Розроблено нову конструкцію повітряного геліоколектора для геліосушарки фруктів, що включає подвійне засклення та селективну поверхню, виготовлену з тонкої металевої підкладки на його днищі з вхідними та вихідними отворами. Встановлено, що для підкладки із подвійного засклення необхідно використовувати скло з тепловідбиваючим покриттям твердого типу K-glass з коефіцієнтом випромінювання ε=0,1...0,15. Це дозволяє одержати максимально великий спектр потоку прямих сонячних променів, які опромінюють поверхню поглинальної пластини та знизити розсіяну складову випромінювання, що забезпечує підвищення ефективності колектора.

Визначено закономірності впливу зміни витрати теплоносія, перепаду температур та інтенсивності радіаційного випромінювання на потужність геліоколектора. Розроблено модель процесів теплообміну, що відбуваються у повітряному геліоколекторі. Наведено методику оцінки тепловтрат повітряного геліоколектора із пасивним використанням сонячної енергії.

Встановлено, що на теплопродуктивність повітряного колектора Q=117...480 Вт суттєво впливає енергетична освітленість E, яка становить від 377 до 1223 Вт/м^2. З'ясовано, що застосування неселективно-поглинаючої поверхні у повітряному геліоколекторі за малого рівня інсоляції E=377 Вт/м^2 дає змогу збільшити ККД на η=70,7 % за селективний, а при великій енергетичній освітленості у E=1000 Вт/м^2 навпаки малий η=54,6 %. Це дає змогу пояснити, як відбувається перерозподіл співвідношень максимальної поточної теплової потужності (NГК=48,8...100 Вт) та ККД геліоколектора.

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

Ключові слова: повітряний геліоколектор, прозоре покриття, температурне поле, тепловий потік, теплообмін, тепловтрати

UDC 631.364:621.311.243
DOI: 10.15587/1729-4061.2018.132090

SUBSTANTIATION OF PARAMETERS AND OPERATIONAL MODES OF AIR SOLAR COLLECTOR

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1. Introduction

There is an ambiguous identification of senses of terms "solar collector" or "helio-collector" in many cases. However, this is one and the same device of the solar power plant designed to collect solar radiation energy in visible and infrared spectrum and to convert it into thermal or electric energy.

There are many varieties of flat or tubular solar-thermal collectors used for hot water supply and heating of industrial premises with known high technical and economic characteristics in the Ukrainian market today. However, their application is unprofitable due to the high cost of collector elements and expensive and complex operation and maintenance conditions.

It is expedient to use an air solar collector (SC) for the process of fruit drying in a solar dryer. There are no mass use of solar dryers and air SC in the Ukrainian market currently. Typically, single explorers produce air SC for solar dryers in single sample with approximate technical and economic characteristics for flat solar-thermal collectors. Therefore, known variants of the design of air SC require modification and improvement in order to increase efficiency of operation under conditions of temperate continental climate of Ukraine and to reduce capital and operation costs.

The main problem is also a lack of scientific and methodical principles for selection and calculation of an air SC for a solar drying plant, which requires additional research and development of scientifically substantiated engineering techniques. In this regard, the subject of the study devoted to substantiation of structural and technological parameters of an air solar collector is relevant.

2. Literature review and problem statement

There are many theories and options for substantiation of the design and technological parameters and improvement of efficiency of an air SC. Particularly, authors of paper [1] developed an engineering methodology for calculation of the required area of SC for a solar drying plant. Authors recommend increasing the area of an absorbing surface of heat exchange and convective heat exchange, due to intensification of mixing of air masses by providing the inequality of a surface of an absorbing plate. However, calculation of the area of SC does not take into consideration a flow of heat loss through enclosing elements. This does not make it possible to calculate transient operation modes and performance.

Paper [2] proposed to use additional air collectors with parabolic-cylindrical concentrators for a solar dryer of a tunnel type. However, calculation of a heat output of an air collector does not take into consideration an amplification coefficient of a solar energy flow, which does not give possibility to describe a change in the maximum energy illumination of a horizontal surface of an air collector relatively to the daily one.

Scientists formulated the actual task of increasing thermal efficiency and performance index and developed a new design of an air collector and proposed a method for calculation of its design and technological parameters in work [3]. The described technique is very general and does not correspond to modern achievements of solar energy. Particularly, they calculated heat flows through light transparent enclosures of a collector according to recommendations of the construction climatology [3]. There are tables of standard averaged energy flows from direct and scattered radiation in the clear sky and, based on them – an average energy illumination of a perception surface taking into consideration the cloudiness coefficient. In most application tasks, we can replace such calculations with computational results for typical orientation surfaces given on NASA's website [4].

Authors of work [5] proposed to calculate the area of SC as a ratio of its daily heat production to a sum of values of heat flows incoming through an enclosure of an absorbing surface, as well as from insulated walls. However, the proposed method does not take into consideration components of radiation (radiant) heat transfer resistances and a heat loss coefficient (heat transfer) of $K_{SC}=1/R_{SC}$.

The general theory of SC presented in a literature review [6] relates to flat and tubular solar-thermal collectors with a liquid heat-transfer agent. According to the above methodology, the following equation for performance index fully describes efficiency of a SC unit area in a wide range of changes in input and output parameters:

$$\eta = F' \cdot \left(\tau \cdot \alpha - U_L \cdot \frac{\bar{t} - t_{env}}{E} \right), \qquad (1)$$

In accordance with the above scientific and methodical principles, authors agreed and established input parameters of a collector during substantiation of the method for calculation of SC on a base of specialized reference methodological literature sources. Particularly, $\bar{t} = (t_{\text{in}} + t_{\text{out}})/2$ is the average temperature at an inlet and an outlet of a collector, °C [7]; t_{env} is the ambient temperature, °C; F is the coefficient of heat transfer of a light-absorbing panel [8]; $\tau \cdot \alpha$ is the optical performance index of a collector equal to a product of transmission of transparent coating τ and an absorption coefficient α of a coating of a perception panel [9]; U_L is the total coefficient of heat loss of a collector, W/(m^2·°C) [10]; E is the energy illumination of perceiving surface, W/m^2.

Here, authors determine numerical values of the coefficients $F(\tau\alpha)$ and FU_L as a result of tests. Exposure to standard solar energy irradiation AM 1.5 regulates conditions and order of the tests [11].

Particularly, dark tests determine a product FU_L. Dark tests mean passing of a hot heat-transfer agent through a collector, in the case of an air collector – air.

$$F'U_L = \frac{G_{\text{sp}} \cdot c_p \cdot (t_{\text{in}} - t_{\text{out}})}{(\bar{t} - t_{\text{env}})}, \qquad (2)$$

where G_{sp} is the specific (per unit surface area) air consumption, kg/s·m^2; c_p is the specific heat capacity of air kJ/kg·K.

Instead, a graphical extrapolation to the zero of a functional dependence of the current performance index (determined by the ratio of heat output to energy illumination $\eta = Q_u/E$) of a complex

$$\eta = \frac{Q_u}{E} = \frac{G_{sp} \cdot c_p \cdot (t_{in} - t_{out})}{E} = = f \cdot \left(\frac{\bar{t} - t_{env}}{E} \right) = f \cdot (t*). \qquad (3)$$

determines a product $F(\tau\alpha)$.

But for air collectors, where temperature of a heat-transfer agent at the input, as a rule, is equal to the ambient temperature, a product $F(\tau\alpha)$ is determined by

$$F'(\tau \cdot \alpha) = \frac{F' \cdot U_L \cdot (t_{out} - t_{in})}{E \cdot \left[1 - \exp\left(\frac{F' \cdot U_L}{G_{sp} \cdot c_p} \right) \right]}. \qquad (4)$$

However, in the proposed method, consumption of a heat-transfer agent in a collector is included in an exponent indicator, which requires special attention to the accuracy of measurement of flow speed, which is commensurate with speed of the wind. Because additional scales are used during a use of a collector for other angles of inclination of a horizontal plane of a collector β: 0°; 45°; 60° and 75° and temperature limits within 10° on the coordinate axes. This, coupled with an uneven cost of a scale division, greatly increases complexity of the process, and worsens the accuracy of determining the Rayleigh number and Nusselt number.

