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REGULAR ARTICLE

Analysis of Thermal Processes in the Heat Exchange Unit of a Combined Photovoltaic Plant with Solar Radiation Concentration

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This paper is devoted to improve the design solution of combined thermo-photovoltaic systems which works in concentrated solar radiation. The main problem is ensuring the efficient heat removing from the photoreceiving surface of such systems and transferring this heat energy to liquid heat carrier. In case of using concentrated solar radiation with high density of energy onto the photoreceiving surface the problem of intensifying the heat exchange processes is critical for ensuring the efficient and long-term work of combined thermo-photovoltaic systems. Based on the analysis of thermal processes, the design of a heat exchange unit for a combined photovoltaic plant is proposed. The main design feature of proposed solution is microchannels made on the back plate of photoreceiving surface. It is shown that this design creates a transient regime of coolant flow, which allows efficient cooling of solar cells under conditions of concentrated solar radiation and potentially will allow to receive record values of heat exchange coefficient between the coolant and the upper plate of the radiator. An analysis technique using criterial hydrodynamic equations and numerical modeling were used; the results obtained in both cases are in good agreement. The efficiency, thermal and electrical characteristics of a combined photovoltaic system with solar radiation concentration have been assessed. Preliminary estimations have been made for the specific parameters of installed energy characteristics of a heat-electric power plant, which under AM1 conditions and 400 times the concentration of solar radiation for solar cells with an efficiency of 30 % have the following values: electrical power 650 W, thermal power 2370 W, total installed net power 3020 W, overall efficiency 87 % which are significantly higher than parameters achieved for existing solar power plants.

Keywords: PV/T system, Thermal process, Heat exchange, Solar energy, Concentrated radiation, Cooling.

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1. INTRODUCTION

The development of photovoltaic installations operating at high concentrations of solar radiation is one of the directions for reducing the cost of the electrical energy they produce [1]. In such installations, this is achieved by reducing the area of semiconductor device structures in which solar energy is converted into electrical energy [2, 3]. The base semiconductor material of such structures is traditionally gallium arsenide, which provides the highest efficiency of photoelectric conversion of solar energy among semiconductor structures. Thus, in [4] it was shown that multistage solar cells (SCs) based on gallium arsenide achieve an efficiency of 39.2 % under AM1 lighting conditions and 47.1 % at 143 times the concentration of solar radiation.

In conditions of concentrated radiation, cooling of solar cells is necessary, since their efficiency decreases with increasing temperature. Moreover, under conditions of strong concentration of solar radiation, an increase in temperature can lead to the destruction of multi-stage solar cells. Therefore, when using concentrated solar radiation, it becomes important to ensure the removal of thermal energy from the solar cell. The rate of thermal energy removal must be sufficient to limit the operating temperature of the solar cell to a given level, which is determined by the temperature reduction coefficient of the photovoltaic conversion efficiency. Currently, photovoltaic installations are being commercially produced, in which Fresnel lenses are used as a concentration system [5], concentrating solar radiation onto solar cells with an area (0.5-1) cm^2. In these installations, massive copper plates are used as heat exchangers for heat removal, ensuring the dissipation of thermal energy from the solar cell into the surrounding space [5, 6]. This design of a photovoltaic installation does not provide for the recovery of thermal energy. At the same time, the utilization of thermal energy will significantly increase competitive advantages and reduce the payback period for photovoltaic installations operating in conditions of concentrated solar radiation.

Recently, a number of companies have launched the industrial production of solar cells and modules based on gallium arsenide with an efficiency of more than 30%, and with a fairly large area [7], which fundamentally allows them to be compactly placed on the surface of one compact heat exchange block, in which heat transfer occurs. energy from the solar cell to the coolant. This paper examines the energy balance of a photovoltaic installation operating under conditions of concentrated solar radiation, and based on

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an analysis of thermal processes, a design of a heat exchange unit with "micro" channels is proposed.

