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MODELLING OF HEAT TRANSFER IN AN AIR SOLAR COLLECTOR

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MODELLING OF HEAT TRANSFER IN AN AIR SOLAR COLLECTOR

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Abstract. *This study is the prediction, depending on time and different geographical and climatic situations, of different operating parameters: solar radiation, ambient and outlet coolant temperature, heat exchange coefficients and losses. All these determine, in the absence of measurements, the efficiency of the system modelled with existing models with respect to these parameters. These models are usually compared with experimental results. Before modelling, it is necessary to investigate the different heat transfer and exchange regimes that exist between the different elements of this system in order to eventually have a system of equations governing this exchange. The modelling principle is to record the energy balances of each element that makes up the collector, absorber, glass, insulator and coolant. For this purpose, several models are encountered, two modelling approaches are also encountered.*

Keywords: *ventilation system, dyer, temperature, thermal energy, absorber, equipment.*

Annotatsiya. *Ushbu tadqiqot vaqtga va turli xil geografik va iqlim sharoitlariga qarab, turli xil operatsion parametrlarning prognozidir: quyosh nurlanishi, atrof-muhit va chiqish harorati, issiqlik uzatish koyeffitsiyentlari va yo'qotishlar. Bularning barchasi, o'lchovlar bo'lmagan taqdirda, ushbu parametrlarni hisobga olgan holda mavjud modellar yordamida simulyatsiya qilingan tizimning samaradorligini aniqlaydi. Ushbu modellar odatda eksperimental natijalar bilan taqqoslanadi. Modellashtirishdan oldin, oxir-oqibatda ushbu almashinuvni boshqaradigan tenglamalar tizimini olish uchun tizimning turli elementlari o'rtasida mavjud bo'lgan issiqlik uzatish va issiqlik almashinuvining turli usullarini o'rganish kerak. Modellashtirish prinsipi kollektor, absorber, shisha, izolyator va sovutish suvini tashkil etuvchi har bir elementning energiya balanslarini qayd etishdan iborat. Ushbu maqsadlar uchun bir nechta modellar, shuningdek, ikkita modellashtirish yondashuvi mavjud.*

Kalit so'zlar: *shamollatish tizimi, bo'yoq, harorat, issiqlik energiyasi, absorber, uskunalar.*

Аннотация. *Данное исследование представляет собой прогнозирование, в зависимости от времени и различных географических и климатических ситуаций, различных рабочих параметров: солнечной радиации, температуры окружающей среды и выходящего теплоносителя, коэффициентов теплообмена и потерь. Все это определяет, в отсутствие измерений, эффективность системы, смоделированной с помощью существующих моделей с учетом этих параметров. Эти модели обычно сравниваются с экспериментальными результатами. Перед моделированием необходимо изучить различные режимы теплопередачи и теплообмена, существующие между различными элементами системы, чтобы в итоге получить систему уравнений, управляющих этим обменом. Принцип моделирования заключается в записи энергетических балансов каждого элемента, составляющего коллектор, абсорбер, стекло, изолятор и теплоноситель. Для этой цели встречается несколько моделей, а также два подхода к моделированию.*

Ключевые слова: *система вентиляции, краситель, температура, тепловая энергия, абсорбер, оборудование.*

Introduction

Current state of thermal power engineering The Republic of Uzbekistan is characterized by growth fuel prices and, as a consequence, high cost of thermal energy. Due to depletion of traditional hydrocarbons energy carriers, emissions into the atmosphere greenhouse gases at alarming levels and noticeable

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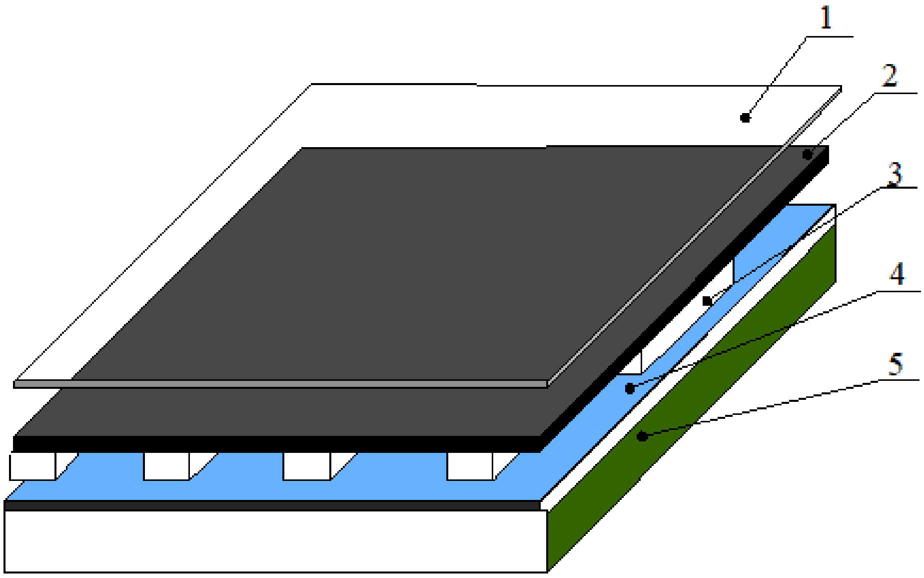
climate change, increasingly acute prospects become more relevant use of solar energy. Saving traditional fuel - energy resources are currently also is the main task of the world economy. Thus, on the initiative of the International Energy Agency already in these years almost all countries are planning large-scale implementation activities environmentally friendly technologies, including solar energy.