Work [12] substantiates structural and technological parameters of an air collector for a solar dryer of a mine type. Particularly, it proposes engineering scientific-and-methodical principles for determination of the optimal area of a metal subtract for an absorbing surface and raising of temperature of a heat-transfer agent in a collector due to additional irradiation of a solar dryer. However, it neglects the method of evaluation of heat loss of SC and links it with a change in physical parameters of the environment. We cannot apply the proposed method, because it has no specification (estimation), which covers a complex system – "radiation generator – an object of drying – kinetics and dynamics of a drying process – drying plant." It does not provide a solution for the corresponding tasks during diagnosis and forecasting of properties of an object of drying or heating in the process of convective and convective-radiation drying.

Authors of paper [13] developed an engineering methodology for calculation of the main characteristics of a plane solar-thermal collector (the iteration method). A given method reveals features of a structure of a flat solar-thermal collector. Given equations give possibility to calculate heat exchange characteristics in the process of falling of solar energy to a solar plant. But during design calculations, authors choose the most likely conditions for a solar collector and non-temperature dependent thermophysical properties of materials of which a collector is made. This means that they neglect contact thermal supports on both sides of an insulating layer of a collector. This does not make it possible to calculate the amount of energy consumed and discharged from SC, as well as a quality of this energy, that is, its ability to be converted into useful work, that is, values of efficiency of a collector.

Authors of publication [14] suggest using a flat mirror concentrator in the morning and evening in order to increase thermal efficiency of an air SC. But in many cases, effectiveness of implemented installations does not match expectations. The reason is the unjustified choice of constructive solutions between a flat mirror concentrator and a horizontal absorbing plane of SC. That is due to that the authors of publication [14] do not take into consideration peculiarities of generation of sun radiation on a horizontal absorbing plane of SC under standard lighting conditions other than a natural solar flow [10, 11]. Therefore, the main disadvantage of the developed air SC as a source of heat is instability of output power due to daily unevenness of solar energy flow with a casual component of heat flows during weather variability.

The developed methods for selection and calculation of air SC are suitable for countries with hot climates under laboratory conditions or simulation models during computer simulation. Known designs of air SC require modification and improvement in order to increase efficiency of operation under conditions of temperate continental climate of Ukraine and reduction of capital and operation costs.

Thus, the decisive aspect for deciding on using air SC in a solar dryer is the substantiation of its optimal structural and technological parameters.

3. The aim and objectives of the study

The objective of this study is to improve the efficiency of drying process in a solar dryer based on the development of design and substantiation of structural and technological parameters of SC, which will ensure reduction of energy costs due to solar energy.

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

– development of a design of an air SC and substantiation of its design and technological scheme;

– development of a methodology for calculation of basic structural and technological parameters of an air SC;

– estimation of heat losses of an air SC.

4. Materials and methods for the substantiation of structural and technological parameters of a solar dryer

4. 1. Studying the distribution of temperatures along a solar collector

A flow of heated air performs simultaneously two functions in a solar dryer – heating of dried material and removal of moist air outside. At an early stage of the process, intensity of evaporation is high even at small temperatures and multiplicity of air exchange and low temperature of a heat-transfer agent is decisive. Accordingly, SC operates in a mode closer to the maximum heat output, which numerically characterizes energy performance index. Instead, activation of diffusion processes within dried material requires higher temperature and low multiplicity of air exchange at the final drying stage. Therefore, the algorithm for control of operation of a solar dryer is to provide an optimal ratio between flow temperature and its consumption.

Taking into consideration variability of parameters of an incoming solar energy flow, SC operates in both close to stationary and transient modes during daylight hours. If heat capacity of SC is small, then we can neglect a contribution of transition states to heat output in the first approximation. Therefore, its performance characteristics correspond to the steady state mode, which is determined by the equilibrium between input and output energy flows. For convenience, we took the area of a light-receiving surface of SC equal to one and established thermal balances between the specific energy flows.

Input heat flow is a sum of absorbed solar energy q_s and reduced mass of an input flow of external air q_{m}:

$$q_{in}=q_s+q_m. \tag{5}$$

We determine absorbed solar energy q_s, W/m^2:

$$q_s=\eta_0\cdot E, \tag{6}$$

where E is the energy illumination of the surface of SC, W/m^2; $\eta_0=\tau\cdot\alpha$ is the optical performance index of a collector equal to the product of transmission coefficients of translucent coating τ and absorption of rays by perceiving surface α.

We determine mass of the input flow of external air q_{m}, W/m^2:

$$q_m=c_p\cdot G_m\cdot t_{in}, \tag{7}$$

where c_p is the specific heat capacity of air at constant pressure, kJ/(kg·°C); G_m is the mass air consumption, kg/m^2·s; t_{in} is the temperature of an input air flow, °C.

Instead, output flows consist of useful, carried by the mass of heated air q_{ha} (heat-transfer agent) and a flow of loss q_l to the environment equal to a sum of convective q_c and radiation q_r heat exchange:

$$q_{ha}=G_m\cdot c_p\cdot t_{out}, \tag{8}$$

$$q_1=q_c+q_r. \tag{9}$$

Under stationary mode, temperature distribution along a collector is constant over time, and therefore the condition of the equilibrium of input and output flows is saved for each element of its width b and its length dy. In this case, an increase in temperature Δt in each element is proportional to its gradient $\Delta t/\Delta y$, which, at a boundary, goes into derivative dt/dy. For each of them, an elemental flow of heat loss to the environment is proportional to a surface area bdy and temperature differences inside a collector t and the environment t_{env}:

$$dq_1=K\cdot(t-t_{\mathrm{env}})\cdot bdy, \tag{10}$$

where K is the effective coefficient of heat transfer of a collector element, W/m^2·°C; t, t_{ha} is the temperature, respectively, inside a collector and the environment, °C; b is the absorber width, m; dy is the absorber length, m.

The following differential equation describes the thermal equilibrium of each section [8]:

$$\eta_0 \cdot E \cdot bdy = \\ = K \cdot (t - t_{\text{env}}) \cdot bdy + G_{\text{m}} \cdot c_p \frac{dt}{dy} dy. \quad (11)$$

After a standard transformation

$$\frac{dt}{dy} + \frac{K \cdot b}{G_m \cdot c_p} \cdot t - \frac{\eta_0 \cdot E \cdot b + b \cdot K \cdot t_{env}}{G_m \cdot c_p} = 0. \quad (12)$$

The general solution to the equation is as follows:

$$t = \frac{\eta_0 \cdot E}{K} + t_{env} - \\ -\left(\frac{\eta_0 \cdot E}{K} - t_{env} + t_{in}\right) \exp\left(-\frac{K \cdot b}{G_m \cdot c_p} \cdot y\right). \quad (13)$$

In the absence of a supply of a heat-transfer agent (G_m=0), the third summand of the formula (13) is zero, so a collector is heated to the maximum temperature, which is called equilibrium and determined:

$$t_{\text{e}} = \frac{\eta_0 \cdot E}{K} + t_{\text{env}}, \quad (14)$$

where t_e is the equilibrium temperature of the input air flow, °C.

The excess equilibrium temperature is called the difference between temperatures of a heat-transfer agent and the environment:

$$\theta = t_{\text{e}} - t_{\text{env}} = \frac{\eta_0 \cdot E}{K}, \quad (15)$$

where θ is the average logarithmic difference of temperature of the input air flow, °C.

Taking these notations into consideration, formula (9) takes the following form:

$$t = t_e - (t_e - t_{in}) \exp\left(-\frac{K \cdot b}{G_m \cdot c_p} \cdot y\right), \quad (16)$$

$$\theta = \theta_p - (\theta_p - \theta_{in}) \exp\left(-\frac{K \cdot b}{G_m \cdot c_p} \cdot y\right). \quad (17)$$

If we assume temperature of an output flow equal to temperature of the last element, which formally corresponds to the replacement $y=L$, and accept a notation for reduction of the recording

$$N_{SC} = \frac{K_{SC} \cdot b_{SC} \cdot L_{SC}}{G_m \cdot c_p} = \frac{K_{SC} \cdot F_{SC}}{G_m \cdot c_p}, \quad (18)$$

where F_{SC} is the area of a perceiving surface of SC, m; K_{SC} is a coefficient of heat loss by collector, W/m^2·°C; b_{SC} is the width of SC, m.