2. ENERGY BALANCE OF A PHOTOVOLTAIC INSTALLATION

Energy is supplied to the surface of the heat-photo-energy block from the solar radiation concentrator, Q, which is partially converted into electrical energy Q_E using photoelectric converters, and partly into thermal energy Q_T.

$$Q = Q_E + Q_T, \tag{1}$$

$$Q_E = r_z \alpha_1 \eta S_{ph} K Q_s, \tag{2}$$

$$Q_T = [\alpha S_2 + (1 - \eta - r_{ph}) S_{ph}] r_z K Q_s, \tag{3}$$

where Q_s – solar power per unit surface, $K = S_a/S_p$ – concentration coefficient, S_f – focal spot area, S_a – concentrator aperture area, r_z – concentrator reflection coefficient, η – efficiency coefficient of solar cell, S_{ph} – solar cell area, r_{ph} – reflection coefficient from the solar cell surface, α_1 – solar cell surface absorption coefficient, $S_2 = S_f - S_{ph}$ – surface area in the focal spot not occupied by the solar cell, α – free surface absorption coefficient.

The thermal energy received on the surface of the heat exchange block will be partially transferred to the coolant Q_f, and partially will be lost due to thermal radiation Q_c and convection heat exchange with the atmosphere Q_w:

$$Q_T = Q_f + Q_w + Q_c. \tag{4}$$

Since the surface of the working block is not covered with anything, complete heat exchange with the atmosphere Q_a, determined by one equation:

$$Q_a = Q_w + Q_c = h_w[(T_p - T_a)S_{ph} + [(T_{p1} - T_a)S_2 + (T_f - T_a)(S_l + S_p)] + \sigma_0[\varepsilon_p(T_p^4 S_{ph} + T_{p1}^4 S_2) + \varepsilon_{pb}(S_l + S_p)T_f^4 - (\varepsilon_{pb}(S_p + S_l) + \varepsilon_p(S_{ph} + S_2))T_s^4], \tag{5}$$

where $\sigma_0 = 5.6697\ 10^{-8}$ W/(m²·K⁴) – Stefan-Boltzmann Constant, ε_p – degree of emissivity of the working surface of the heat exchange block, ε_{pb} – degree of emissivity of the non-working surface of the heat exchange block, $h_w = 5.7 + 3.8V$ – coefficient of convection heat exchange between the surface and the atmosphere [8], V – wind speed m/s), T_p – surface temperature of the heat exchange block, T_{p1} – temperature of the free surface of the heat exchange block, T_a – atmospheric temperature, T_f – average coolant temperature, T_s – sky high temperature, S_l – lateral surface area of the heat exchange block.

The terms in Q_a, including T_f, take into account the heat exchange of the back side of the heat exchange block with the atmosphere. The temperature of the back side of the heat exchange unit, due to the high thermal conductivity of the housing material, can be approximately taken to be equal to the temperature of the coolant. Negative terms in Q_a take into account the arrival of the scattered component of infrared radiation from the atmosphere on the surface of the heat exchange block.

The formula for calculating heat losses can be simplified if we take only the temperatures of the illuminated light as different (T_p) and unlighted surface (T_f). This will lead to a slightly larger loss, but, as follows from the calculations, these losses still remain less than 1 – 2 %.

To assess the possible values of thermal energy that must be removed from solar cells in order to provide them with an acceptable operating temperature, it is necessary to set the external conditions, the possible overall dimensions of the heat exchange unit and the parameters of the concentration system.

For reasons of the need to remove peak heat loads, we select environmental parameters characteristic of southern latitudes (Table 1).

Based on the characteristics of solar cells based on gallium arsenide, which are produced industrially and are available on the market, it is possible to set the necessary geometric dimensions of the heat exchange unit for their placement. We will consider a heat exchange unit to accommodate two solar cells, which are halves of a square with cut corners [7, 9].

Table 1 – Atmospheric parameters

Parameter	Values
Maximum instantaneous solar radiation power Q_s, W/m²	1000
Atmospheric temperature T_a, °C	30
Sky temperature, for the summer period T_s, K	290
Average wind speed V, m/s.	5

Table 2 – Calculation of the dimensions of the heat exchange unit, concentrator and energy collected by the concentrator and generated by electrical and thermal systems