The system under study consists of two systems, the first system is a solar thermal air of flat geometric shape. The second system is a drying chamber that receives this energy in the form of thermal energy carried by the heat carrier (air).

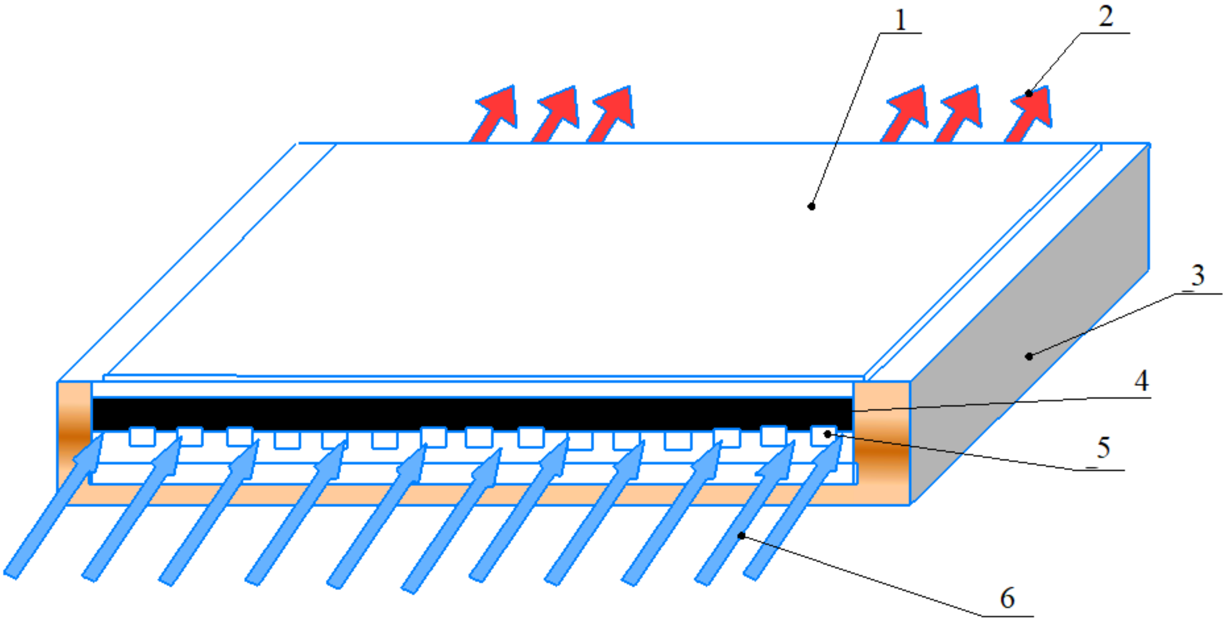
Materials and methods

Our study is based on the first system in Figures 1 and 2 which consists of a transparent double-layered glass cover, under which there is a fixed air vane, and a metal plate (painted in black matte colour) with a baffle to ensure maximum absorption of solar radiation. This plate plays the role of an absorber-converter, as it absorbs the sun's rays and converts them into thermal energy to be transferred to the heat transfer medium.

The second subsystem is a drying chamber consisting of four racks and equipped with a ventilation system. The air enters from below, passes through the four pallets and is extracted to the outside by a centrifugal fan [1-3].



1-Two-layer glass cover; 2-Absorber painted in black matte colour; 3-Baffle; 4-Metal sheet; 5-Isolation.
Fig.1. Schematic diagram of the main components



1-two-layer glass cover; 2-outlet coolant; 3-insulation; 4-absorber; 5-baffle; 6-air inlet.
Fig.2. Schematic diagram of a solar air collector

Conduction is the mode of heat transfer usually in solids, but sometimes also in liquids or air, except that in this case it is negligible compared to convection or radiation. The law governing this process of heat propagation is Fourier's law. In a planar collector, the exchange of conductivity is mainly between:

- top and bottom edges are glass;
- top and bottom edges of the insulation;
- upper and lower edges of the absorber;
- absorber and insulator on the sides of the collector [3-5].

The thermal balance of a flat collector can be obtained by simply assuming that the absorbed energy ($I_т$) is distributed as follows:

$$I = Q + Q_{ттkз} \quad (1)$$

Where, $I_т$ is the total heat flux received by the insulator; $Q_т$ is the amount of useful heat recovered by the collector and carried away by the heat transfer medium; $Q_к$ is the amount of heat lost by the collector through various methods of transfer to the environment; $Q_з$ is the amount of heat stored by the collector in these various components.