Then:

$$t_{\text{out}} = t_{\text{e}} - (t_{\text{e}} - t_{\text{in}}) \exp(-N_{\text{SC}}). \quad (19)$$

The dimensionless exponential multiplier $N_{SC}=q_l/q_{ha}$ – depends on a temperature mode of a collector. The process of heat transfer from a heated panel to a heat-transfer agent goes always due to a temperature difference, so a heat loss flow is never equal to zero. Instead, a flow of energy to a heat-transfer agent cannot exceed a certain maximum $q_{\max}$ that cannot exceed an input one, which is equivalent to the following condition:

$$q_{\max} \le q_s = q_{\text{ha}} + q_1. \quad (20)$$

Therefore, we can estimate efficiency of a light-absorbing panel as a ratio of a useful flow to the maximum possible one:

$$\xi = \frac{q_u}{q_{ha} + q_{in}}, \quad (21)$$

where q_u is the maximum heat flow of a heat-transfer agent between double glazing and an absorbing surface, W/m^2.

Taking into consideration this ratio, formula (15) for a collector of a unit length L=1 takes the following form:

$$t_{\text{out}} = t_{\text{e}} - (t_{\text{e}} - t_{\text{in}}) \exp(-N_{\text{SC}} \cdot \xi) \quad (22)$$

or

$$\theta = \theta_{\text{p}} - (\theta_{\text{p}} - \theta_{\text{in}}) \exp(-N_{\text{SC}} \cdot \xi), \quad (23)$$

where θ_p, θ_{in} are the average logarithmic differences in temperature of the input and output air flows, °C.

We can obtain a formula for calculation of a temperature increase in SC if we subtract input temperature t_{in} from the right and the left:

$$\Delta t = t_{out} - t_{in} = (t_e - t_{in}) \cdot \left(1 - e^{-N_{SC} \cdot \xi}\right), \quad (24)$$

$$\Delta t = \vartheta_{out} - \vartheta_{in} = (\vartheta_p - \vartheta_{in})\left(1 - e^{-N_{SC} \cdot \xi}\right), \quad (25)$$

where υ_{in}, υ_{out}, υ_e are, respectively, input, output and equilibrium air speed, m/s;

We determine a useful specific heat output of a collector

$$q_k = G_m \cdot c_e \cdot (t_e - t_{in}) \cdot \left(1 - e^{-N_{SC} \cdot \xi}\right), \quad (26)$$

or

$$q_k = G_m \cdot c_e \cdot t_e \left(1 - e^{-N_{SC} \cdot \xi}\right) - G_m \cdot c_e \cdot t_{in} \left(1 - e^{-N_{SC} \cdot \xi}\right), \quad (27)$$

$$q_k = G_{\text{m}} \cdot c_{\text{e}} \cdot \frac{\eta_0 \cdot E}{K_{\text{SC}}} \left(1 - e^{-N_{SC} \cdot \xi}\right) - G_{\text{m}} \cdot c_{\text{e}} \left(1 - e^{-N_{SC} \cdot \xi}\right) \cdot t_{in}. \quad (28)$$

At the inlet of an air collector, flow temperature is equal to the ambient temperature $t_{in}=t_{env}$, and at the output – almost equal to temperature of the last element of the air $t_{in} \approx t_{out}$. Considering that the second summand of the right side (28) expresses a flow of heat loss, it is also valid for a collector in general if we substitute a difference between average temperature and the environment instead of t_{in}:

$$t_{\text{in}} \to t_{\text{av}} - t_{\text{env}}, \quad (29)$$

where t_{av} is the average equilibrium temperature of an input air flow with the environment, °C.

Under the substitution, formulas for heat efficiency and performance index of a collector take the form:

$$q_{SC} = G_m \cdot c_e \frac{\eta_0 \cdot E}{K_{SC}}\left(1-e^{-N_{SC}\cdot\xi}\right) - -G_m \cdot c_e\left(1-e^{-N_{SC}\cdot\xi}\right)\cdot(t_{av}-t_{env}), \quad (30)$$

$$\eta_{SC} = \frac{q}{\eta_0 \cdot E} = \frac{G_m \cdot c_e}{K_{SC}}\left(1-e^{-N_{SC}\cdot\xi}\right) - -\frac{G_m \cdot c_e}{E}\left(1-e^{-N_{SC}\cdot\xi}\right)\cdot(t_{av}-t_{env}), \quad (31)$$

where q is the daily demand for thermal energy, kJ.

Dependences (30) and (31) are similar to known ones in solar engineering [9, 10]:

$$q_u = F_R \cdot E - F_R \cdot U_L \cdot (t_{\text{av}} - t_{\text{env}}), \quad (32)$$

$$\eta = F_R \cdot \eta_0 - F_R \cdot U_L \cdot \frac{t_{\text{av}} - t_{\text{env}}}{E}. \quad (33)$$

F_R value is the total coefficient of heat transfer from SC to a heat-transfer agent:

$$F_R = \frac{G_m \cdot c_e}{U_L}\left(1-e^{-\frac{F' \cdot U_L}{G_m \cdot c_e}}\right), \quad (34)$$

where $U_L \equiv K_{\text{SC}}$ is the total coefficient of heat loss of SC to the environment; $F \equiv \xi$ is the coefficient of efficiency of heat transfer of an absorbing panel, which is equal to the ratio of useful heat production to the maximum production according to the definition, which coincides with the condition (21). Therefore, the second summands of formulae (28) and (32) are identical to (32) and (33), respectively:

$$G_m \cdot c_e\left(1-e^{-N_{SC}\cdot\xi}\right)\cdot(t_{av}-t_{env}) = = F_R \cdot U_L \cdot (t_{av}-t_{env}). \quad (35)$$

According to the equivalence of formulae (26) to (28), we can verify the theoretically calculated values of corresponding coefficients ($F_R \cdot \eta_0$) and ($F_R \cdot U_L$) experimentally using known standardized methods [7, 11].

In air collectors of the simplest design, heat transfer usually occurs from a heated panel directly to a heat-transfer agent without additional ribs to increase a thermal contact. An appropriate design is called a slit collector [8], for which a ratio of flows is equal to the ratio of corresponding heat transfer coefficients:

$$\xi = F' = \frac{\alpha_{ha}}{\alpha_{ha} + K_{SC}}, \quad (36)$$

where α_{ha} is the coefficient of heat transfer of SC, W/m^2·K.

We estimate values of ξ and η theoretically according to known structural parameters and operational characteristics of a collector.

4. 2. Substantiation of design of the air solar collector

The simplest one is a collector in the form of a box with insulated walls, double glazing, and a selective surface on its bottom. Double glazing minimizes a flow of convective heat loss through an absorbing surface due to a large heat transfer coefficient across a turbulent air flow. The regime of laminar flow reduces a convective component of heat loss, which makes it possible to apply a single-layer transparent coating. However, at the same time, a coefficient of heat transfer through an absorbing panel decreases, which requires extension of a contact path for a full assimilation of absorbed solar energy. Otherwise, temperature of an absorbing surface rises and radiation component of heat loss increases. Thus, according to a study [6], a performance index of a collector cannot exceed 24 %. Collectors with a flow of a heat-transfer directed into a gap between the rear side of a perceiving surface and the bottom of a collector have a higher efficiency, see Fig. 1.