Heat exchange block dimensions	
Length of the diagonal of the area occupied by the solar cell, cm	10.2
Heat exchange block length H, cm	9.7
Heat exchange block width W, cm	9.7
Heat exchange block thickness L, cm	1
Surface area of the heat exchange block $S_p = H \times$ W, cm²	94.1
Side surface area of the heat exchange block $S_l = 2\ (H + \text{W})L$, cm²	38.8
Length of the cooled part of the surface of the heat exchange block l, cm	8.2
Width of the cooled part of the surface of the heat exchange block h, cm	8.2
Active area of solar cell S_{ph}, cm²	30.15
Area required to accommodate two solar cells S, cm²	60.3
Dimensions of the concentrator and the collected energy	
Geometric concentration coefficient $K = 4S_a/\pi d^2$	400
Focal spot diameter d, cm	10.5
Focal spot area $S_f = \pi d^2/4$, cm²	86.6
Concentrator aperture area S_a, m²	3.46
Concentrator diameter d, m	2.1
Concentrator surface reflectance r_z	0.95
Energy collected by the concentrator $Q = r_z S_a Q_s$, W	3290
Calculation of generated electrical energy	
Efficiency coefficient of SC η, %	30
SC absorption coefficient α_1	0.95
Expected electrical power Q_E, W	653

Calculation of the received thermal energy	
Absorption coefficient of the free illuminated surface of the heat exchange block α_2	0.9
Free illuminated surface area of the heat exchange block $S_2 = S_f - S$, cm^2	26.3
Thermal energy absorbed by the solar system and the free illuminated surface of the heat exchange block Q_T, W	2389
Total absorbed energy $Q_{ET} = Q_E + Q_T$, W	3042
Energy reflected from the surface $Q_l = Q - Q_{ET}$, W	248

The shape of the site for placing the solar cell will also be close to a square, which is most optimal when using a concentrating system with a round focal spot. The concentration system parameters can be set based on the geometric concentration coefficient, which will then determine the required concentrator aperture area (based on the required focal spot size) and the energy collected by it. Next, Table 2 shows the main geometric dimensions of the heat-electric-energy unit, which determine the values of thermal and electrical energy production, as well as calculations of achievable energy production values.

Thus, for a concentrating system comparable to the size of a standard solar battery, we have an estimate of the possible values of electric power generation Q_E = 653 W. In this case, the solar energy system and the free illuminated surface of the heat exchange unit will absorb thermal energy Q_T = 2389 W. Received value Q_T is the main initial data for calculating the necessary characteristics for heat removal from the thermo-photo-energy unit.

3. ANALYSIS OF THERMAL PROCESSES IN THE HEAT EXCHANGE BLOCK

The main element of a photovoltaic installation is a photodetector, which is a solar module located on the surface of a heat exchange unit. The photodetector ensures the conversion of solar energy into electrical energy, through a solar module, and into thermal energy, due to the heat exchange of heated surfaces with the coolant. Since a large amount of energy is concentrated on a small area, in order to achieve acceptable temperatures of the photovoltaic converters that are part of the solar module, it is necessary to have a fairly intense heat exchange between the surface on which the energy is supplied and the coolant. To fulfill this requirement, there are two options that aim to:

– increasing the heat transfer area by increasing the surface of the radiator; as a rule, this can be achieved using plate radiators;

– creation of conditions for turbulent or transient coolant flow.

As calculations have shown, a significant increase in the heat transfer surface requires the use of high radiator plates. Moreover, in order to achieve the required heat transfer area, under laminar flow conditions, the height of the fin must be such that the temperature difference across it will be more than 15 °C, which significantly reduces the efficiency of heat transfer. In addition, with high plates it is difficult to ensure even distribution of coolant throughout all radiator channels. Therefore, a design was chosen with a smaller effective heat transfer area, but with the possibility of increasing the heat transfer coefficient by creating a turbulent flow.

The heat flow during heat transfer from the surface of the radiator to the liquid can be represented using Newton's heat transfer law:

$$Q_f = h_f(T_w - T_f)S_r, \tag{6}$$

where h_f – heat exchange coefficient between the coolant and the body of the heat exchange unit, S_r – radiator surface area, T_w – average radiator wall temperature.

Using formulas (6), (4) and (5), you can determine the required value of the heat transfer coefficient from the radiator surface to the coolant:

$$h_f = (Q_T - Q_a)/(T_w - T_f)/S_r. \tag{7}$$

Another important characteristic that determines the required pump performance is the coolant mass flow m'. This value is determined from the condition of equality of the energy transferred to the coolant from the heat transfer surface of the radiator and accumulated by it due to the heat capacity:

$$Q_f = m'C_{pw}(T_{fo} - T_{fi}) \Rightarrow m' = (Q_T - Q_a)/C_{pw}(T_{fo} - T_{fi}), \tag{8}$$

where C_{pw} – heat capacity of water J/(kg·K), T_{fi} and T_{fo} – water temperature at the inlet and outlet of the heat exchange block, respectively.

