transfer coefficients from the film absorber of solar energy to the heated air are determined. The results of the experimental determination of the thermal characteristics of the solar collector and its energy efficiency are presented. As the calculations showed, a further increase in the performance of the tubular solar collector is possible with an increase in the flow rate in the film absorber, which can be realized by reducing the diameter of the pipeline.*

**Keywords:** *solar radiation, heliocollector, drying agent – air, heat transfer, film sleeves, solar absorber.*

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## 1. Introduction

Thermal drying is currently the main method of preparing agricultural plant products for preservation (long-term storage). When drying agricultural products, heated air is mainly used as a heat carrier and drying agent. In the main types of drying technologies, namely drying of seed (grain) materials, fruits, nuts, spicy-aromatic and medicinal plants, there are restrictions on drying modes in relation to the heating temperature of products (and, accordingly, the drying agent), due to the biological value of the products. Therefore, the drying equipment must provide a significant amount of air heated to a temperature of 40–50 °C, which necessitates the use of solar energy to heat the drying agent. Specific features of the use of solar radiation energy for heating moving coolants (air, water, antifreeze, etc.) have led to the emergence of a wide variety of helioheater constructions – solar collectors with different energy and technological efficiency indicators. The unsuitability of certain well-known constructions of solar collectors used for heat supply of domestic and industrial premises for heat supply of drying plants stems from the need to use a significant amount of coolants, which requires large surfaces of solar receiving equipment. The seasonal and short-term operation of drying equipment for drying plant products is the basis for the development of efficient designs, portable (folding), lightweight film solar helio heater for drying plants of plant products. Structural use of film heliocollectors for air heating is usually realized according to the scheme of a pipe-in-pipe heat exchanger.

The inner pipe made of black film (in the form of a sleeve) is used as an absorber-scavenger of solar radiation. The outer shell-sleeve is made of transparent film. The required thermal power is obtained by a set of several elements connected in parallel, but despite the simplicity of tubular film heliocollectors, they allow to obtain thermal power from 130 to 500 W/m$^2$ of the total area. Known collectors have been used for drying grass fodder and grain materials for a long time and have positive characteristics. In the operated heliocollectors of this type, the air is heated at 5–7 °C, which is sufficient for drying grass and grain by active ventilation with significant air consumption (up to 100 $m^3/h \cdot m^2$).

Drying a plant product that contains a significant amount of bound moisture (oilseeds, nuts, certain types of fruits) requires a higher temperature potential of the drying agent up to 30–55 °C and significantly lower air consumption.

Literature [1–7] provides data on experimental studies of the performance characteristics of devices for heating air in drying plants when drying various plant products, which differ significantly in energy characteristics obtained for different weather conditions. Although these data confirm the high efficiency of solar heating in the processes of drying plant products, it is very difficult to determine the heat exchange and heat transfer characteristics of individual structural elements. Therefore, to determine the suitability and evaluate the effectiveness of film heliocollectors for intensive drying, it is necessary to conduct separate studies of steady-state and transient heat transfer processes.

On this topic of this paper shows significant advantages of solar heat supply for technological facilities of agricultural production [1, 2]. Dryers with helioheating of the drying agent are quite effectively used for drying plant products:

– fruit [3, 4];
– cake [5];
– grain [6];
– hay [7].

The variety of constructions for heating the drying agent [8–11] indicates that there is no single approach to creating a universal type of solar heaters. At the same time, the constructions of existing heliodryers [1, 2] are developed for stationary operation, while the supply of raw materials has a short-term seasonal period and is not subject to long-term storage [9–11]. Papers [12, 13] highlight the results of studies of the processes of drying plant products using solar energy to heat the drying agent, without considering the intensification of thermal processes in solar collectors. A mathematical description of the dynamics of heat transfer in a three-channel heliocollector is given in [2, 14], which can be used as a methodological basis for analytical studies of heat transfer processes and determining quantitative assessments of the effectiveness of various methods of heat transfer intensification.

*The aim of research* is to increase the thermal energy efficiency of film-tube-sectional heliocollectors for heating the drying agent in batch dryers by intensifying the heat exchange of air flows with film surfaces absorbing solar radiation.

## 2. Materials and Methods

The conclusions reached as a result of the analysis of our theoretical studies [15] were the basis for substantiating the method of intensifying the heat exchange of air flow with the wall of a tubular pneumatic duct with a plastic surface. The construction of the solar collector taken as the basis of the studied design is a distributing and collecting pneumatic collectors connected to solar heaters in the form of air-pressure tubular translucent sleeves, in the middle of which black absorbing leeves are placed (Fig. 1).