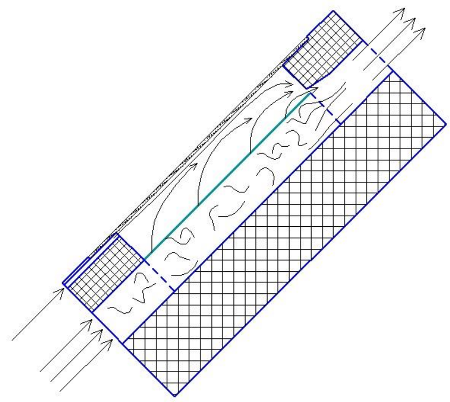

Fig. 1. Schematic structure of air SC

To increase efficiency of heat transfer from a heated panel, an air flow must be turbulized by selection of a slit cross-section, flow speed or additional structural elements. Instead, it is necessary to provide removal of superheated air from its upper part in a slit under a glass coating. To do this, it is enough to make several inlets of small diameter directly under roof glass, and upper outlets – near an absorbing panel. If holes are connected to a lower slit, then due to suction, a weak draught appears for effective ventilation of a space under glass. In this case, a weak laminar flow of fresh air tangent to glass will partially cut off emerging starts of vertical streams. This eliminates the need for double glazing, and an increase in heat output of a collector compensates a slight decrease in temperature of an outflow. Such constructive performance aligns at least a temperature field of transparent coating, a level of convective and radiation components of heat loss decreases. Due to the design, we can adjust characteristics of an air collector within certain limits by changing lower and upper gaps between a glass coating and a body. A length of a collector should be not more than 1.5 m, otherwise local chaotic convective microtubes, in the presence of draught, merge into a continuous steady flow along a transparent coating [15] with an inevitable increase in convective heat loss.

The transfer of heat from a perceiving surface to the useful stream of air occurs through an intermediate layer of a metal substrate with a high coefficient of thermal conductivity. We attach additional supports in the form of bent leaf-sheet copper channels of 2–3 cm of height to increase efficiency of heat transfer to the rear side of a perceiving panel. Supports increase stiffness of a thin sheet, which prevents it from fluctuation at interaction with a turbulized flow of air.

In order to reduce boundary effects, a pressure fan pre-turbulizes an input air flow. And at low speeds, the turbulent mode provides transverse inserts between longitudinal channel racks and a profiled surface of the collector bottom.

If the ratio of a length of an air line to its diameter is greater than 10, then contribution of boundary effects of the developed turbulent flow is insignificant and we can ignore heat transfer effects associated with it. At low flow speeds, the mode of advanced turbulence exists along a collector under condition $L/2l>10$, where L is the length of channel L, and $2l$ is its double height [9]. For a collector of 1 m in length, a height of a channel with a turbulized flow should not exceed 5 cm. The combination of the above constructive solutions contributes to an increase in the coefficient of heat transfer through a metal substrate at reduced flow speeds, output flow temperature and energy efficiency of a collector in general.

As we know [16], a radiation heat loss flow is approximately twice larger than the free convection flow at temperatures close to room temperature, so it is necessary to use glass with a heat-reflective coating of a solid type (K-glass) to reduce it. The advantage of this coating is high mechanical resistance, which makes it possible to use it on open surfaces, and the disadvantage is a slightly higher (than for "soft" coverage) radiation coefficient ε=0.1...0.15. It is metal oxides by composition, it is often tin doped with fluorine. However, it is expedient to apply such coatings only on a special glass with low iron content, which passes inside the much larger solar energy flow compared to a usual window glass, as we can see from the comparison of their spectral coefficients of transmission in Fig. 2.

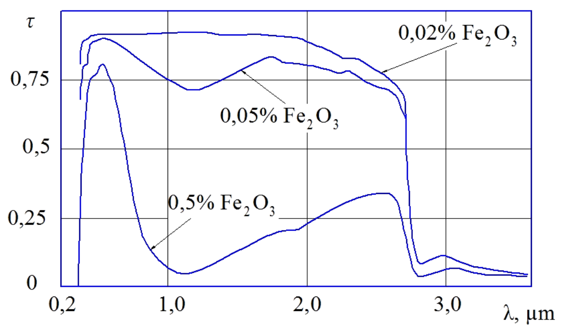

Fig. 2. Spectra of passing through glass with different iron content [3]

Both a solar flow and a return flow emitted from a heated surface of a perceiving panel with its subsequent injection into the environment heat glass with high iron content. Therefore, non-selectively transparent glass with a heat-reflective film, which returns a large part of a radiation flow, will provide a significant increase in efficiency of a solar collector. Based on the above estimates, we propose the design of a collector. Fig. 3 shows its scheme.

The walls of a collector are insulated with foam-plates with a thickness of 6 cm with a thermal conductivity λ=0.040 W/m·K. Heat loss from an absorbing surface to the environment passes through a glass covering and thermal insulation. The total area of thermal insulation includes the amount of areas of glass covering and the bottom F_{gl}=1.5 m^2, side ones Fs=0.18 m^2 and ends of both air chambers F_{in}= =0.06 m^2 of a thermal contact with heated air.

A heat loss flow through the insulation passes through a large area behind a glass covering, which we can express as a sum

$$F_{ins} = F_{gl} + F_s + F_{in} \simeq F_{gl}\left(1 + \frac{F_s + F_{in}}{F_{gl}}\right). \quad (37)$$

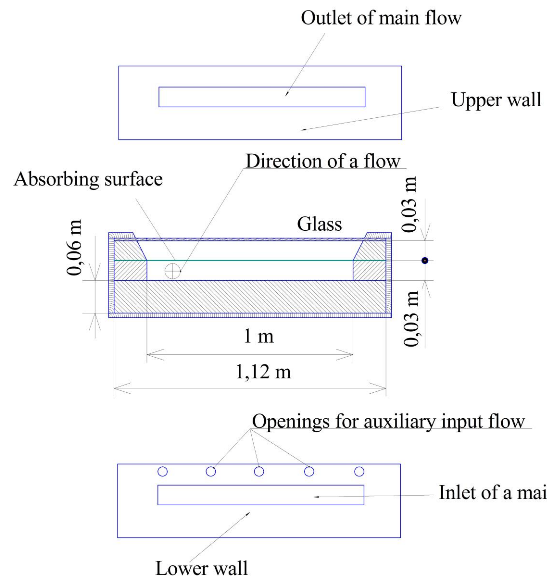

Fig. 3. Schematic design of an air collector

Thus, the proposed engineering solutions make it possible to obtain the maximum possible spectrum of direct sunlight that irradiates a surface of an absorbing plate and to reduce a diffuse component of radiation that is a part of a radiation flow and provides an increase in efficiency of a solar collector.

4. 3. Substantiation of the inclination angle of the receiving plane of solar collector to the horizon

We determine the optimum angle of inclination of an air collector to the horizon β_{opt} by the maximum solar energy that enters its surface during daylight hours. Instead of cumbersome calculations for each month of the drying season (for our case from July to October), it is expedient to use results of calculations provided on NASA's website [4]. According to calculations, the optimum angles of inclination and daily average solar energy inflows are:

– July – β_{opt}=13°; H_β=5.02 kW·h/m^2;
– August – β_{opt}=25°; H_β=4.77 kW·h/m^2;
– September – β_{opt}=39°; H_β=3.35 kW·h/m^2;
– October – β_{opt}=54°; H_β=2.51 kW·h/m^2.

Fig. 4 shows the corresponding graphical dependences.

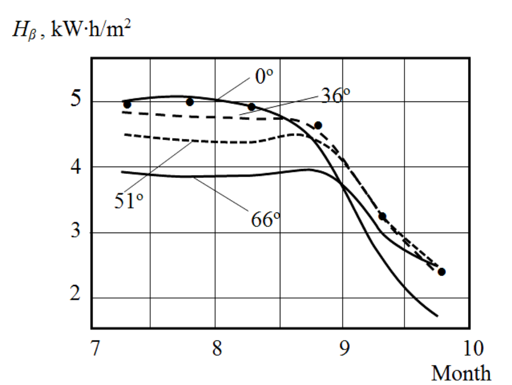

Fig. 4. Average daily arrival of solar energy on an inclined surface of the southern orientation for a month. The dots indicate an arrival of solar energy at optimal angles

The maximum sensitivity of a growth of heat output due to a change in the angle of inclination β_{opt} to the horizon is 20 % for stationary SC in the two summer months, and in the two autumn months – heat output of a collector depends on the angle of inclination insignificantly. Therefore, we choose the optimal angle of inclination of SC to the horizon β_{opt} close to the average annual optimum – 40.4°.