In the case of constant thermal regime and for air collectors the amount of energy stored in different parts of the insulator is negligible, the previous relation takes the form:

$$I_и = Q_т + Q_к \quad (2)$$

Where, I_u is the radiant power received by the surface; $Q_т$ is the amount of useful heat recovered by the collector and carried away by the heat transfer medium; $Q_к$ is the amount of heat lost by the collector through various methods of transfer to the environment.

The modelling principle is to record the energy balances of each element that makes up the collector, absorber, glass, insulator and coolant. For this purpose, several models are encountered, two modelling approaches are also encountered.

Concluding in the recording of an overall energy balance for each collector element and their detailed approach, where balances are recorded for slices of the collector components.

Conductive transfer. The conductivity coefficients through the insulation and towards the back of the insulator are determined by the formula:

$$\frac{\mu_и}{e_и} \text{ and } \frac{\mu_п}{e_п} \quad (3)$$

Where,$\mu_и$ and$\mu_п$ - respectively thermal conductivity of polystyrene insulator; $e_и$ and $e_п$ - respectively thicknesses of polystyrene insulator [6-7].

Radiation transfer. Consider the transfer of radiation between two parallel plates with temperatures T_1 and T_2. Apply the formula:

$$h = \frac{\sigma(T_1+T_2)(T_1^2+T_2^2)}{\frac{1-\varepsilon_1}{\varepsilon_1}+\frac{1}{F_г}-\frac{1-\varepsilon_2}{\varepsilon_2}\frac{S_1}{S_2}} \quad (4)$$

Where, T_1 and T_2 are the absolute temperatures of the two faces considered to be homogeneous;ε is the emissivity of the medium i (surface S_i); $F_г$ is the geometric shape factor between surfaces S_1 and S_2 , usually taken to be 1 for different parts of the insulator;$\sigma$is the Stefan-Boltzmann constant.

The transfer of radiation between glass and air. The radiative transfer coefficient is written:

$$h_и = \frac{1}{2}\sigma\varepsilon_v(1-\cos\beta)\big((T_{ст.}+T_{воз.})(T_{ст.}^2+T_{ст.}^2)\big) \quad (5)$$

Where,β is the slope of the collector with respect to horizontal and temperature; $T_{ст.}$ is the glass temperature; $T_{воз.}$ is the equivalent air temperature.

$$T_{воз.} = 0{,}0552 \cdot T_{окр.}^{1.5} \quad (6)$$

In the case of a collector equipped with baffles, the length L in equation (7) is replaced by the distance x, the length of the row. The exchange coefficient between the metal plate and the heat transfer medium is determined by the formula:

$$h_{м.п.} = h_{п.с.} \quad (7)$$

Where, $h_{м.п.}$ is the convective transmission coefficient between the internal insulating metal plate and the heat transfer medium; $h_{п.с.}$ is the radiative transfer coefficient between the absorber and the glass.

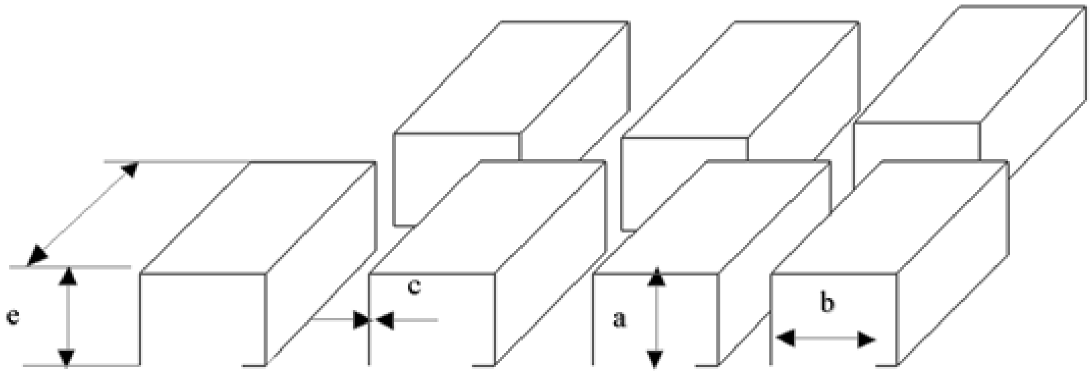


Fig.3. Geometry and arrangement of rectangular partitions studied and established

The air velocity in the movable duct $v_в$ depends on the geometrical parameters of the useful air flow and the shape of the baffles (Fig. 3), it is expressed:

$$v_{в} = \frac{m}{\rho_{об.} A_{в}} \text{ (flow rate)} \quad (8)$$

Where, $v_в$ - kinematic viscosity of the coolant; m-mass flow rate of the coolant; $\rho_{об.}$ - volumetric mass; $A_в$ - air passage section.

The air passage cross-section Aв in a moving air flow is calculated as follows:

Collectors without partitions:

$$A_{в} = LE \quad (9)$$

Where, *L-width of the* air plane sensor; *E-flow of* air circulating around the absorber.