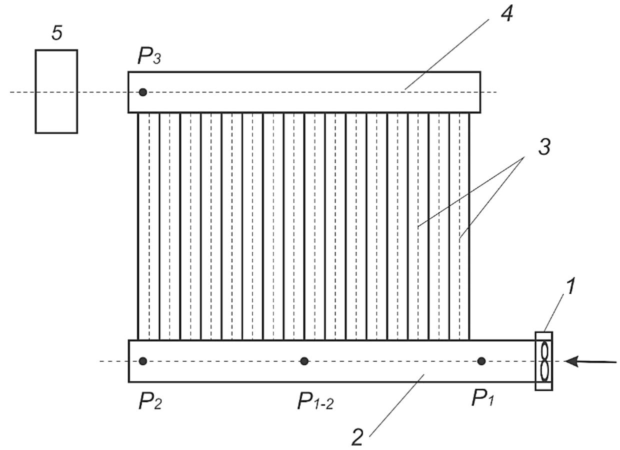


**Fig. 1.** Scheme of a frameless helioheater:
1 – fan; 2 – distributing collector; 3 – absorbing elements; 4 – collecting collector; 5 – drying object

The choice of this construction is based on the simplicity of manufacturing and low hydraulic resistance of the sleeve air ducts.

The main disadvantage of this construction in comparison with frame planar collectors is the reduced area of the absorbing surface (in plan), i. e. in comparison with the area of the entire collector.

One of the promising ways to increase the efficiency of this type of heliocollector is to intensify heat transfer in the channels.

Intensification of heat exchange can be achieved by turbocharging the boundary layer of the heat exchange surface of the pipe by increasing the air velocity in the channel or by discrete turbocharging using protrusions or depressions on the surface of the absorber wall [16], i. e., periodic changes in the cross-section of the channel. The technical implementation of this method is carried out using flexible ring loops or ring-screws on the outer surface of the absorber shell (Fig. 2).

When air is supplied through the channel under a certain pressure, the shell is deformed to form annular and helical vortex zones that become a source of turbulence.

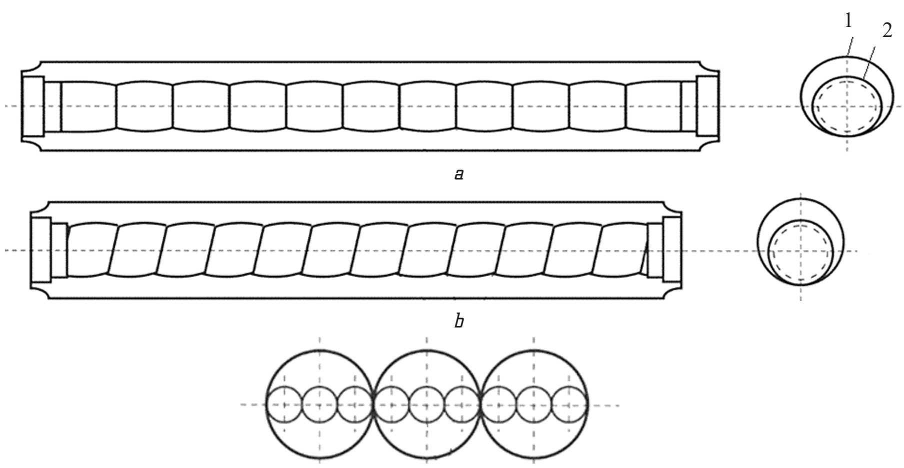


**Fig. 2.** Schematic of heat-absorbing elements: *a*, *b* – with turbolizing elements; *c* – with sectional elements; 1 – transparent shell; 2 – absorbent (black) shell

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The effectiveness of the use of ring and screw turbulators in a circular channel is determined by the ratio [16]:

$$\frac{Nu}{Nu_T} = I - 0.64\left[1-\exp\left(0.35\frac{h}{D}\right)\right]\left(1-0.275\frac{l}{D}\right), \qquad (1)$$

while $h/D=0.16$; $l/D=0.02-2$, where $D$ – channel diameter; $h$ – crimp depth; $l$ –pitch of the loop winding. $d/D=0.88-0.9$, where $d$ – the diameter of the smaller section; $l/D=1.0-1.5$; $Nu=\alpha d_e/\lambda_n$ – Nusselt's criterion; $a$ – heat transfer coefficient; $de=D-d$ – equivalent channel diameter; $\lambda_n$ – air thermal conductivity coefficient.

A theoretical (calculated) and experimental method was used to study the thermal characteristics of the heliocollector.

