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Improving the Solar Collector Base Model for PVT System

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The paper proposes a framework of a general model of the thermal exchange of the solar collector and reviews the most important parameters for the heat-transfer processes such as the coefficient of thermal convection of water, the water flow rate and water flow power which are mainly determined by the system parameters and fluid flow rates and also based on experimental studies of the systems. In the model under consideration, it is proposed to take into account the turbulence of the fluid flow, as well as to choose the most effective mode of fluid motion, and, consequently, the fluid flow rate for heat transfer. Methods for increasing the amount of heat, which is removed from the surface of the solar collector, coming to it, at the expense of the total solar radiation, as well as methods of optimization for different temperature gradients are briefly presented in this article. An analysis of the results shows that methods for increasing the efficiency of heat removal from the surface, which corresponds to the achievement of the maximum power withdrawn, strongly depend on the fluid flow rate and temperature gradient. The optimal values of the temperature gradient are identified based on analytics of the fluid flow rate and magnitude of power withdrawn.

Keywords: Solar collector, PVT system, Power withdrawn, Water flow rate, Temperature gradient, Heat transfer.

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1. INTRODUCTION

The generation of thermal energy from solar is of great interest among researchers. The electrical efficiency of solar cells is low, so most of the incident sunlight is lost in the form of heat. The temperature of the solar cell increases due to this heat, which leads to a decrease of η of the photovoltaic system [1]. Therefore, the removal of thermal energy arising and associated with the solar cell is necessary to achieve maximum efficiency. Creation of combined photovoltaic-thermal system (PVT), which converts incident radiation into electricity and heat at the same time, is gaining popularity [2, 3]. The system can be significantly improved by cooling the PV surface with water as a coolant [4]:

$$\eta = \eta_0\left[1-\beta_0(T_c - T_0)\right], \tag{1}$$

where η_0 is the solar efficiency at standard test conditions (STC), β_0 is the temperature coefficient of efficiency (1/K), T_c and T_0 are the solar cell and coolant temperatures (K), respectively. The coolant in the PVT system is then used to heat the water in the secondary circuit and its further use in domestic needs [5]. Modern humanity is accustomed to the presence of hot water in home and PVT cooling system satisfies the need for electricity and hot water simultaneously.

However, the lack of a good consistent model is an obstacle to the widespread optimization of such systems. And the creation of such a generalized model for taking into account all the necessary parameters is an urgent task to this day.

For example, earlier in [6] efforts were made to create a model for adsorption cooling: innovative adsorbing materials and heat pipes were applied, the specific heat removal capacity was improved, and an efficiency equal to 0.39 was obtained. The numerical modeling and optimization carried out in [7] allowed us to obtain typical values for the efficiency of about 0.15 that is not consistent with the experimental results. Also it was conducted experimental studies of two-stage systems [8], various cycles (e.g. continuous heat recovery, mass recovery, heat wave, cascade effect) [9, 10], the calculated value for the efficiency of such systems is close to 0.5.

Further research, like the previous ones, considered the disparate experimental results and described them as particular physical models that do not work at all [11-13]. Therefore, the aim of this work was to propose the basis of a general model of heat exchange of the solar collector, which allows us to take into account all the necessary parameters, including parameters for the development of PVT systems.

2. MODEL BASIS

The Navier-Stokes equations are the basis for the processes under consideration and describe the fluid flow for various variations and systems. Newton's second law describes the change in the velocity of an object when a force is applied. The Navier-Stokes equations are equivalent to Newton's law, when many objects, such as microscopic particles in air or water, are considered as a group, that is, independently of each other. Ocean currents, smoke movement, dynamics of hurricanes or weather conditions, air/liquid flow when a detonating device is triggered, heat removal by an air/liquid flow from a heated object, and many other interesting phenomena are all examples of situations where the Navier-Stokes equations can be applied. We write the energy equations and the Navier-Stokes equations in a general form for the system under consideration [14, 15]:

$$\begin{cases} \dfrac{\partial(\rho c_p T)}{\partial t} + \dfrac{\partial(\rho u c_p T)}{\partial x} + \dfrac{\partial(\rho v c_p T)}{\partial y} + \dfrac{\partial(\rho w c_p T)}{\partial z} = \nabla(k\nabla T) \\ \rho\dfrac{\partial u}{\partial t} + \rho u\dfrac{\partial u}{\partial x} + \rho v\dfrac{\partial u}{\partial y} + \rho w\dfrac{\partial u}{\partial z} = \nabla(\mu\nabla u) - \dfrac{\partial P}{\partial x} \\ \rho\dfrac{\partial v}{\partial t} + \rho u\dfrac{\partial v}{\partial x} + \rho v\dfrac{\partial v}{\partial y} + \rho w\dfrac{\partial v}{\partial z} = \nabla(\mu\nabla v) - \dfrac{\partial P}{\partial y} \\ \rho\dfrac{\partial w}{\partial t} + \rho u\dfrac{\partial w}{\partial x} + \rho v\dfrac{\partial w}{\partial y} + \rho w\dfrac{\partial w}{\partial z} = \nabla(\mu\nabla w) - \dfrac{\partial P}{\partial z} - \rho g \end{cases}, \tag{2}$$

where ρ is the liquid density (kg/m³); c_p is the specific heat (J/(kg·K)); T is the temperature (K); u, v, w are the velocities (m/s); x, y, z are the coordinates (m), k is the coefficient of thermal conductivity (J/(m·K)), P is the pressure (Pa).

Important parameters for the heat-transfer processes are thermal convection coefficient of water h_w and power of water flow N_{flow}, which are mainly determined by the system parameters and fluid flow rate. Calculation formulas for h_w and N_{flow} are the following [16]:

$$\begin{aligned} h_w &= k_w Nu / D_h \\ N_{flow} &= f D_h l \frac{u_w^2}{2} \rho_w u_w n \end{aligned}, \tag{3}$$

where $k_w = 0.599$ W/(m·K) is the thermal conductivity of water; Nu is the Nusselt number; $D_h = 10^{-2}$ m is the pipe diameter (m); f is the viscous friction coefficient; $l = 2$ m is the pipe length; $\rho_w = 998.2$ kg/m³ is the water density; u_w is the water flow rate (m/s); $n = 8$ is the number of pipes in the system.

Since in the system under consideration the inner diameter of the pipe $D_h \geq 10^{-3}$ m, the dimensions of the system can be considered as macroscopic channels, the values of Nu and f can be found using the classical relations of convective heat transfer in a pipe [17, 18].

$$\begin{cases} \begin{cases} \text{Laminar flow, Re} \leq 2300: Nu = 4.364, \\ f = 64/\text{Re} \end{cases} \\ \begin{cases} \text{Transient flow, } 2300 \leq \text{Re} \leq 10000, \\ \text{Turbulent flow, Re} \geq 10000: \\ Nu = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.4},\ f = \left[1.14 + 2\log(D_h / R_q)\right]^{-2} \end{cases} \end{cases}, \tag{4}$$

where $\text{Re} = u_w D_h / \mu_w$ is the Reynolds number, $\text{Pr} = \mu_w / \alpha_w \approx 7$ is the Prandtl number, $\mu_w = 1.006 \cdot 10^{-6}$ m²/s is the kinematic viscosity of water, $\alpha_w = 0.151 \cdot 10^{-6}$ m²/s is the thermal diffusivity of water [19, 20], $R_q = 3 \cdot 10^{-6}$ m is the rms surface roughness coefficient.

At low liquid flow rates in the pipe (Reynolds number $\text{Re} \leq 2300$) the fluid flow is laminar, and at high velocities ($\text{Re} \geq 10000$) it is turbulent. In the range of $2300 \leq \text{Re} \leq 10000$ there is a transient flow from laminar to turbulent [18]. In all cases, even when the flow in the pipe is turbulent, the flow in the narrow wall layer is laminar.

In the considered model, it is proposed to take into account that the strong turbulence of the entire flow, which occurs even in the transition mode, provides good heat transfer in the near-wall zone, as well as the absence of deposits on the wall. Powerful vortices destroy the wall laminar layer. They are formed when the flow is twisted in twisted pipes and when the flow is cut off from protrusions, depressions, ribs, various static defects and irregularities placed on smooth pipes. The same vortices prevent the formation of deposits. The flow is constantly strongly turbulized in the entire volume, both in the central zone and in the near-wall layer, i.e. we can consider the problem in the turbulent flow approximation.

The coefficient of viscous friction for systems of transitional and turbulent types is $f = 0.015$.