The collector is equipped with baffles:

$$A_{в} = LE - \frac{L\,c}{b+c}\left(a + b + \frac{3c}{2}\right) \quad (10)$$

The cross-section of the air channel $A_в$ is determined. For a flat solar collector without baffles.

$$A_{в} = LE \quad (11)$$

For a flat air solar collector with rectangular baffles [8-13]:

$$A_{в} = LE - \frac{l}{b+c}\; c\,(a + b + c) \quad (12)$$

$$D_{г} = \frac{4\,LE}{2\,(L+E)} = \frac{2\,LE}{L+E} \quad (13)$$

Where, $D_г$ -hydraulic diameter of the air flow; *a-height of the* baffle; *b-length of the baffle*; *c-thickness of the* plate from which the baffles are made.

For a planar solar collector with rectangular baffles [8-10]:

$$D_{г} = \frac{4\,a\,b\,x}{2(x\,b + x\,a + x\,c) + b\,c} \quad (14)$$

The incident power. The incident power $I_т$, received by a square metre of insulator surface inclined at an angle β , can be determined theoretically, written [8-11]:

$$I_{т} = Y_{b\beta} + Y_{d\beta} \quad (15)$$

Where, $Y_{b\beta}$ is the direct component; $Y_{d\beta}$ is the diffuse component(s).

Thus, the power transmitted through the glazing is ($Y_{b\beta}\tau + Y_{d\beta}\tau_n$), then τ is the total directional transmittance, τ_n is the total hemispherical transparency. The absorber receives part of the transmitted power $\alpha_{пог.}$ and reflects part ($1-\alpha_{пог.}$) towards the glazing. If $r_{отр.}$ is the hemispherical reflectivity of the glazing, it reflects a fraction ($1-\alpha_{пог.}$) $r_{отр.}$ towards the absorber. The latter again absorbs a fraction $\alpha_{пог.}$ ($1-\alpha_{пог.}$) $r_{отр.}$ and reflects a fraction ($1-$)$\alpha_{пог.}^2$ $r_{отр.}$ towards the glazing. The fraction ($1-$)$\alpha_{пог.}^2$ $r_{отр.}^2$ is again reflected towards the absorber and so on.

The reflection fraction from the glass towards the absorber is equal to ($1-$)$\alpha_{пог.}^p$ $r_{отр.}^p$, then after infinity of reflections the captured power is equal to [8-13]:

$$\text{Y}_{пог.} = \sum_{p=0}^{\infty}(1-\alpha_{пог.})^p r_{отр.}^p\left(\phi_{b\beta}\tau + \phi_{d\beta}\tau_{п}\right) = \frac{\alpha_{ab}(\phi_{b\beta}\tau + \phi_{d\beta}\tau_{п})}{1-(1-\alpha_{пог.})r_{отр.}} \quad (16)$$

Results and discussion

Thus, the optical efficiency η_0 of an insulator, defined by the ratio /I$\phi_{пог.t}$, also referred to as the effective transmittance-absorption coefficient of the insulator and denoted ($\tau_c\alpha_{пог.}$):

$$\gamma_0 = \tau_{c}\alpha_{пог.} = \frac{\alpha_{пог.}(\phi_{b\beta}\tau + \phi_{d\beta}\tau_{п})}{(1-(1-\alpha_{пог.})r_{отр.})(\phi_{b\beta} + \phi_{d\beta})} \quad (17)$$

Then we determine the power $\phi_{пог.}$ per m^2 of the collecting surface of the insulator by the following formula:

$$\text{Y}_{пог.} = (\tau_{c}\alpha_{пог.})I_t \quad (18)$$

Calculation of loss coefficients. Heat losses occur due to the temperature difference between the absorber and the environment. They are manifested by three modes of heat transfer [9-17]. They are divided into three categories: direct losses, back losses and lateral losses. To simplify the determination of loss coefficients, the following assumptions are made:

- the thermal regime is constant;
- temperatures are the same all around the collector;
- the solar energy absorbed by the glass is negligible;
- conduction into the glass is negligible;
- the lateral losses are small compared to the front and rear losses.

These assumptions allow us to express the total absorber losses $Q_к$ per m^2 of collecting surface of the collector using the loss factor U_n , and the inverse, the resistance to heat transfer between potentials $T_{пог.}$ and $T_{окр.}$:

$$Q_{п} = U_{п}\left(T_{пог.} - T_{окр.}\right) = Q_{к1} + Q_{к2} \quad (19)$$

Where, $Q_{к1}$ is the amount of heat lost upstream of the collector; $Q_{к2}$ is the amount of heat lost downstream of the collector.

We define it as follows:

$$R_{с.окр.} = \frac{1}{h_{с.окр.}} \quad (20)$$

$$\text{R}_{с.пог.} = \frac{1}{\text{h}_{с.пог.}} \quad (21)$$

$$\text{R}_{изл.} = \frac{1}{\text{h}_{изл.}} \quad (22)$$

$$\text{R}_{соп.} = \frac{1}{\text{h}_{соп.}} \quad (23)$$

Where, $R_{с.окр.}$ is the thermal resistance between the glass and the environment due to convective exchange; $R_{с.пог.}$ is the thermal resistance between the absorber and the glass due to convective exchange; $R_{изл.}$ is the thermal resistance between the glass and the environment due to radiated exchange; $R_{соп.}$ is the thermal resistance between the absorber and the glass due to radiated exchange?