To reduce the number of experiments at the first stage of research, a theoretical analysis of heat transfer based on the formed mathematical model of the thermal process in a single-section heliocollector channel was used. The mathematical model is identified by the experimental data. After that, the analysis of variants is performed.

The solar collector (air), as a modeling object, is a tubular heat exchanger consisting of two polyethylene sleeves with a diameter of 0.38 m (inner black) and 0.45 m (outer transparent) and a length of 20 m. Solar radiation penetrates the translucent shell and is actively absorbed by the surface of the inner sleeve (Fig. 2). The air is pumped into both channels by a fan and, while moving, receives heat transferred by convection from the heated surface of the inner absorber sleeve. Thermal inertial properties of the solar collector are determined by the thermal capacities of the outer translucent shell (sleeve), the heat capacity of the inner black shell (Fig. 2), and the air in the volume between the shells (sleeves) and in the inner sleeve (absorber). Thus, an object consisting of four containers is considered.

## 3. Results and Discussions

Solar radiation with a flux density $I$ (W/m$^2$) passes through a transparent shell with a transmittance $\alpha_0$ and an absorption capacity $\tau_0$, $t_a$ with a density $I(\alpha_0\tau_0)$ is absorbed by the surface of the outer absorber (part of the heat is transferred by thermal conduction through the shell to the inner surface of the absorber). The heat is transferred from the surface of the absorber by the air moving inside the absorber and in the gap between the absorber and the translucent shell. Part of the heat is transferred through the outer shell to the atmosphere.

The following assumptions were made when formulating the mathematical description. The thermophysical parameters (properties) of the air and the shell material are assumed to be independent of temperature and equal to the average value (in time and coordinate); heat transfer by thermal conductivity and radiation is taken into account by heat transfer coefficients determined experimentally. The air temperature in the cross-section is uniform and distributed along the coordinate in the direction of air movement.

The mathematical description is based on the heat balance of the shells and air.

Consider the elements of heliocollector channels of length $dy$ (Fig. 3).

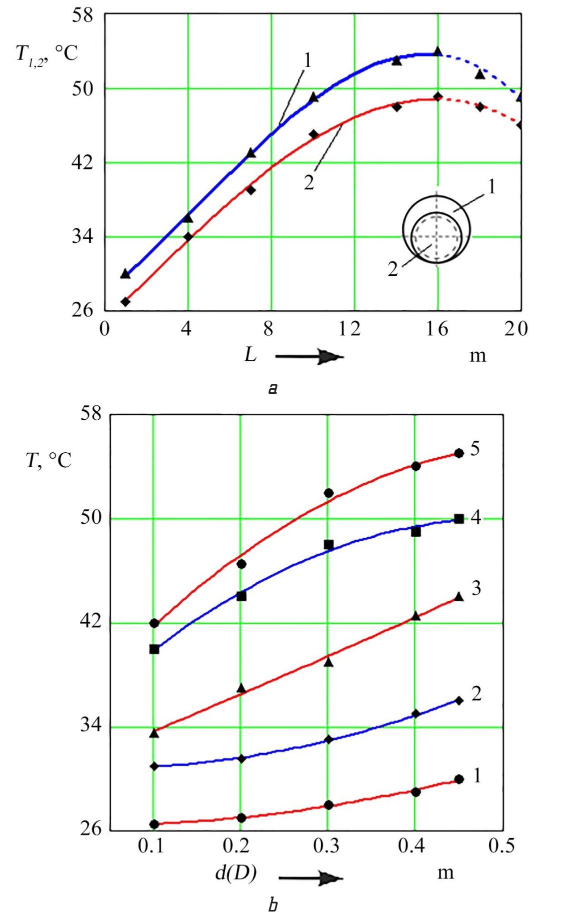


**Fig. 3.** Air temperature distribution: *a* – along the length; *b* – in the cross-section of the solar collector element: 1 – 1 m; 2 – 3.5 m; 3 – 7 m; 4 – 11 m; 5 – 19 m

The heat balance equation that describes the change in the surface temperature of the solar radiation absorber (absorber) for the element $dy$ will be as follows:

$$C_a S_a \rho_a dy dT_a = I(\alpha,\tau_0)dy -$$
$$-\left[\frac{\alpha_1 f_1}{l}(T_a - T_1) - \frac{\alpha_2 f_2}{l}(T_a - T_2)\right]dy d\tau. \qquad (2)$$

The heat balance equation that describes the change in air temperature in the absorber channel $T_a$ for the element $dy$ will be as follows:

$$C_p S_1 \rho_p dy dT_1 = \frac{\alpha_1 f_1}{l}(T_a - T_1)dy d\tau. \qquad (3)$$