Knowing the mass flow rate, you can calculate the flow speed

$$w = m'/\rho S_n = (Q_T - Q_a)/C_{pw}(T_{fo} - T_{fi})/\rho S_n, \tag{9}$$

where ρ – coolant (water) density, $S_n = nS_1$ – total cross-sectional area for coolant passage, S_1 – cross-sectional area of one radiator channel, n – number of channels.

Next, use the criterion equations of hydrodynamics to calculate the heat transfer coefficient from the radiator surface to the coolant. These equations look like this:

$$h_f = \lambda Nu/d_e, \tag{10}$$

where $d_e = 4S_1/P$ – effective channel diameter, P – channel intersection perimeter, λ – thermal conductivity coefficient of the coolant, Nu – Nusselt number, which for the transition regime is equal to [10-12]:

$$Nu = K_0\varepsilon_l Pr^{0.43}(Pr_f/Pr_w)^{0.25}, \tag{11}$$

and for turbulent

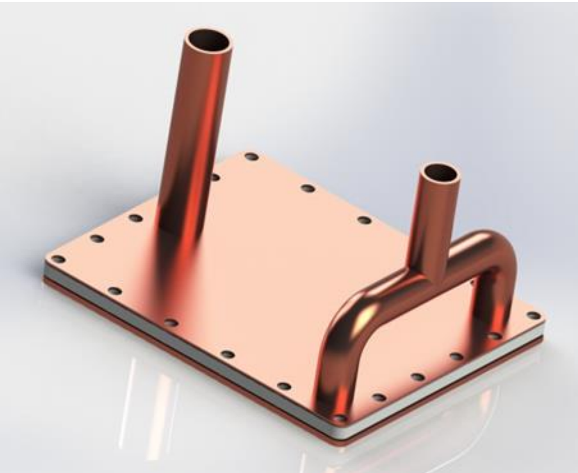
$$Nu = 0.021\varepsilon_l Re^{0.8}Pr^{0.43}(Pr_f/Pr_w)^{0.25}. \tag{12}$$

Here ε_l – coefficient depending on the ratio of channel length to effective diameter, $Re = wd_e/\nu$ – Reynolds number, w – flow rate, ν – kinematic viscosity coefficient, complex K_0 depends on Reynolds number, Pr_f and Pr_w – Prandtl numbers at liquid and wall temperatures, respectively.

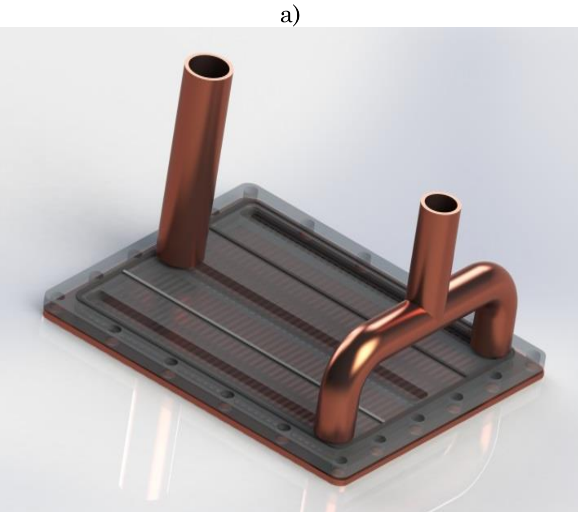
Value negotiation h_f obtained by formulas (7) and (10), is produced by varying the average temperature of the coolant and (or) the temperatures of the coolant at the inlet and outlet of the heat exchange unit.

To increase the heat transfer coefficient, it is necessary to create conditions for a turbulent or transient flow regime of the coolant. This can be achieved by increasing the fluid flow rate in the channels between the radiator plates, which is possible by reducing the effective diameter of the channel [13-15]. An image of a model of a heat exchange unit with a plate radiator is shown in Fig. 1.