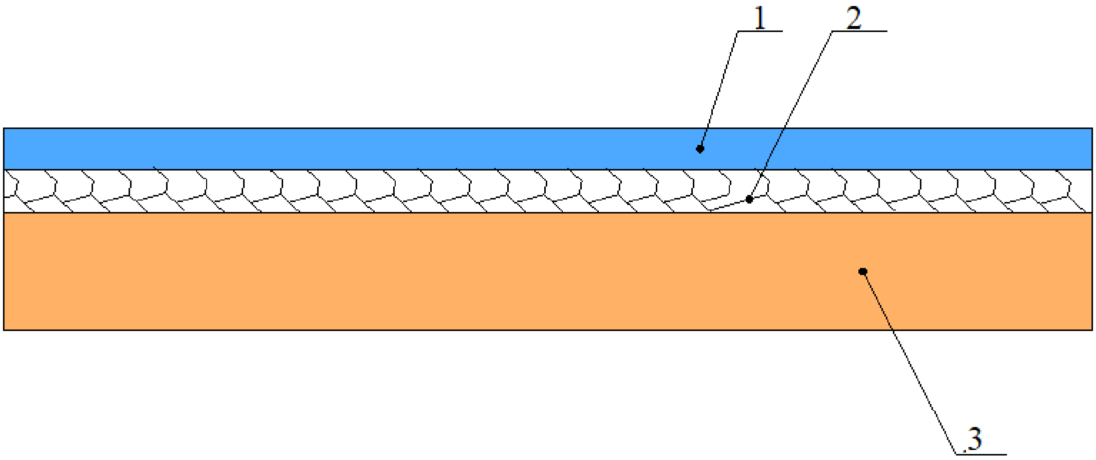
Calculation of the direct loss factor. If only the effect of wind and the radiation of the glass with sky vault are taken into account, the equivalent thermal resistance on the collector face is written:

$$\frac{1}{R_{\text{с.окр.}}} = \frac{1}{R_{\text{с.окр.}}} + \frac{1}{R_{\text{изл.}}} = h_{\text{с.окр.}} + h_{\text{изл.}} = U_{\text{т.п.}-1} \quad (24)$$

$$\frac{1}{\mathrm{R_{соп.}}} = \frac{1}{\mathrm{R_{с.окр.}}} + \frac{1}{\mathrm{R_{изл.}}} = \mathrm{h_{с.окр.}} + \mathrm{h_{изл.}} = \mathrm{U_{т.п.-2}} \quad (25)$$

So,

$$\mathrm{R_{с.окр.}} = \mathrm{R_{соп.}} + \mathrm{R_{с.окр.}} = \frac{1}{\mathrm{U_{т.п.-1}}} + \frac{1}{\mathrm{U_{т.п.-2}}} \quad (26)$$



1-metal sheet; 2-foam; 3-back wall.

Fig.4. Schematic of the insulating layers on the back side of the collector

Thus, the total heat loss coefficient at the front of the collector is determined by the following relationship:

$$U_{\text{т.п.}} = \frac{1}{R_{\text{с.окр.}}} = \left(\frac{1}{U_{\text{т.п.}-1}} + \frac{1}{U_{\text{т.п.}-2}}\right)^{-1} \quad (27)$$

Calculation of the total heat loss coefficient U_n . The total outward heat loss coefficient is the sum of two coefficients:

$$U_n = U_{m.n.} + U_m \quad (28)$$

We take into account all heat exchanges inside the collector, such as radiation and convective exchange and in the case $h_{n.c.} = h_{м.n.}$ the expression of the total loss factor and after simplification is written in the following form:

$$U_{\text{п}} = \frac{(U_{\text{т.п.}} + U_{\text{т}})(h_{\text{п.с.}} + 2h_{\text{м.п.}}) + 2U_{\text{т.п.}}U_{\text{т}}}{U_{\text{т.п.}} + 2h_{\text{и.п.}} + h_{\text{п.с.}}} \quad (29)$$

The algorithm for calculating the performance of a solar air collector consists of the following steps [10-13].

Data Introduction:

- collector data (characteristics and dimensions);
- collector tilt;
- meteorological data, I_t , $T_{окр/}$, V_c , H, etc.;
- inlet temperature of the heat transfer medium $T_{m.вх}$.
- mass flow rate of the working body (coolant).

Initialisation of average temperatures:

- $T_т$ - temperature of the heat transfer medium;
- $T_{cm.}$ -temperature glass;
- $T_{пог.}$ -temperature of the absorber;
- $T_{м.п.}$ -temperature of the metal plates placed on the insulation.