The heat balance equation describing the change in air temperature in the containment channel for an element of length $dy$ will be:

$$C_p S_2 \rho_p dy dT_2 =$$
$$= \frac{\alpha_1 f_1}{l}(T_a - T_2)dy d\tau - \frac{\alpha_2 f_2}{l}(T_2 - T_0)dy d\tau. \qquad (4)$$

The heat balance equation describing the change in temperature of the outer shell for an element of length $dy$ is written in the form:

$$C_0 S_0 \rho_0 dy dT_0 = +\left[\frac{\alpha_3 f_3}{l}(T_2 - T_0) + \frac{\alpha_3 f_3}{l}(t_3 - T_0)\right]dy d\tau. \qquad (5)$$

In (2)–(5): $C_a$, $S_a$, $\rho_a$ – specific heat capacity, cross-sectional area and density of the absorber material; $C_p$, $S_a$, $S$ – specific heat capacity, air density and cross-sectional area of the absorber air channel; $S_2$, $S_0$ – cross-sectional area of the air channel between the absorber and the protective shell; $f_1$, $f_2$ – absorbing surface of the absorber and the outer surface of the shell; $a_1$, $a_2$, $a_3$, $a_4$ – heat transfer coefficients of the absorber surface with air in the absorber channel, the absorber surface with air in the channel between the shell and the absorber, the shell surface with air inside the shell and the shell surface with atmospheric air; $T_a$, $T_1$, $T_2$, $T_0$ – the temperature of the absorber surface, air in its channel, in the channel between the shell and the absorber, and the temperature of the shell; $t_3$ – the temperature of atmospheric air.

Expanding the full differentials $dT_a$, $dT_1$, $dT_2$, $dT_0$ in the $y$-coordinate and time $\tau: dT=\frac{\partial T}{\partial \tau}d\tau+\frac{\partial T}{\partial y}dy$, taking into account $dx/d\tau=V$ the air velocity in the channels (in the equation for the absorber and shells $dx/d\tau=0$), and taking the notation $s\rho v=G$ and $s\rho l=m$ in equations (2)–(5), after the transformations let's obtain:

$$I(\alpha_0,\tau_0)F_a=m_a c_a\frac{\partial T_a}{\partial \tau}+\alpha_1 f_1(T_a-T_1)+\alpha_2 f_1(T_a-T_2), \quad (6)$$

$$m_1 c_p\frac{\partial T_1}{\partial \tau}+G_1 C_p l\frac{\partial T_1}{\partial y}=\alpha_1 f_1(T_a-T_1), \quad (7)$$

$$m_2 c_p\frac{\partial T_2}{\partial \tau}+G_2 C_p l\frac{\partial T_2}{\partial y}=\alpha_2 f_1(T_a-T_2)-\alpha_2 f_2(T_2-T_0), \quad (8)$$

$$m_0 c_0\frac{\partial T_0}{\partial \tau}=\alpha_3 f_3(T_2-T_0)+\alpha_4 f_3(t_3-T_0). \quad (9)$$

The initial and boundary conditions are written as follows:

$$\begin{aligned}&\tau=0;\ T_a=T_{a0};\ T_1=T_{10};\ T_2=T_{20};\\&T_0=T_{00};\ y=0;\ T_1=t_3;\ T_2=t_3.\end{aligned} \quad (10)$$

The resulting system of equations describes the process of heat transfer between the surfaces of the heliocollector (absorbing solar energy and transferring part of it to the heated air, part of the energy is dissipated into the environment).

In the problem of analytical analysis of heat transfer in heat exchangers, such a methodological approach is allowed when it is possible to consider heat transfer processes for transient and stationary (steady-state) modes separately. In this case, for the stationary process in equations (6)–(9), the time derivatives are equal to zero:

$$\frac{\partial T_a}{\partial \tau}=\frac{\partial T_1}{\partial \tau}=\frac{\partial T_2}{\partial \tau}=\frac{\partial T_0}{\partial \tau}=0.$$

For the transient process, the dynamic characteristic of the heliocollector is used, provided that there are concentrated parameters, to determine the parameters of the heat carriers at the outlet of the heliocollector.

To study the steady-state regime, the system of equations is simplified (the order of the system is reduced) and used in the form:

$$\begin{cases}I(\alpha_0,\tau_0)=\alpha_1(T_a-T_1)+\alpha_2(T_a-T_2),\\ G_1,C_p\,l\dfrac{dT_1}{dy}=\alpha_2 f(T_a-T_1),\\ G_2C_p\,l\dfrac{dT_2}{dy}=\alpha_2 f(T_a-T_2)-K_0 f_3(T_2-t_3),\end{cases} \quad (11)$$

where $K_0$ – heat transfer coefficient through the protective shell:

$$K_0=\frac{1}{\frac{1}{a_2}+\frac{\delta}{\lambda}+\frac{1}{\alpha_3}},$$

where $\delta$ is film thickness; $\lambda$ is conductivity coefficient of the film.