The heat exchange unit has one inlet and an outlet manifold that combines two outlets for the coolant. The flow distribution along the "microchannels" of the radiator occurs from the central channel, which allows doubling the effective cross-section for the flow of flow, and significantly reducing the hydrodynamic resistance, thereby reducing the power requirements of the pump for pumping the coolant.



a)



b)

Fig. 1 – Image of the heat exchange block model a) bottom view; b) view in the middle

Table 3 shows the selected geometric parameters of the heat exchange block, which make it possible to achieve the required values of the heat transfer coefficient, and provides calculations of thermal processes that demonstrate this. Calculations were carried out for the maximum expected value of the concentration coefficient and, accordingly, for the maximum value of thermal energy input that must be removed (see Table 2). In addition, the temperature of the coolant at the inlet is taken to ensure the most difficult cooling conditions that may arise for a closed cooling system closer to the end of the daily operating cycle.

As a result of the calculations carried out for the radiator of the heat exchange unit, the following peak values were obtained: water flow ~8.6 l/min, flow velocity in the spaces between the plates is 2.06 m/s, Reynolds number $Re = 2675$, which corresponds to the transient flow regime. Calculation of the heat transfer coefficient between the coolant and the top plate of the radiator gives $h_f = 9556$ W/(m^2·K). Comparing this value with the required value calculated using the formula (6) ($h_f = 9420$ W/(m^2·K)), we can conclude that the selected design of the heat exchange unit and its selected parameters are capable of providing cooling mode at peak heat flow.

Table 3 – Calculation of characteristics of the heat exchange unit

Parameter	Values
Basic geometric characteristics of the radiator of the heat exchange unit	
Surface area of the heat exchange block S_p, cm^2	94.1
Free illuminated surface area of the heat exchange block S_2, cm^2	26.3
Side surface area of the heat exchange block S_l, cm^2	38.8
Length of the cooled part of the surface of the heat exchange block l, cm	8.2
Width of the central distribution channel of the radiator w_1, mm	6
Length of the "micro channel" of the radiator $l_1 = (h - w_1)/2$, cm	3.8
Width of the "micro channel" of the radiator a, mm	0.5
Radiator fin width w, mm	1
Radiator fin height b, mm	1.3
Plate thickness in the channel area t, mm	0.7
Number of "micro radiator channels", $N = 2l/(a + w)$	110
Radiator surface area $S_r = N(bw + l_1(a + 2b)) + hw_1$, cm^2	135.93
Coefficient of increase in surface area from which heat is removed $k = S_r/hl$	2.02
Channel cross-sectional area $S_1 = ab$, cm^2	0.0065
Total cross-sectional area for water flow $S_n = NS_1$, cm^2	0.715
Effective diameter $d_e = 4S_1/P$, cm	0.00722
Main thermal characteristics of the radiator of the heat exchange unit	
Absorption coefficient of the free illuminated surface of the heat exchange block α	0.9
Thermal energy absorbed by the PV system and the free illuminated surface of the heat exchange block Q_T, W	2389
PV operating temperature T_p, °C	70
Thickness of the germanium substrate of the solar cell, μm	200
Thermal conductivity coefficient of germanium, W/m	55
Copper thermal conductivity coefficient, W/m	400
Temperature drop across the thickness of the substrate and heat-conducting paste, °C	1.4
Temperature drop across the thickness of the copper body of the block, °C	0.58

Temperature drop along the length of the copper fin of the radiator, °C	1.1
The average temperature drop under the surface of the solar cell before contact with the coolant can be taken as, °C	3.1
The temperature of the free surface of the heat exchange block will be less than the surface temperature of the solar cell T_{p1} °C	68.6
Average radiator wall temperature T_w, °C	68.5
Average coolant temperature T_f, °C	50