With forced convection and in the case of rectangular ducts, the exchange between the absorber and the heat transfer medium is characterised by a coefficient:

$$h_{\text{п.с.}} = \frac{Nu\mu_f}{D_{\text{г}}} \quad (30)$$

Calculating the convective transfer coefficient $h_{к.п.}$ due to air. To calculate the coefficient $h_{к.п.}$ we use the following equation:

$$h_{\text{к.п.}} = 5{,}67 + 3{,}86\ v_{\text{с}}$$

Calculating the convective transfer coefficient $h_{n.c.}$ To calculate the coefficient $h_{n.c.}$ we use the following equation:

$$h_{\text{п.с.}} = \frac{Nu\mu_{\text{в}}}{D_{\text{г}}} \quad (31)$$

For collectors equipped with baffles, $h_{n.c.}$ - consider the average exchange coefficient, it is also calculated by formula (30). Nusselt number is calculated on the basis of the following Keys correlation [13-15]:

$$Nu = 0{,}0158\ Re^{0.8} \qquad \text{для} \quad L/D_{\text{г}} > 10 \quad (32)$$

Where the Reynolds number is defined by the formula:

$$R_e = v_{\text{в}} \frac{D_{\text{г}}}{v_{\text{в}}} \quad (33)$$

Calculation of the natural conduction-convection loss coefficient $h_{e.к.}$. To calculate the coefficient $h_{e.к.}$ we use the following equation:

$$h_{\text{е.к.}} = 1{,}42\left(\frac{(T_{\text{пог.}} - T_{\text{окр.}})\sin\beta}{L}\right) \quad (34)$$

Calculation of radiative transfer coefficients, a-calculation *of* radiative transfer coefficients h_u.

Radiation transfer coefficient h_u is calculated by the formula:

$$h_{и} = \sigma\varepsilon_{и}(T_{\text{ст.}} + T_{\text{воз.}})(T_{\text{ст.}}^2 + T_{\text{воз.}}^2) \quad (35)$$

Where, σ-calculation of radiative transfer coefficients $h_{\text{п.с.}}$.

Radiation transfer coefficient $h_{n.c.}$ is calculated by the formula:

$$h_{\text{п.с.}} = \frac{\sigma(T_{\text{ст.}} + T_{\text{пог.}})(T_{\text{ст.}}^2 + T_{\text{пог.}}^2)}{\frac{1}{\varphi_{и}} + \frac{1}{\varphi_{\text{пог.}}} - 1} \quad (36)$$

Radiation transfer coefficient $h_{u.n.}$ is calculated by the formula:

$$h_{\text{и.п.}} = \frac{\sigma(T_{\text{пог.}} + T_{\text{м.п.}})(T_{\text{пог.}}^2 + T_{\text{м.п.}}^2)}{\frac{1}{\varphi_{\text{п.м.п.}}} + \frac{1}{\varphi_{\text{м.п.}}} - 1} \quad (37)$$

Where, $\varepsilon_{\text{м.п.}}$ is the emission coefficient of the metal plate on the absorber side; ε p.m.p.-the emission coefficient of the absorber on the metal plate side.

Calculation of the front loss factor $U_{m.n.}$. Using the following relationship allows you to calculate the loss $U_{m.n.}$ of the front face of the insulator:

$$U_{\text{т.п.}} = \frac{1}{\frac{1}{h_{\text{к.п.}}} + \left[\frac{N_v}{\left[\frac{C}{T_{\text{пог.}}}\left(\frac{T_{\text{пог.}} + T_{\text{окр.}}}{N_{\text{ст.}} + f}\right)\right]^{0.33}}\right]} + \frac{\sigma(T_{\text{окр.}}^4 - T_{\text{пог.}}^4)}{\frac{1}{\varphi_{\text{п.м},\beta} + 5\cdot 10^{-2}\cdot N_{\text{ст.}}\cdot(1 - \varphi_{\text{п.м},\beta})} + \frac{2\cdot N_{\text{ст.}} + f - 1}{\varphi_{и,\beta}} - N_{\text{т}}} \quad (38)$$

Calculating the losses $U_{\text{т}}$ on the underside of the insulator. Using the following relationship, the losses $U_{\text{т}}$ can be calculated.

$$U_{\text{т}} = \frac{1}{\sum_{i=1}^{i=n}\frac{e_i}{k} + \frac{1}{h_{\text{в}}}} \quad (39)$$

Calculation of the U coefficient$_{\text{п}}$. Taking into account the radiated exchange losses, the total *U factor*$_{\text{п}}$ of the insulator is written:

$$U_{\text{п}} = \frac{(U_{\text{т.п.}} + U_{\text{т}})(h_{\text{п.с.}} + 2h_{\text{и.п.}}) + 2U_{\text{т.п.}}U_{\text{т}}}{U_{\text{т.п.}} + 2h_{\text{и.п.}} + h_{\text{п.с.}}} \quad (40)$$

Calculation of the efficiency factor $F'_{э.п.}$ taking into account radiated exchanges:

$$F'_{\text{э.п.}} = \frac{h_{\text{и.п.}}h_{\text{п.с.}} + h_{\text{м.п.}}U_{\text{т.п.}} + h_{\text{и.п.}}h_{\text{и.п.}} + h_{\text{п.с.}} + h_{\text{м.п.}}}{(U_{\text{п}} + h_{\text{и.п.}} + h_{\text{п.с.}})(U_{\text{п}} + h_{\text{м.п.}} + h_{\text{и.п.}}) - h_{\text{и.п.}}^2} \quad (41)$$