In the period close to the autumn equinox (September 21), the sun's inclination $\delta \to 0$, $\sin\delta \to 0$, $\cos\delta \to 1$; ω_0=15 deg/h takes the following form:

$$\cos\theta = (\sin\varphi \cdot \cos\beta - \cos\varphi \cdot \sin\beta) \cdot \sin\delta +$$
$$+(\cos\varphi \cdot \cos\beta + \sin\varphi \cdot \sin\beta \cdot \cos\gamma) \cdot \cos 15\tau \cdot \cos\delta,$$

where θ is the illumination angle of an arbitrarily oriented plane; φ is the latitude of the area; δ is the inclination (an angular position of the Sun at a sunny noon relative to the plane of the equator); γ is the azimuthal angle of a plane of deviation of the normal to a plane of the local meridian; ω_0 is the time angle (note that every hour ω value varies by 15° with the sign "+" (from morning till 12 o'clock) and "–" (from 12 o'clock to pm). For example, ω=+150 at 11 o'clock, and ω=–150 at 1 pm); τ is the duration of solar energy inflow, s.

Since cosγ=1 for the southern-oriented SC, we use simple trigonometric relations and simplify the last expression to the form:

$$\cos\theta = \cos(\varphi - \beta)\cos 15 \cdot \tau. \quad (38)$$

According to the well-known differentiation rule, the cosine increase of an angle of illumination is as follows:

$$\Delta\cos\theta \approx \cos 15\tau \cdot \sin(\varphi - \beta)\Delta\beta. \quad (39)$$

Since the intensity of the cosine of the angle of illumination determines energy illumination of SC

$$E = I \cdot \cos\theta, \text{ then } \Delta E = I \cdot \Delta(\cos\theta), \quad (40)$$

where I is the intensity of solar energy inflow on an inclined surface of an air collector, W/m^2.

We can estimate relative change in the energy intensity of SC by the following ratio:

$$\frac{\Delta E}{E} = \frac{I\Delta(\cos\theta)}{I\cos\theta} = \frac{\Delta(\cos\theta)}{\cos\theta} =$$
$$= \frac{\cos 15\tau \cdot \sin(\varphi - \beta) \cdot \Delta\beta}{\cos(\varphi - \beta) \cdot \cos 15 \cdot \tau} = \mathrm{tg}(\varphi - \beta)\Delta\beta. \quad (41)$$

For φ=50.6° (a latitude of Korets city) and the optimal angle of inclination from July to October β≈40.4° tg(φ–β)≈0.21. Taking into consideration a transfer coefficient for the transition from degree to radial, we have:

$$\frac{\Delta E}{E} \approx 0{,}21 \cdot \frac{\pi}{180}\Delta\beta = 0{,}0037 \cdot \Delta\beta. \quad (42)$$

For example, a change in the angle of inclination of SC by 10° will change its energy illumination, and hence hourly thermal efficiency only by

$$\frac{\Delta E}{E} \cdot 100\,\% \approx 0{,}0037 \cdot 10 \cdot 100\,\% = 3{,}7\,\%.$$

The change does not exceed an error of calculation of energy illumination, therefore a change of the angle of inclination by 10° is invisible.

Consequently, the optimum angle of inclination of an inclined recepient plane of an air collector to the horizon is equal to β_{opt}=40.4°.

4. 4. Substantiation of the design and technological parameters of a solar collector

The basis for calculation of SC is heat output of an air solar collector Q determined by the formula

$$q = F \cdot F_R \cdot \left[\begin{gathered} \left(k(\tau) \cdot R_\beta \cdot E^{\max} \cos\pi \frac{\tau}{\tau_c} \right) \times \\ \times(\alpha \cdot \tau) - U_L \cdot (t_{\text{out}} - t_{\text{env}}) \end{gathered} \right], \text{ kJ/h or kW}, \quad (43)$$

where S_{ac} is the area of an air collector, m^2; F_R is the coefficient of heat output from an air collector; k is the coefficient of amplification of solar energy by a mirror concentrator; R_β is the coefficient of average monthly income of solar radiation; E^{max} is the maximum energy illumination of a horizontal surface of an air collector, W/m^2; U_L is the coefficient of heat loss of an air collector, W/m^2·°C); τ_c is the duration of receipt of solar energy, s; α, τ are coefficients of absorption and transmission of solar radiation; t_{env}, t_{ha1} are air temperatures at the inlet and the outlet of a collector, °C.

The area of *SC F* will take the form

$$F_{\text{SC}} \geq \frac{q/2}{3600 \cdot \eta \cdot H_\beta^{\text{i}}}, \text{ m}^2, \quad (44)$$

where q is the daily demand for thermal energy, kJ; η is the performance index of solar collector; H_β^{i} is the solar energy inflow, kW·h/m^2.

The obtained analytical equations (43) and (44) make it possible to calculate heat output and the area of the perceiving surface of SC.

4. 5. Procedure for the estimation of heat loss by a solar collector

We determine the total heat loss of a collector to the atmosphere by the sum of radiation and convective heat loss from an external surface – a glass covering and a body. We can simulate the complete resistance of heat transfer conveniently according to the scheme of electrothermal analogy. Fig. 5 shows it.

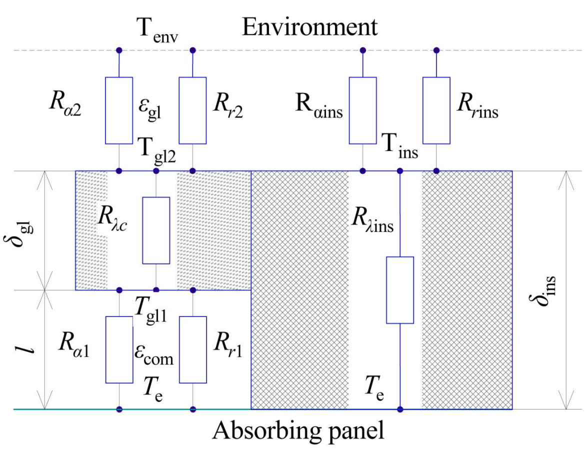

Fig. 5. Equivalent circuit of heat loss flows from an absorbing panel to the environment through a glass covering and thermal insulation

The chain of heat loss resistances through a glass covering of R_{gl} consists of two parallel supports from the absorbing surface to the glass, thermal conductivity of the glass successively resisting to them and two parallel resistances from the glass to the environment. Particularly: $R_{\alpha 1}$, $R_{\alpha 2}$ i $R_{\alpha ins}$ are resistances to flows of heat transfer; $R_{\lambda c}$, and $R_{\lambda ins}$ are resistances to heat conductivity flows; R_{r1}, R_{r2} and R_{rins} are resistances to radiation (radiant) heat transfer flows. The total heat loss resistance is equal to the effective sum of the above resistances R_{SC}, and the return value is equal to the heat transfer coefficient ($K_{SC}=1/R_{SC}$).

We estimate flows of convective heat loss in an inclined slit with a use of known Rayleigh and Nusselt criteria using standard techniques given, for example, in papers [9, 16–18]:

$$\alpha=\frac{\lambda\cdot Nu}{l},\quad Ra=\frac{g\cdot\beta_V\cdot\Delta\mathrm{T}\cdot l^3}{\nu\cdot a}, \tag{45}$$

where α is the coefficient of heat transfer, W/m^2·K; λ is the mean by a temperature range coefficient of air conductivity, W/m·K; ν is temperature range, kinematic air viscosity, m^2/s; a is the average in a temperature range air temperature conductivity, m^2/s; g=9.81 m/s^2 is the acceleration of free fall; $\beta_V=1/T_{av}$ is the coefficient of volumetric expansion of air; T is absolute temperature, K; ΔT is the temperature range or a temperature difference between parallel surfaces, K; l is the distance between parallel surfaces, m (a height of a slit and not a hydraulic diameter) [9].

In the range of inclinations of parallel plates to the plane of the horizon β from 0 to 75°, the numbers of Nusselt and Rayleigh relate to the following ratio:

$$Nu=1+1{,}44\cdot\left[1-\frac{1708\cdot(\sin\beta)^{1{,}6}}{Ra\cdot\cos\beta}\right]\times$$
$$\times\left[1-\frac{1708}{Ra\cdot\cos\beta}\right]+\left[\left(\frac{Ra\cdot\cos\beta}{5830}\right)^{1/3}-1\right]^{+}. \tag{46}$$

The sign (+) instead of the exponent of the last summand means that only a positive value of the expression in square brackets is accepted, whereas in the negative case we consider it equal to zero. And for estimation calculations, we usually use graphical dependences of Nusselt and Rayleigh numbers [9] for several typical angles of a collector and a temperature difference ΔT=10°.