Determining $T_a$ from equation (10) and substituting the obtained value into equations (11) and (12) after transformations, let's obtain a system of two differential equations:

$$\begin{cases}A_1\dfrac{dT_1}{dy}+a_1T_1-b_1=T_2,\\ A_2\dfrac{dT_2}{dy}+a_2T_2-b_2=T_1.\end{cases} \quad (12)$$

Equations (11), (12) represent the following:

$$A_1=\frac{\alpha_1+\alpha_2}{\alpha_2}\frac{G_1C_pl}{\alpha_1 f};\ a_1=1;\ b_1=\frac{I(\alpha_2,\tau_0)}{\alpha_1},$$

$$A_2=\frac{\alpha_1+\alpha_2}{\alpha_1}\frac{G_eC_bl}{\alpha_2 f};$$

$$a_2=\frac{\alpha_1 f(\alpha_1+\alpha_2)+K_0f_3(\alpha_2+\alpha_2)+\alpha_2^2 f}{\alpha_1\alpha_2 f},$$

$$b_2=\frac{K_0 f_3(\alpha_1\alpha_2)}{\alpha_2 f}t_3+\frac{I(\alpha_0\tau_0)}{\alpha_1}.$$

The solution of equations (11), (12) with regard to $T_1(y)$ and $T_2(y)$:

$$A\frac{d^2T_1}{dy^2}+B\frac{dT_1}{dy}+cT_1=D_1, \quad (13)$$

$$A\frac{d^2T_2}{dy^2}+B\frac{dT_2}{dy}+cT_2=D_2, \quad (14)$$

where $A=A_1A_2$; $B=a_1A_2+a_2A_1$; $C=A_1A_2-1$; $D_1=b_2+b_1a_2$; $D_2=b_1+b_2a_1$.

The solution of inhomogeneous equations (13) and (14) under the boundary conditions: $y=0$, $T_1=t_3$, $T_2=t_3$ is obtained as follows:

$$T_1(y)=\frac{t_3C-D_1}{C(r_1-r_2)}\left[r_1e^{r_2y}-r_2e^{r_1y}\right]+\frac{D_1}{C}, \quad (15)$$

$$T_2(y)=\frac{t_3C-D_2}{C(r_1-r_2)}\left[r_1e^{r_2y}-r_2e^{r_1y}\right]+\frac{D_2}{C}, \quad (16)$$

where $r_{1,2}=\frac{-B\pm\sqrt{B^2-4AC}}{2A}$ – roots of the characteristic equation.

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Equations (15) and (16) determine the temperature distribution in the collector channels along the length.

To identify the parameters of the model: to calculate the heat transfer coefficients $a_1$, $a_2$ and $k_0$ experiments were conducted to determine (measure) the air temperature in the channels at different distances from the inlet section of the channels (Fig. 4).

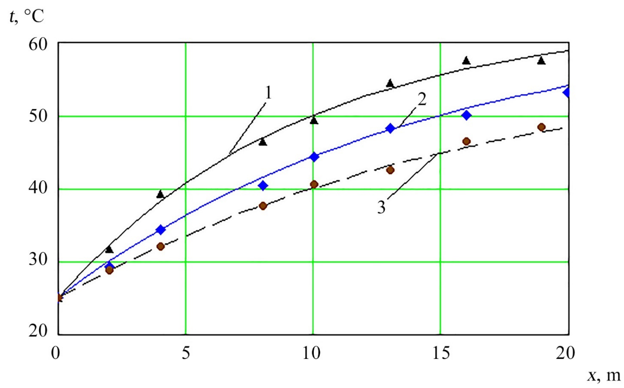


**Fig. 4.** Temperature distribution along the length of the collector (1 – α=9; 2 – α=6; 3 – α=3)

After measuring the air temperature $t_i$ at different collector lengths $y_i$ and substituting the values of $t_i$ and $y_i$ into equations (15) and (16), let's obtain the systems of equations:

$$t_i(y_i)=\frac{t_0C-D_1}{C(r_1-r_2)}\left[r_1e^{r_2y_i}-r_2e^{r_1y_i}\right]+\frac{D_1}{C}, \qquad (17)$$

$$t_i(y_i)=\frac{t_0C-D_2}{C(r_1-r_2)}\left[r_1e^{r_2y_i}-r_2e^{r_1y_i}\right]+\frac{D_2}{C}, \qquad (18)$$

where $i=1, 2, 3, 4, 5...n$; $j=1, 2, 3, 4, 5...m$.