Subsequently, to determine the amount of heat that is removed from the surface, the Newton-Richmann law is used: the amount of heat given off by a unit surface of a body per unit time

$$Q = h_w S_{SC} (T_c - T_0), \tag{5}$$

where Q is the amount of heat removed (W). It is necessary to determine the effective area of the solar collector S_{SC}: let take it as part of the segment area (Θ is the angle on which a segment rests) of a cylinder lateral multiplied by $n = 8$ (number of pipes in the system):

$$S_{SC} = (\pi \frac{D_h \cdot \theta}{180} + D_h \cdot \sin(\theta)) l \cdot n \approx 0.1 m^2. \tag{6}$$

Table 1 – Reynolds and Nusselt numbers depending on the water flow rate for pipes with an inner diameter $D_h = 10^{-2}$ m

u_w, m/s	Re	Nu	Flow type
0.1	994.04	4.364	Laminar flow
0.2	1988.07	4.364	Laminar flow
0.25	2485.09	26.06	Transient flow
0.3	2982.11	30.16	Transient flow
0.4	3976.14	37.96	Transient flow
0.5	4970.18	45.38	Transient flow
0.6	5964.21	52.5	Transient flow
0.7	6958.25	59.4	Transient flow
0.8	7952.29	66.09	Transient flow
0.9	8946.32	72.62	Transient flow
1	9940.36	79.01	Transient flow
1.1	10934.39	85.27	Turbulent flow

Heating of the coolant in the coolant circuit in the solar collector is due to the useful energy of the power Q_{eff} withdrawn from the solar collector at a given time. Energy Q_{eff} represents the difference between the amount of solar energy absorbed by the collector plate and the heat loss to the environment. Equation (7) applicable to the calculation of almost all existing designs of flat plate solar collectors (FPSC) [11, 12] has the form:

$$Q_{eff} = F_R S_C \left[I_T (\tau\alpha) - U_L (T_{c_{max}} - T_0) \right], \tag{7}$$

where Q_{eff} is the useful energy withdrawn from the collector per unit time (W); F_R is the collector heat removal coefficient; $S_C = 2$ m² is the total area of the solar collector (m²), $I_T = 1000$ W/m² is the total solar radiation flux density in the plane of the collector; τ is the transmittance of the solar collector transparent coating in relation to solar radiation; α is the absorption capacity of the collector plate relative to the solar radiation; U_L is the total loss coefficient of the thermal collector (W/(m·K)).

3. MODEL TESTING

The coefficients $F_R(\tau\alpha)$ and U_L, depending on the design of FPSC, fluid flow through the collector and their size, are determined by standard tests of the collectors [21], in which the total solar radiation flux falls vertically onto the surface of the solar collector. For the system under consideration, the reservoir parameters are equal:

$$F_R(\tau\alpha) = 0.78; \quad F_R U_L = 7.62 \frac{W}{m^2 \cdot K}. \tag{8}$$

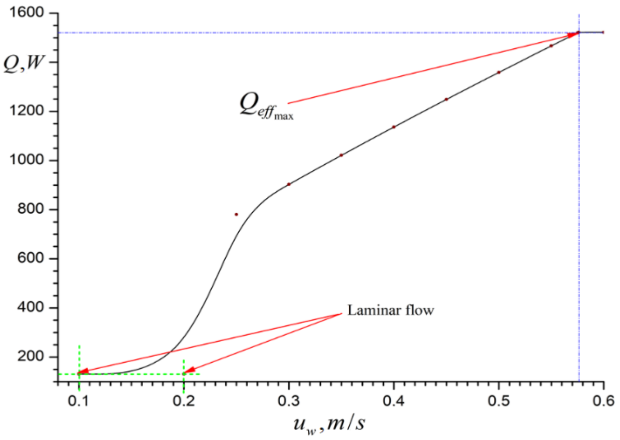


Fig. 1 – The dependence of the amount of heat Q, which is diverted from the system when the coolant flows through it at $T = 293$ K, by water flow rate u_w