Table 3 (continued) – Calculation of characteristics of the heat exchange unit

The temperature of the rear and side surfaces of the heat exchange block is taken equal to the average temperature of the coolant, °C	50
The degree of emissivity of the solar cell surface ε_{ph}	0.95
The degree of emissivity of the surface of the heat exchange block ε_p	0.9
The degree of emissivity of the back and side surfaces of the heat exchange block is taken equal to ε_{pb}	0.8
Convection loss coefficient h_w, W/(m²·K)	24.7
Convection losses Q_w, W	15
Radiation losses Q_c, W	5.1
Total thermal energy loss Q_a, W	20.1
Thermal energy that must be transferred to the coolant Q_f, W	2369
Heat transfer coefficient from the radiator wall to the liquid, which is necessary for heat transfer Q_f to the coolant h_f, W/(m²·K) (calculation using formula (6))	9420
Water temperature at the inlet to the heat exchanger T_{fi}, °C	48.1
Water temperature at the outlet of the heat exchanger T_{fo}, °C	51.9
Heating of water after passing through the heat exchanger, $\Delta T = T_{fo} - T_{fi}$, °C	3.8
Heat capacity of water C_{pw} at 50 °C, J/(kg·K)	4174
Mass flow m', kg/s	0.146
Mass flow д m', l/min	8.6
Flow velocity w, m/s	2.06
Reynolds number Re	2675
Transitional model K_0	5.75
Nusselt number Nu	10.65
Heat transfer coefficient from the radiator wall to the liquid, calculated based on the flow mode h_f, W/(m²·K) (formula (10))	9556

The main energy characteristics of the proposed thermo-photovoltaic unit under conditions of 400 times the concentration of solar radiation [16] are given in Table 4. The characteristics of the thermal part of a photovoltaic installation make it possible to estimate the heating of the coolant per day of operation of the installation. In a regime where the daily maximum solar radiation is 1000 W/m², the temperature of 200 liters of water can be increased per day from the initial temperature by 42 – 3 °C.

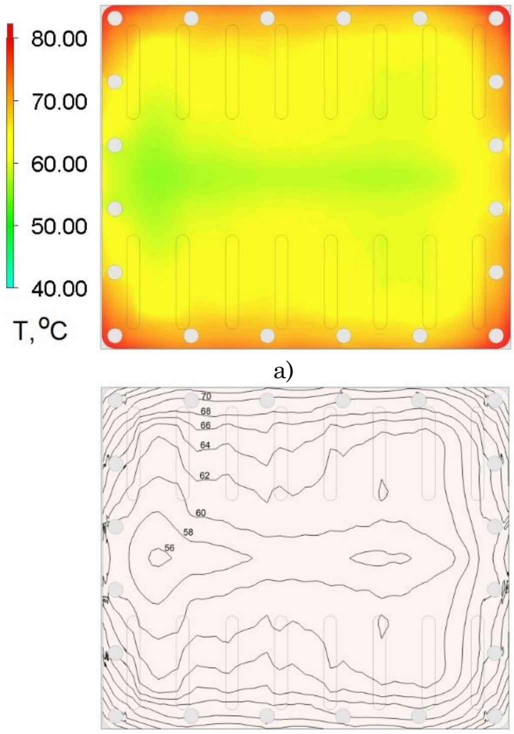
To determine the uniformity of cooling of the surface of the heat exchange block and to study possible variations in the speed and temperature of the coolant flow, computer modeling was carried out. The results of model calculations are shown in Table 5, and in Figure 2 shows a typical picture of the expected temperature distribution on the surface of the solar cell. The input data for the temperature of water entering the heat exchanger simulation was: expected maximum thermal power Q_T, temperature of water entering the heat exchanger T_{fi} and mass flow m' (Table 3).

Table 4 – Energy characteristics of photovoltaic installation

Parameter	**Values**
Number of solar cells/and their efficiency, %	2 / 30
Expected maximum electrical power Q_E, W	653
Expected maximum thermal output Q_T, W	2369
Gross maximum net power $Q = Q_E + Q_T$, W	3022
Overall efficiency $\eta = Q/(Q_s S_a)$, %	87
Percentage of electrical energy	21.6
Percentage of thermal energy	78.4

Table 5 – Results of model calculations of the SC surface temperature

Flow rate, l/min (kg/s)	Water temperature, °C: Input	Water temperature, °C: Output	SC surface temperature, °C: Channel area	SC surface temperature, °C: Main area	SC surface temperature, °C: Edge
8.6 (0.146)	20	23.62	28-32	37	42
	48	51.62	50-60	60-66	71
7.5 (0.125)	48	52.27	~ 52	62	75
5 (0.083)	20	26.89	~ 37	48	52
	48	54.45	~ 65	72	80



a)

b)