The speed of air flow usually changes in wide limits with the corresponding redistribution of contributions of mechanisms of heat conduction and heat transfer in the total flow of heat transfer. In the range of values $0<Nu<1$, heat transfer prevails due to heat conduction, and for $Nu\geq 1$, it is mixed with the predominant contribution of a heat transfer mechanism. By definition, the Nusselt number is equal to the ratio of heat transfer coefficients between surfaces with the same temperature difference, but with different mechanisms of heat transfer – convective and thermal conductivity:

$$q_c=\alpha_c\cdot(t_1-t_2)\ \text{and}\ q_r=\frac{\lambda}{l}\cdot(t_1-t_2),$$

where α_c is the coefficient of heat transfer of an absorbing surface, W/m^2·K; t_1, t_2 are temperatures of inlet and outlet flows of a heat-transfer agent, °C.

We divide the first equality into the second and obtain the known relation:

$$\mathrm{Nu}=\frac{q_c}{q_r}=\frac{\alpha_c}{\lambda/l}, \tag{47}$$

where l is the distance between inclined surfaces and a non-hydraulic diameter is usually taken for a flow along plates.

If values of the Nusselt number close to one, contributions of thermal conductivity of convective heat transfer are comparable, so it is expedient to calculate their total contribution by such an obvious ratio:

$$q=q_c+q_\lambda=\left(\alpha_c+\frac{\lambda}{l}\right)\cdot(t_1-t_2)=k\cdot(t_1-t_2) \tag{48}$$

or

$$k=\alpha_c+\frac{\lambda}{l}=\alpha_c+\frac{\alpha_c}{\mathrm{Nu}}. \tag{49}$$

In a spatially isolated interplanar space of an inclined collector, heated air accumulates in its upper part with the additional heating of the absorbing panel, the heat carrier flow and the corresponding redistribution of heat loss. With a loose connection of glass coating in the lower and upper parts, there is a slight draught and convective flows go into the atmosphere or into the main flow of a heat-transfer agent. The second option is more expedient, because it can additionally increase heat output of an air collector with simultaneous reduction of heat loss. Due to it, we can adjust characteristics of an air collector within certain limits by changing lower and upper gaps between a glass cover and a body.

The radiation component of the heat loss flow from a heated surface with a temperature T_e to a glass coating with a temperature T_c at low overheats is convenient to evaluate according to the Stefan-Boltzmann formula modified in [8, 16]:

$$\alpha_r=0{,}227\cdot\varepsilon_{\mathrm{com}}\cdot\left(\frac{T_e+T}{200}\right)^3. \tag{50}$$

We determine the combined coefficient of radiation heat exchange $\varepsilon_{\mathrm{com}}$ for parallel surfaces of the same area from formula

$$\varepsilon_{\mathrm{com}}=\frac{1}{\frac{1}{\varepsilon_1}+\frac{1}{\varepsilon_2}-1}, \tag{51}$$

where $\varepsilon_{1\ldots n}$ is the double radiation coefficient due to glazing.

If one of surfaces is selectively radiating with a coefficient ε=0.1, then the combined coefficient is also appropriate to assume equal to $\varepsilon_{com}\approx 0.1$. We can regard radiation heat loss of a glass surface of a collector to the environment as an exchange of radiation between a glass covering and the sky with a conventional temperature, a few degrees less than the environment. But for the non-commensurability of both areas, the combined degree of blackness ε_{com} is equal to the larger one in this case glass ε_{gl}=0.93.

A specific flow of heat loss q_{gl} passes through the first chain with a total thermal resistance R_{gl}:

$$q_{gl}=\frac{t_e-t_{env}}{R_{gl}}=K_{gl}\cdot(t_e-t_{env}), \tag{52}$$

where

$$R_{gl}=\frac{1}{\alpha_{c1}+\alpha_{r1}}+\frac{\delta_{gl}}{\lambda_{gl}}+\frac{1}{\alpha_{c2}+\alpha_{r2}} \text{ and } \frac{\delta_{gl}}{\lambda_{gl}}=R_{\lambda c}, \qquad (53)$$

where δ_{gl} is the thickness of SC thermal insulation, m; $\alpha_{c...n}$ is the heat transfer coefficient of SC heat insulation, W/m^2·K; $\alpha_{r...n}$ is the coefficient of heat transfer of the enclosure of SC, W/m^2·K; λ_{gl} is the coefficient of thermal conductivity of thermal insulation of SC, W/m·K.

$$K_{gl}=\frac{1}{R_{gl}}=\frac{1}{\frac{1}{\alpha_{c1}+\alpha_{r1}}+R_{\lambda c}+\frac{1}{\alpha_{c2}+\alpha_{r2}}}. \qquad (54)$$

The second, much smaller, flow of heat loss overcomes a temperature difference (t_e-t_{env}) through the line of thermal resistances of thermal insulation

$$q_{ins}=K_{ins}\cdot\left(t_e-t_{env}\right)\approx\frac{t_e-t_{env}}{R_{ins}}, \qquad (55)$$

where K_{ins} is the effective coefficient of heat transfer of SC heat insulation, W/m^2·°C.

High resistance of the thermal conductivity determines this weak flow mainly, due to which temperatures of a surface and the environment differ slightly. Formally, this reduces to the sum of both thermal resistances – convection one and radiation one – equal to zero:

$$R_{ins}=\frac{\delta_{ins}}{\lambda_{ins}}+\frac{1}{\alpha_{r_{ins}}}+\frac{1}{\alpha_{in_{ins}}}\rightarrow\frac{\delta_{ins}}{\lambda}, \qquad (56)$$

where δ_{ins} is the thickness of enclosing elements of SC, m; α_{cins} is the coefficient of heat transfer of enclosing elements of SC, W/m^2·K; α_{rins} is the coefficient of heat transfer of thermal insulation of SC, W/m^2·K; λ_{ins} is the coefficient of thermal conductivity of thermal insulation of enclosing elements of SC, W/m·K.

Thus

$$K_{ins}\approx\frac{1}{R_{ins}}=\frac{\lambda_{ins}}{\delta_{ins}}. \qquad (57)$$

Polyfoam body is covered with wood of 2 cm thickness and the coefficient of thermal conductivity λ≈0.1 W/m·K for durability. We estimate its total thermal resistance by the following value:

$$R_{\lambda ins}=R_p+R_w=\frac{\delta_p}{\lambda_p}+\frac{\delta_w}{\lambda_w}=$$
$$=\frac{0{,}06}{0{,}04}+\frac{0{,}02}{0{,}1}=1{,}5+0{,}2=1{,}7.$$

Therefore, the total coefficient of heat loss of a collector is equal to the sum of the coefficient for the first flow and the second one multiplied by the area of its thermal contact:

$$K_{SC}=K_{gl}+\frac{1}{R_{\lambda\iota\nu\sigma}}\left(1+\frac{F_s+F_{in}}{F_{gl}}\right). \qquad (58)$$

After substitution of known values, the total coefficient of heat loss collector takes the following form:

$$K_{SC}=K_{gl}+\frac{1}{1{,}7}\cdot\left(1+\frac{0{,}18+0{,}06}{1{,}5}\right)=K_{gl}+0{,}68.$$

The partial coefficient of heat loss through glass depends indirectly on temperature by coefficients of convective and radiation heat transfer. Therefore, we calculate it for conditions of equilibrium temperature when a collector is uniformly warmed up, and heat transfer processes are stationary. We should perform the calculation by the iteration method, we start with the zero value of K_{SC0} with a close to statistically average K_{gl}=3.5 W/m^2·K for selective coverage and 7.0 for non-selective one:

$$K_{SC0}=3{,}5+0{,}68=4{,}18 \text{ W/m}^2\text{·K}. \qquad (59)$$

It is expedient to estimate thermal and energy characteristics of a solar collector for medium conditions of insolation and for clear sky conditions. In the second case, it is advisable to take an insolation level at 1,000 W/m^2, which corresponds to the AM 1.5 flow standard [11]. The equilibrium temperature is convenient to accept as typical temperature for operation conditions of a collector for its average energy illumination $\bar{E}$ by formula:

$$t_{p0}=\frac{\eta_0\cdot\bar{E}}{K_{SC0}}+t_{env}. \qquad (60)$$

In turn, we determine $\bar{E}$, value by the known meteorological database, for example, NASA [4], an average daily energy exposure H_β, which should be divided into the duration Δτ of sun irradiation of an inclined surface in hours:

$$\Delta\tau=\frac{2\arccos\left[-\text{tg}(\varphi-\beta)\cdot\text{tg}\delta\right]}{\omega}, \qquad (61)$$

where φ is the geographical latitude of the area; β is the angle of inclination of a collector of the southern orientation; δ is the current value of the sun's inclination; ω=15 deg/h the angular speed of rotation of the Earth around its own axis.