Solving (using the Mathcad computer program) the systems of equations (17) and (18), let's define the complexes $A$, $B$, $C$, $D_1$, $D_2$, and reveal the symbolic expressions for defining complexes:

$$A=\left(\frac{a_1+a_2}{a_1a_2f}\right)^G\frac{G_1G_2C_p^2l^2}{f};\; D_2=2\frac{I}{a_1}+\frac{K_0F_0(a_1+a_2)}{a_2f}t_3,$$

where $C=A-1$ – the coefficients $a_1, a_2, a_3, K_0$ of the system of equations (6)–(9) or (11). The identified systems of equations can be used to analyze the outflow of constructive $f_1, f_2, F_0, l$ and operational $G_1, G_2, G_1/G_2$ parameters.

The results of the research are shown in the graphs (Fig. 3), which show the distribution of air temperature along the length of the channels (Fig. 3, *a*) and in the cross section (Fig. 3, *b*). The temperature in the cross-section of the channels is extremely uneven. In the channels between the shells, the temperature is higher than inside the absorber's absorption channel, which is explained by the air flow through this section. The graphs (Fig. 4) show the approximated dependences of the temperature distribution according to equations (15) and (16). The temperature (average) along the length of Θ=61 °C of the absorber surface was determined and the heat transfer coefficients were calculated: in the inner channel – 3.6 W/m$^2$k, in the outer channel – 2.08 W/m$^2$k which exceeds the values of the heat transfer coefficients calculated by the criterion equation $Nu=0.018\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.33}$, which can be explained by the presence of radiation heat removal by the upper part of the absorber (from the upper half-cylinder to the lower one).

The results of calculating the air temperature distribution in the channels at different values of the surface heat transfer coefficients are shown in Fig. 5. The analysis of the graphs shows a significant effect of the heat transfer coefficient on the air temperature and the real possibility of increasing the degree of air heating in the heliocollector by increasing the value of the heat transfer coefficient.

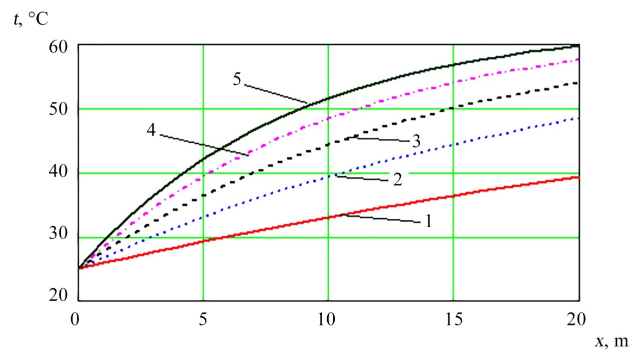


**Fig. 5.** Temperature distribution along the length of the collector at different values of the heat transfer coefficient (1 – α=2; 2 – α=4; 3 – α=6; 4 – α=8; 5 – α=10)

To intensify the heat exchange between the wall of the "pipeline" and the moving medium, the surface of the film "elastic pipeline is marked with pinches that form annular protrusions on the inner surface that turbocharge the boundary layer".

The turbolization parameters are calculated by the empirical formula:

$$Nu_mNu_0\left[1-0.64\left[1-e^{-0.35\frac{n}{\mu}}\right]\left(1-0.274\frac{S}{01}\right)\right], \qquad (19)$$

where $Num=\frac{a\cdot d}{\lambda_n}$; $Nu_0=\frac{\alpha de}{\lambda n}$ – Nusselt criterion for a smooth absorber and a turbocharged one; $d_1$, $d_e$ – diameter of the absorber and equivalent diameter, $h$ – height of the annular projection $h=(d-d_e)$; $S$ – pitch of the ring turbines.

The diagrams (Fig. 6, 7) show the experimentally obtained dependences of the excess air temperature at the outlet of the heliocollector element on the specific air flow rate for a smooth-tube absorber and an absorber with spiral turbines. The temperature increase (maximum) at a flow rate of 23 m$^3$/h·m$^2$ is 4–5 °C. This made it possible to increase the thermal capacity of the solar collector by 18–19 % (0.18 kW/m$^2$), which corresponds to a collector capacity of 43.2 kW.