Fig. 2 – Results of modeling the temperature distribution over the area of the heat exchange block at a mass flow of 0.146 kg/s (8.6 l/min) and an inlet coolant temperature of 48 °C

As the modeling results showed (Fig. 2), the temperature distribution over the area of the heat exchange block is quite uniform, the areas of the input distribution channel and the output manifold are cooled most effectively. In this case, between the area of the distribution channel and the edges of the area occupied by the solar cell, a temperature difference of ~ 12 °C is observed (such a temperature difference can give a difference in short circuit current of about 9 %), which increases with decreasing flow speed. In addition, as the flow rate decreases, the temperature at the edges of the heat exchange block also increases. On the other hand, in the morning hours at the beginning of system operation, with low coolant temperatures and lower instantaneous solar radiation power, lower flow rates can be used to save energy on pump operation. Simulation also shows that to reduce the temperature difference across the solar cell area, the design of the heat exchange block can be optimized by moving the coolant inlet to the center and creating two outlets on opposite edges of the block. In this case, it is possible to improve the cooling efficiency and reduce the temperature difference between the area of the supply channel and the rest of the solar cell surface by approximately two times.

4. CONCLUSIONS

The proposed design of the radiator of the heat exchange unit provides a transient mode of coolant flow with a heat exchange coefficient between the coolant and the upper plate of the radiator h_f ~ 10000 W/(m^2·K) at the flow rate in the spaces between the plates ~2.1 m/s. This allows solar cells to be effectively cooled in conditions of concentrated solar radiation. Under AM1 lighting conditions and at 400 times the concentration of solar radiation, even at an inlet coolant temperature of 48 °C, the temperature of gallium arsenide solar cells remains acceptable for their efficient operation at coolant flow 8.6 l/min. Modeling showed that modernizing the design of the heat exchange block by moving the coolant inlet to the center and creating two outlets on opposite edges of the block will further improve the uniformity of cooling of the solar cell surface.

Estimates have been made of the expected installed energy characteristics of a heat-electric power plant, which under AM1 conditions and 400 times the concentration of solar radiation for solar cells with an efficiency of 30% have the following values: electrical power 650 W, thermal power 2370 W, total installed net power 3020 W, overall efficiency 87%.

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Аналіз теплових процесів в теплообмінній установці комбінованої фотоелектричної установки з концентрацією сонячної радіації

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Дана стаття присвячена вдосконаленню проектного рішення комбінованих термофотоелектричних систем, що працюють в умовах концентрованого сонячного випромінювання. Основною проблемою є забезпечення ефективного відведення тепла від фотоприймальної поверхні таких систем і передача цієї теплової енергії рідкому теплоносії. У разі використання концентрованого сонячного випромінювання з високою щільністю енергії на фотоприймальній поверхні проблема інтенсифікації процесів теплообміну є критичною для забезпечення ефективної та довгострокової роботи комбінованих термофотоелектричних систем. На основі аналізу теплових процесів запропоновано конструкцію теплообмінної установки комбінованої фотоелектричної установки. Основною конструктивною особливістю запропонованого рішення є мікроканали, виконані на задній пластині фотоприймальної поверхні. Показано, що така конструкція створює перехідний режим потоку теплоносія, що дозволяє ефективно охолоджувати сонячні батареї в умовах концентрованого сонячного випромінювання та потенційно дозволить отримати рекордні значення коефіцієнта теплообміну між теплоносієм і верхньою пластиною радіатора. Використано методику аналізу з використанням критеріальних гідродинамічних рівнянь та чисельного моделювання; результати, отримані в обох випадках, добре збігаються. Оцінено ефективність, теплові та електричні характеристики комбінованої фотоелектричної системи з концентрацією сонячного випромінювання. Зроблено попередні оцінки питомих параметрів встановлених енергетичних характеристик ТЕЦ, які за умов AM1 та 400-кратної концентрації сонячного випромінювання для сонячних елементів з ККД 30 % мають такі значення: електрична потужність 650 Вт, теплова потужність 2370 Вт, загальна встановлена корисна потужність 3020 Вт, загальна ефективність 87 %, що значно перевищує параметри, досягнуті для існуючих сонячних електростанцій.

Ключові слова: PV/T система, Тепловий процес, Теплообмін, Сонячна енергія, Концентроване випромінювання, Охолодження.