Conditions for moderate level of insolation

The average daily energy exposure in Korets (φ≈50°) in June is ≈4.7 kW/h, and duration of solar illumination of a collector during a period of equinox at δ=23.45° is equal to:

$$\Delta\tau=\frac{2\arccos\left[-\text{tg}(50-45)\cdot\text{tg}23{,}45\right]}{15}=12{,}3 \text{ hours.}$$

According to this

$$\bar{E}=\frac{4637}{12{,}3}=377{,}0 \text{ W/m}^2.$$

Thus, the analytical equations obtained give possibility to calculate Rayleigh (45) and Nusselt (46) criteria and to estimate heat loss of a solar collector (58).

5. Results of the iterative calculation of structural and technological parameters of an air solar collector

Based on the developed method of calculation of structural and technological parameters of SC, we determine numerical values of the main parameters of a collector. Tables 1–4 show the obtained results of iteration calculations. The first and subsequent steps begin with determination of equilibrium temperature for a new value of K_{SC}. We stop calculations if the next value of a heat loss coefficient does not differ from the previous one by more than 5 %.

We accept the following values as input parameters for the calculation:

– solar energy flow q_E=324.2 W;

– wind speed υ=3.0 m/s;

– we take a degree of blackness and a coefficient of absorption of a panel as equal and equal to ε_{gl}=0.93, and a radiation coefficient of a selective panel as ε_c=0.1;

– thermal resistance of glass with thickness δ_{gl}=0.004 m and a coefficient of thermal conductivity λ_{gl}=0.76 W/m·K equal to $R_{\lambda c}$=0.00526 m^2·K/W;

– optical performance index of a collector η_0=0.86;

– an area of a perceiving surface of a collector F_{SC}= =1.5 m^2;

– coefficient of heat loss of a collector in the zero approximation of K_{SC}=4.18 W/m^2·K;

– we assume that the angle of inclination of a collector β is 40.4°;

– equilibrium temperature in the zero approximation t_{p0}=103 °C=376 K.

The heat-transfer agent flows along two parallel planes in a slit, the defining size of which is called a hydraulic diameter D_h.

In the case of a rectangular section of 0.03×1.0 m, the hydraulic diameter is almost equal to the double distance between the planes D_h=0.06 m [9, 18].

Conditions for standard insolation E=1,000 W/m^2. As a zero approximation, we take the value K_{SC} equal to the obtained for moderate conditions 7.3 W/m^2·K in the case of a non-selective absorbing panel, and 3.76 W/m^2·K for selective ones.

Table 1

Results of iterative calculation of K_{SC} for moderate insolation. The results for SC with a selective surface and a radiation coefficient ε=0.1 are in bold type

№	t_{gl}, T_{gl}	T_{gle}, T_{gl}	ΔT_1	Panel-glass					Glass-Environment			
				ν, *a*	*Ra* *Nu*	λ	α_c, *k*	α_r α_r	υ	α_c	α_r	K_{gl}
0	K_{SC0}=4.18 t_e=103 °C=376 K; ε=0.93; ε=0.1											
1	64, 337	83, 357	39	22.2 31.4	43,100 3.05	0.03074	3.58 4.16	8.85 1.02	3.0	12.76	6.1	7.4 4.0
	Non-selective K_{SC1}=7.4; t_e=69 °C=342 K; C*el.* K_{SC1}=4.68; t_e=94 °C=367 K											
2	47, 320 59.5 332.5	58, 329.5 77 350	22 34.5	19.1 25.8 21.2 29.8	29,552 2.79 36,2019 4.84	0.0289 0.0303	1.34 1.82 2.44 2.94	7.08 0.97	3.0	12.76	6.0 5.97	6.52 3.18
	K_{SC2}=7.2; t_e=70.0 °C=343 K; K_{SC2}=3.86; t_e=84.0 °C=357 K											
3	47.5 320.5 54.5 327.5	58.5 331.5 69 342	22.5 29.5	19.2 27.1 20.3 28.6	34,300 2.89 314,387 4.79	0.0289 0.0297	2.78 3.74 2.37 2.86	7.13 0.91	3.0	12.76	6.0 5.83	6.64 3.08
	K_{SC1}=7.3; t_e=69.4 °C=342 K; K_{SC1}=3.76 t_e=86 °C=359 K											

Table 2

Results of evaluation of collector efficiency at υ=3 m/s

D_h, cm	G_m, Kg/s	*Re*	Nu	α_{ha}, W/m^{2}K	ξ	N_{SC}	t_{out}, °C	Δ*t*	q_k, W	q_E, W	η, %
Non-selective absorbing surface ε=0.93; t_e=69.4 °C											
8	0.24	25,316	60.0	19.7	0.730	0.0454	26.4	1.4	337.7	486.3	69.4
6	0.18	188,970	47.6	20.9	0.741	0.0605	26.9	1.9	343.7	486.3	70.7
4	0.12	12,658	34.4	22.6	0.756	0.0908	27.9	4.0	482.4	486.3	71.9
Selective absorbing surface ε=0.1; t_e=86.0 °C											
8	0.24	25,316	60.0	19.7	0.840	0.0234	26.2	1.2	289.4	486.3	59.5
6	0.18	188,970	47.6	20.9	0.847	0.0312	26.6	1.6	289.4	486.3	59.5
4	0.12	12,658	34.4	22.6	0.857	0.0468	27.4	2.4	289.4	486.3	59.5

Table 3

Results of iterative calculation of K_{SC} for conditions of standard insolation

№	t_{gl}, T_{gl}	t_{gle}, T_{gle}	ΔT_1	Panel-glass					Glass-environment			
				ν, a	Ra Nu	λ	α_c, k	α_r	υ	α_c	α_r	K_{gl}
0	K_{SC0}=7.3 t_e=143 °C=416 K; K_{SC0}=3,76 t_e=254 °C=527 K											
1	84, 357 139.5 412.5	113.5 386.5 197 470	59 115	24.7 35.0 34.6 49.2	45,096 3.09 38,113 2.97	0.0323 0.0384	1.66 2.20 1.90 2.54	11.27 2.35	3.0	12.76	7.16 9.27	7.78 4.44
	K_{SC1}=7.78; t_e=110.5 °C=383.5 K; K_{SC1}=4.44; t_e=194°C=467 K											
2	68 341 110 383	89 362 152 425	42.5 84	22.5 31.8 29.2 41.44	43423 3.06 43211 3.1	0.0312 0.0349	3.18 4.22 3.61 4.77	10.04 1.74	3.0	12.76	6.42 8.15	7.84 4.84
	K_{SC2}=7.84; t_e=110 °C=383 K; K_{SC2}=4,84; t_e=178 °C=451 K											
3	67.5 340.5 101.5 374.5	89 362 140 413	42.5 76.5	–‖– –‖– 26.45 39.4	–‖– –‖– 47059 3.11	–‖– –‖– 0.0341	–‖– –‖– 3.54 4.68	–‖– –‖– 1.60	3.0	12.76	–‖– –‖– 7.85	7.84 4.70
	K_{SC3}=7.84 t_e=110 °C=383 K; K_{SC3}=4.70 t_e=183 °C=383 K											