For: $h_{м.п.} = h_{п.с.}$ we end up with:

$$F'_{\text{э.п.}} = \frac{h_{\text{п.с.}}(U_{\text{т.п.}} + 2h_{\text{и.п.}} - h_{\text{п.с.}})}{(U_{\text{п}} + h_{\text{и.п.}} + h_{\text{п.с.}})(U_{\text{п}} + h_{\text{п.с.}} + h_{\text{и.п.}}) - h_{\text{и.п.}}^2} \quad (42)$$

Calculation of the conductivity coefficient $F_{в.п.}$. This coefficient, reflecting the conductivity of the absorber, is calculated using the following equation:

$$F_{\text{в.п.}} = \frac{\dot{m}C_{\text{т}}}{AU_{\text{п}}}\left(1 - \left(-\frac{F'_{\text{э.п.}}U_{\text{п}}A}{\dot{m}C_{\text{т}}}\right)\right) \quad (43)$$

Calculation of the useful power $Q_{\text{т}}$, provided by the insulator. To calculate the useful power $Q_{\text{т}}$, provided by the insulator, we use the following relationship:

$$Q_{\text{т}} = F_{\text{в.п.}}\left[(\tau_{\text{с}}\alpha_{\text{пог.}})I_{и} - U_{\text{п}}(T_{\text{т.вх.}} - T_{\text{окр.}})\right] \quad (34)$$

Recalculates the average temperature of the heat transfer medium $T_{\text{т}}$. Using Klein's relation, the average temperature of the heat transfer medium can be calculated as follows:

$$T_{\text{т}} = T_{\text{т.вх.}} + \frac{Q_{\text{т}}}{U_{\text{п}}F_{\text{в.п.}}}\left(1 - \frac{F_{\text{в.п.}}}{F'_{\text{э.п.}}}\right) \quad (35)$$

Calculation of the average temperature of metal plates $T_{м.п.}$, [8-13].

a. Collector to parts,

$$T_{\text{м.п.}} = T_{\text{т}} + \frac{h_{\text{и.п.}}(\tau_{\text{с}}\alpha_{\text{пог.}})I_{и}}{(U_{\text{т.п.}} + h_{\text{и.п.}} + h_{\text{п.с.}})(U_{\text{т.п.}} + h_{\text{м.п.}} + h_{\text{и.п.}}) - h_{\text{и.п.}}^2} + \frac{-(T_{\text{т}} - T_{\text{окр.}})(U_{\text{т.п.}}U_{\text{т}} + U_{\text{т}}h_{\text{п.с.}} + h_{\text{и.п.}}(U_{\text{т.п.}} + U_{\text{т}}))}{(U_{\text{т.п.}} + h_{\text{и.п.}} + h_{\text{п.с.}})(U_{\text{т}} + h_{\text{м.п.}} + h_{\text{и.п.}}) - h_{\text{и.п.}}^2} + \frac{(T_{\text{т}} + 273)(h_{\text{м.п.}} - h_{\text{п.с.}})(U_{\text{т.п.}} + h_{\text{и.п.}} + h_{\text{п.с.}})}{(U_{\text{т.п.}} + h_{\text{и.п.}} + h_{\text{п.с.}})(U_{\text{т}} + h_{\text{м.п.}} + h_{\text{и.п.}}) - h_{\text{и.п.}}^2} \quad (36)$$

For, $h = h_{м.п.п.с.}$

We'll have:

$$T_{\text{м.п.}} = T_{\text{т}} + \frac{h_{\text{и.п.}}(\tau_{\text{с}}\alpha_{\text{пог.}})I_{и}}{(U_{\text{т.п.}} + h_{\text{и.п.}} + Xh_{\text{м.п.}})(U_{\text{т}} + h_{\text{м.п.}} + h_{\text{и.п.}}) - h_{\text{и.п.}}^2} + \frac{-(T_{\text{т}} - T_{\text{окр.}})(U_{\text{т.п.}}U_{\text{т}} + U_{\text{т}}h_{\text{п.с.}} + h_{\text{м.п.}}(U_{\text{т.п.}} + U_{\text{т}}))}{(U_{\text{т.п.}} + h_{\text{и.п.}} + h_{\text{п.с.}})(U_{\text{т}} + h_{\text{м.п.}} + h_{\text{и.п.}}) - h_{\text{и.п.}}^2} \quad (37)$$

Where, $T_{окр.}$ - ambient air temperature; $U_{\text{т.п.}}$ - heat loss coefficient at the front of the absorber.