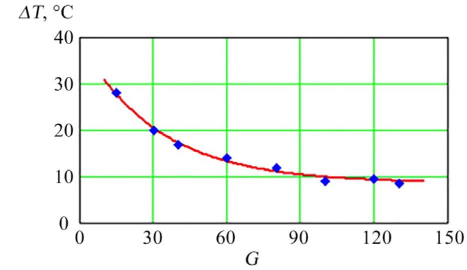


**Fig. 6.** Dependence of air temperature rise on specific flow rate for a smooth-tube absorber

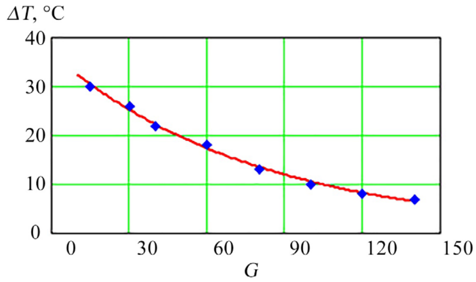


**Fig. 7.** Dependence of air temperature rise on specific flow rate for an absorber with spiral turbochargers

Calculations have shown that further improvement of the tubular heliocollector performance is possible with an increase in the flow rate in the film absorber, which can be realized by reducing the diameter of the "pipeline". But this will reduce the area of solar radiation absorption. To avoid this contradiction, it is proposed to make the absorber in the form of a package of several tubular film channels (Fig. 2, *c*). With this construction of the absorber, the absorption area of solar radiation is maximized and it is possible to significantly increase the coolant velocity in the channels at constant specific flow rates. The increase in the coolant velocity is accompanied by an increase in pressure losses and an increase in energy consumption for coolant transportation.

To determine the rational speed and number of tubular elements in the protective shell, calculations were performed using a system of obvious balance equations:

$$\begin{cases} \Delta P_a = \Delta P_K, \\ L = L_a + L_K, \end{cases} \tag{20}$$

where $L_a$, $L_K$, $L$ – volumetric flow rates of the coolant in the absorber channel, intermediate channel and total; $\Delta P_a$, $\Delta P_k$ – hydraulic resistance of the absorber channel and intermediate channel.

The system of equations can be rewritten in expanded form [15]:

$$\begin{cases} \xi_a \dfrac{V_a^2}{d_a} = \xi_K \dfrac{V_K^2}{d_K}, \\ 1.273L = Vd_a^2 + V_K d_K^2, \end{cases}$$

where $\xi = 0.316 R_e^{-0.25}$, $\xi_a, \xi_K$ – coefficients of resistance to coolant movement in the absorber and channel; $V_a, V_K$ – speed of coolant movement in the channels; $d_a$ – diameter of the tubular element of the absorbers; $d_k$ – equivalent diameter of the intermediate channel; $\mathrm{Re} = (d \cdot v)/v$ – Reynolds criterion; ν – coefficient of kinematic viscosity of the coolant.

The results of solving the system of equations (20) determined the optimal number of tubular elements, namely three sections at a coolant velocity in the absorber channels of up to 3.5 m/s. The graph in Fig. 8 shows the dependence of the excess air temperature (heating temperature) on the specific flow rate, the value of which is 36 °C at a specific flow rate 23 $\mathrm{m^3/(h \cdot m^2)}$ that provides a thermal power of 48 kW and is 17 % higher than the power of a heliocollector with spiral turbulators.

Four constructions of film tubular absorbers placed in a translucent shell of the same size (0.45 m) were tested in production conditions. The overheating of air at the outlet of the solar collector element was determined, the coefficient of efficiency (thermal) was calculated depending on the volume flow rate in the range (20 m 200 m$^3$/hm$^2$).

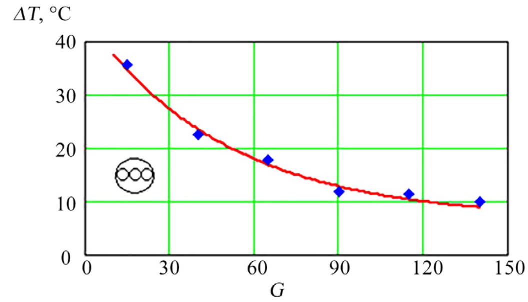


**Fig. 8.** Dependence of air temperature rise on specific flow rate for a sectional absorber

The processing of experimental data for all types of absorber tests allowed to summarize the dependence of the degree of heating $\Delta T$ on the specific air flow rate and the thermal efficiency on the degree of heating by the following dependencies:

$$\Delta T(L) = a_2 \exp(-b_2 L), \tag{21}$$

$$\eta(\Delta T) = a_1 \exp(-b_1 L). \tag{22}$$

The values of the approximation coefficients are shown in Table 1.