Table 4

Results of evaluation of SC efficiency at υ=3 m/s

D_h, cm	G_m, Kg/s	Re	Nu	α_{ha}, W/m^{2}K	ξ	N_{SC}	t_{out}, °C	Δt	q_k, W	q_E, W	η, %
Non-selective absorbing surface ε=0.93; t_e=110 °C											
8	0.24	25,316	60.0	19.7	0.715	0.0488	27.9	2.9	699.5	1290	54.2
6	0.18	188,970	47.6	20.9	0.727	0.0650	28.9	3.9	705.5	1290	54.6
4	0.12	12,658	34.4	22.6	0.742	0.0975	30.9	5.9	711.5	1290	55.2
Selective absorbing surface ε=0.1; t_e=86.0 °C											
8	0.24	25,316	60.0	19.7	0.807	0.0292	28.7	3.7	892.4	1290	69.2
6	0.18	188,970	47.6	20.9	0.816	0.039	29.95	4.95	895.5	1290	69.4
4	0.12	12,658	34.4	22.6	0.828	0.0585	32.5	7.5	904.5	1290	70.1

A contact area of an absorbing panel with thermal insulation, which occurs through an intermediate layer of a heat-transfer agent in a heating channel, is overestimated in calculations of heat loss. At high costs of a heat-transfer agent, its temperature is only a few degrees higher than input one, so only a radiation component determines the corresponding flow of heat loss in the direction of the bottom. We can reduce this component effectively by an additional reflective substrate, for example, a substrate made of a polished metal. Taking into consideration the above circumstances, the overall heat loss ratio reduces significantly reduced due to an insulation component, which will ultimately lead to an increase in SC efficiency.

At small levels of illumination, a performance index of the nonselective SC is higher than the selective one, and at large levels, on the contrary, due to the redistribution of ratios of contributions of convective and radiation heat loss flows.

To increase temperature of the outflow, it is necessary to reduce the flow of the heat-transfer agent with the simultaneous intensification of the process of its turbulization to maintain a coefficient of heat transfer and heat production of SC.

Increasing a section of a channel for the passing of a heat-transfer agent while maintaining its speed does not affect efficiency of SC significantly if the turbulent flow regime does not change.

An increase in temperature of the outflow and its efficiency is also possible by increasing a collector length and the corresponding area of air contact with an absorbing panel. With such a constructive solution, the material content of SC increases, as well as energy costs to overcome increased resistance of the flow of a heat-transfer agent.

Options for the optimal correlation between performance of SC and temperature of the outflow require a separate study, taking into consideration acceptable deviations from the optimal technological parameters of the drying process.

6. Discussion of results of substantiation of structural and technological parameters of a solar dryer

Based on the analysis we established that the use of existing flat or tubular collectors is unprofitable in solar dryers for drying of moist materials of veobtainable origin. Mostly because of the high cost of collector elements and expensive and complex operation and maintenance conditions. Therefore, it is necessary to use air SC for drying of moist materials of veobtainative origin in a solar energy dryer, in order to provide an economically advantageous and energy-saving drying regime.

We developed a new design of an air SC for a solar dryer, which consists of a double glazing and a selective surface made of a thin metal substrate at its bottom with input and output holes. There is a pressure fan mounted in a collector for the turbulence of the inlet air. The walls of a collector are insulated with foam-plates with a thickness of 6 cm with a thermal conductivity λ=0.040 W/m·K. The total area of thermal insulation by the sum of areas of glass covering and the bottom F_{gl}=3 m^2, side F_s=0.18 m^2 and the end (input and output) walls of both air chambers F_{in}=0.06 m^2. We established that for a double-glazing substrate it is necessary to use a glass with a heat-reflective coating of solid K-glass type with a radiation coefficient (ε=0.1...0.15).

Based on the performed study, we propose engineering methods for calculation of basic elements of an air SC. The methods enable:

– analyze and simulate distribution of temperatures along SC;

– substantiate power, heat output, area, and an angle of inclination of a horizontal absorbing surface, SC efficiency;

– estimate a flow of thermal energy and heat loss of SC and carry out calculation-quantitative iterative studies.

During calculation-quantitative iterative studies we established that a contact area of an absorbing panel with thermal insulation over the intermediate layer of a heat-transfer agent in a heating channel is overestimated in heat loss calculations. Therefore, at high costs of a heat-transfer agent, its temperature will be only a few degrees higher relatively to the input, and heat loss at the bottom will be a radiation component only. It is possible to reduce heat loss by additional reflective subtract made of polished metal. Therefore, taking into consideration the above-mentioned factors, the overall coefficient of heat loss K_{SC} will be reduced due to the insulating component of polished metal. This will increase energy efficiency of an air solar collector by 3.25 times. In addition, an increase in a channel cross section for the passing of a heat-transfer agent at the speed υ=3 m/s does not affect performance index of a collector significantly if the turbulent flow regime does not change. We established that efficiency of a collector at low levels of illumination of the non-selective coating of an absorbing plate is higher than η=70.7 % for selective one, and at high energy illumination at E=1,000 W/m^2 small of order η=54.6 %. This makes it possible to explain how distribution of the ratio of contributions of convective and radiation heat loss flows to air SC occurs. Therefore, when temperature of the outflow air in SC increases, it is necessary to reduce a flow of a heat-transfer agent with the simultaneous intensification of the turbulence process to maintain the coefficient of heat transfer and heat output. Thus, an increase in temperature of outflow and the energy efficiency of SC is possible by increasing a length of a collector and the contact area of a heat-transfer agent and an absorbing panel. Due to such a constructive solution, the material content of SC increases by 1.25 times and energy costs to overcome the increased resistance of a heat-transfer agent flow – by 1.1 times. Therefore, the studies carried out in the work require further studies on an increase in power, heat output and a performance index of an air solar collector based on substantiation of optimal parameters and operation modes and energy analysis of estimation of heat energy and heat loss under conditions of temperate continental climate, namely, Korets, Rivne Oblast (Ukraine).

Consequently, the proposed design of an air SC is not inferior to existing solar-thermal installations by technical characteristics, Particularly flat or tubular air collectors.

Thus, a use of an air SC in a solar dryer for drying of moist materials of veobtainable origin is appropriate and effective under conditions of individual farms. This will lead to an increase in a volume of produced high-quality dried products with minimal energy consumption due to solar energy.

7. Conclusions

1. We proposed the design of an air solar collector with flow turbulators. We determined regularities of the influence of a change of heat-transfer agent flow speed, a temperature difference and intensity of solar radiation on power of a solar collector. Particularly, we determined the optimal current power of N_{SC}, which we can obtain by a collector under standard conditions of insolation level E=1,000 W/m^2 with non-selective absorbing surface at ε=0.93; t_p=69.4 °C; N_{SC}=45–98 W and with a selective absorbing surface at ε=0.1; t_p=86.0 °C, N_{SC}=29–58 W. Based on the worked out calculation-quantitative iterative data, we calculated a performance index of the solar collector, which was, for a non-selective absorbing surface at ε=0.93; t_e=69.4 °C; η=69–71 %. and for a selective absorbing surface at ε=0.1; t_e=86.0 °C, η=59 %.

2. We presented the methodology for evaluation of heat loss of an air solar collector based on the comparison of the heat transfer coefficient of a solar collector K_{SC} for non-selective and selective coatings. As a result of computational and quantitative iterative studies, we established K_{SC} for non-selective and selective coatings at ε=0.93; t_p=69.4 °C; K_{SC}=4.18–7.4, and for selective coatings at ε=0.1; t_p= =86.0 °C, K_{SC}=3.76–4.84.

3. We established that energy illumination E, which is from 377 to 1,223 W/m^2, significantly influences heat output of an air collector Q=117...480 W. We found that the use of a non-selective absorbing surface in an air solar collector with a low insolation level of E=377 W/m^2 makes it possible to increase a performance index at η=70.7 % for selective surface, and at high energy illumination at E=1,000 W/m^2 on the contrary small η=54.6 %. This makes it possible to explain how the redistribution of the ratios of the maximum current thermal power (N_{SC}=48.8...100 W) and efficiency of an air solar collector occurs.

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