b. Collectori is equipped with partitions,

$$T_{\text{м.п.}} = T_{\text{т}} + \frac{h_{\text{и.п.}}(\tau_{\text{с}}\alpha_{\text{пог.}})I_{и}}{(U_{\text{т.п.}} + h_{\text{и.п.}} + Xh_{\text{п.с.}})(U_{\text{т}} + h_{\text{м.п.}} + h_{\text{и.п.}}) - h_{\text{и.п.}}^2} + \frac{-(T_{\text{т}} - T_{\text{окр.}})(U_{\text{т.п.}}U_{\text{т}} + U_{\text{т}}Xh_{\text{п.с.}} + h_{\text{и.п.}}(U_{\text{т.п.}} + U_{\text{т}}))}{(U_{\text{т.п.}} + h_{\text{и.п.}} + Xh_{\text{п.с.}})(U_{\text{т}} + h_{\text{м.п.}} + h_{\text{и.п.}}) - h_{\text{и.п.}}^2} + \frac{(T_{\text{т}} + 273)(h_{\text{м.п.}} - Xh_{\text{п.с.}})(U_{\text{т.п.}} + h_{\text{и.п.}} + Xh_{\text{п.с.}})}{(U_{\text{т.п.}} + h_{\text{и.п.}} + Xh_{\text{п.с.}})(U_{\text{т}} + h_{\text{м.п.}} + h_{\text{и.п.}}) - h_{\text{и.п.}}^2} \quad (38)$$

In the case where: $h_{м.п.} = h_{п.с.}$, we end up with:

$$T_{\text{м.п.}} = T_1 + T_2 \quad (39)$$

Such as :

$$T_1 = \frac{h_{\text{и.п.}}(\tau_{\text{с}}\alpha_{\text{пог.}})I_{и} - (T_{\text{т}} - T_{\text{окр.}})(U_{\text{т.п.}}U_{\text{т}} + XU_{\text{т}}h_{\text{п.с.}} + h_{\text{и.п.}}(U_{\text{т.п.}} + U_{\text{т}}))}{(U_{\text{т.п.}} + h_{\text{и.п.}} + Xh_{\text{п.с.}})(U_{\text{т}} + h_{\text{п.с.}} + h_{\text{и.п.}}) - h_{\text{и.п.}}^2} \quad T_2 = \frac{(T_{\text{т}} + 273)h_{\text{п.с.}}(1 - X)(U_{\text{т}} + h_{\text{и.п.}} + Xh_{\text{п.с.}})}{(U_{\text{т.п.}} + h_{\text{и.п.}} + Xh_{\text{п.с.}})(U_{\text{т}} + h_{\text{п.с.}} + h_{\text{и.п.}}) - h_{\text{и.п.}}^2} \quad (40)$$

Recalculates the average absorber temperature $T_{пог.}$. According to the energy balance,

$$(T_{\text{пог.}} + 273) = \frac{(1 + X)(T_{\text{т}} + 273) + \left(\frac{Q_{\text{т}}}{h_{\text{п.с.}}}\right)(T_{\text{м.п.}} + 273)}{X} \quad (41)$$

Calculation of the average glass temperature T_{cm}. According to the energy balance, the glass temperature $T_{cm.}$ is recorded:

$$T_{\text{ст.}} + 273 = \frac{\alpha_{\text{пог.}}\, I_{\text{и}} + (T_{\text{пог.}} + 273) + \left(h_{\text{п.с.}} + \frac{h_{\text{е.к.}}}{2}\right)(T_{\text{окр.}} + 273)h_{\text{к.п.}}}{h_{\text{п.с.}} + \frac{h_{\text{е.к.}}}{2} + h_{\text{к.п.}} + h_{\text{и}}} + \frac{(T_{\text{воз.}} + 273)h_{\text{и}}}{h_{\text{п.с.}} + \frac{h_{\text{е.к.}}}{2} + h_{\text{к.п.}} + h_{\text{и}}} \quad (42)$$

Calculation of the temperature of the heat transfer medium at the collector outlet.

$$T_{\text{т.вы.}} = T_{\text{т.вх}} + \frac{AQ_{\text{т}}}{\dot{m}C_{\text{т}}} \quad (43)$$

Calculating the thermal efficiency of an insulatorγ:

The thermal efficiency of an insulator is the ratio of useful power to incident power:

$$\gamma = \frac{Q_{\text{т}}}{I_{\text{и}}} = \frac{Q_r}{A\, I_{\text{и}}} \quad (44)$$

Conclusion

It can also be determined from the ratio of the temperature difference ($T_{т.вх.}$ - $T_{окр.}$) to the total incident flux $I_и$:

- it follows from the recorded values that the ambient temperature affects the efficiency, and this depends on the incident solar radiation;

- the change in the average temperature of the absorber plate has a linear function depending on the absorbed power. This temperature increases as a function of the amount of power absorbed;

- temperature of the absorber with baffles is higher than in the case of no baffles, the thermal efficiency of the collector is important at significant values of the coolant flow rate. Thus, the yield obtained by modelling and the yield obtained experimentally are very close;

- in the case of a transparent cover with a thickness of 3 mm, a gain of 10 per cent was recorded compared to another thickness of 6 mm;

- the right choice of absorber material ensures good collector efficiency; this is what justifies our choice: a gain of almost 14% in the case of a steel absorber compared to a copper absorber and 28% in the case of an aluminium absorber;

- deflection of the baffles allows increasing the temperature of the absorber. This increase is due to the increase of the absorber's contact surface with air.

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