**Table 1**

Values of approximation coefficients

| Designation of coefficients | Type of solar radiation absorber | | | | |
|---|---|---|---|---|---|
| | Pipe smooth | hinged turbocharger | spiral turbocharger | Planar | Sectional |
| $a_1$ | 96.4 | 129.3 | 99.7 | 68.54 | 114.8 |
| $b_1 \cdot 10^{-2}$ | 5.95 | 7.2 | 6.4 | 6.2 | 6.44 |
| $a_2$ | 23.6 | 26.78 | 27.9 | 22.1 | 36.1 |
| $b_2 \cdot 10^3$ | 8.46 | 7.94 | 8.8 | 9.95 | 9.1 |

The obtained formulas can be used to calculate solar heat generators and dryer parameters.

The analysis of the results shows that the biggest thermal efficiency is achieved by a heliocollector with a sectional absorber. The lowest performance is observed for the plane absorber. The use of turbolizing clamps of the tubular surface intensifies heat transfer less than the sectional absorber, but has less hydraulic resistance.

*Wartime* imposed significant restrictions on energy conservation of the national economy, including agricultural production.

*The use* of solar air heaters for drying plant products will reduce the use of "traditional" types of energy.

*Further research* will be aimed at solving the problem of increasing the efficiency of solar heaters by increasing the specific area of absorbers.

## 4. Conclusions

Developed (formulated) a mathematical model of the temperature regime of a film air-supported heliocollector for heating the drying agent in installations for various purposes (active ventilation with heated air, drying of

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thermostable materials (at $\Theta_m \leq 41$ °C), heating of atmospheric air in front of a heat generator, etc.).

The analytical solutions were used to identify the model from the experimental data: the parameters that cannot be measured directly in experiments (heat transfer coefficients of the heat-receiving surface with the coolant).

Effective ways to increase the thermal power of a heliocollector based on the use of annular and spiral turbochargers of the coolant flow and a sectional multi-tube (multi-element) absorber are theoretically substantiated and experimentally confirmed.

The use of these elements of the heliocollector construction will increase the thermal performance per unit area of the solar radiation absorber, which will increase the heating of the heat carrier from 26 °C to 32 °C for the tube absorber and up to 36 °C for the sectional absorber.

The maximum specific power of a heliocollector with an area of 240 $m^2$ – 0.2 $kW/m^2$ at a specific coolant flow rate of 23 $m^3/h{\cdot}m^2$. For systems of active ventilation of seed material with specific air flow rates up to 100 $m^3/h{\cdot}m^2$ it is possible to heat the atmospheric air up to 10 °C, which ensures round-the-clock drying of the product.

## Conflict of interest

The authors declare that they have no conflict of interest in relation to this study, including financial, personal, authorship, or any other, that could affect the study and its results presented in this article.

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The study was conducted without financial support.

## Data availability

The manuscript has no associated data.

## Use of artificial intelligence

The authors confirm that they did not use artificial intelligence technologies when creating the presented work.

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***Boris Kotov,** Doctor of Technical Sciences, Professor, Department of Agricultural Engineering and System Engineering names Mykhaila Samokisha, Institute of Higher Education "Podilskyi State University", Kamianets-Podilskyi, Ukraine, ORCID: https://orcid.org/0000-0001-6369-3025*

***Roman Kalinichenko,** PhD, Associate Professor, Departments of Electric Power Engineering, Electrical Engineering and Electromechanics, National University of Life and Environmental Sciences, Kyiv, Ukraine, ORCID: https://orcid.org/0000-0001-9325-1551*

***Anatolii Spirin,** PhD, Associate Professor, Department of Engineering and Agronomy, Separated Structural Unit "Ladyzhyn Professional College of Vinnytsia National Agrarian University", Ladyzhyn, Ukraine, ORCID: https://orcid.org/0000-0002-4642-6205*

***Natalia Gudzenko,** PhD, Associate Professor, Department of Economic, Separated Structural Unit "Ladyzhyn Professional College of Vinnytsia National Agrarian University", Ladyzhyn, Ukraine, ORCID: https://orcid.org/0000-0002-0978-4257*

*✉**Andrii Didyk,** PhD Student, Department of Engineering Mechanics and Technological Processes in Agriculture, Vinnytsia National Agrarian University, Vinnytsia, Ukraine, e-mail: anddidyk99@gmail.com, ORCID: https://orcid.org/0000-0002-0524-0017*

***Mykhailo Zamrii,** PhD Student, Department of Labor Protection and Biotechnical Systems in Animal Husbandry, Vinnytsia National Agrarian University, Vinnytsia, Ukraine, ORCID: https://orcid.org/0000-0002-9433-6714*

*✉Corresponding author*