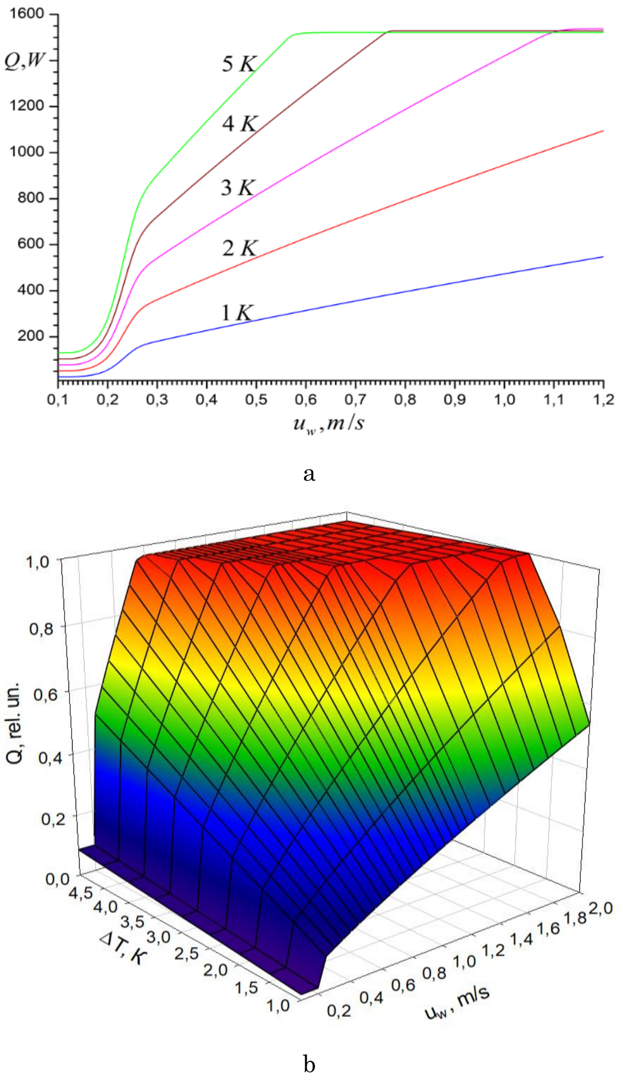


Fig. 2 – The dependence of the amount of heat Q, which is diverted from the system when the coolant flows through it at $T = 293$ K, by water flow rate u_w for different temperature gradients $\Delta T = 1,2,3,4,5$ K

That is, the boundary value of the effective energy of the system under consideration cannot exceed the value $Q_{eff} \leq 1483.8$ W at $T_{cmax} - T_0 = 5$ K (Fig. 1).

For values of the water flow rate u_w up to 0.2 m/s the regime of fluid motion is laminar and the amount of heat given off to the fluid is too small, and the coolant passes through the pipes practically without heating. With an increase in the water flow rate u_w, a sharp jump in the amount of heat of the transmitted liquid and, consequently, its heating are observed in the future. At the water flow rate u_w of the order of 0.6 m/s, the amount of heat, which the water takes off, reaches the maximum value, i.e., the value of the effective energy that the system under consideration can provide with the given parameters for heating the water flow by 5 K in one pass through the pipes of this solar collector (Fig. 2).

Thus, we can conclude that the obtained physical model $Q(u_w, \Delta T)$ is applicable to classical solutions of thermal collectors in a general form. However, under real operating conditions, depending on the various design features of the collector, the determination of optimal parameters should be carried out taking into account the design parameters.

To test the model, laboratory studies of the collector were carried out, a fragment of which is shown in Fig. 3a according to its classical connection diagram (Fig. 3b) [22].

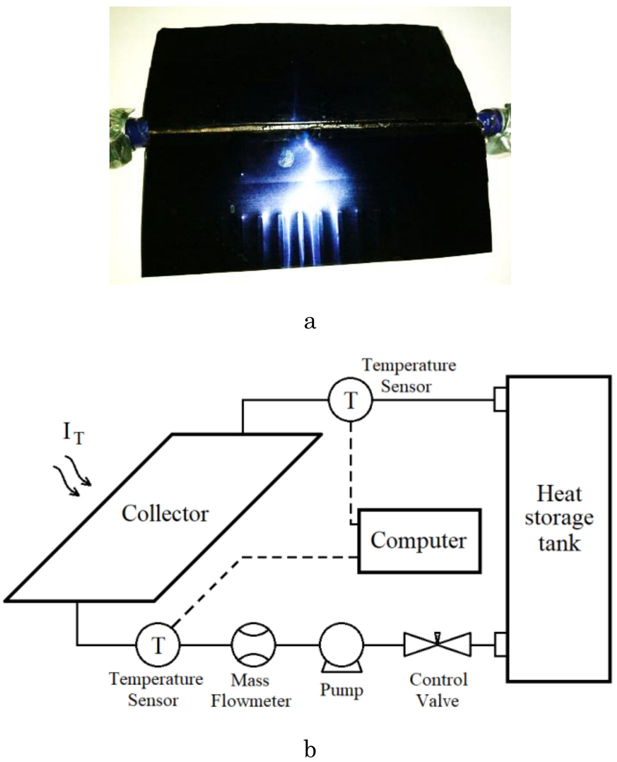
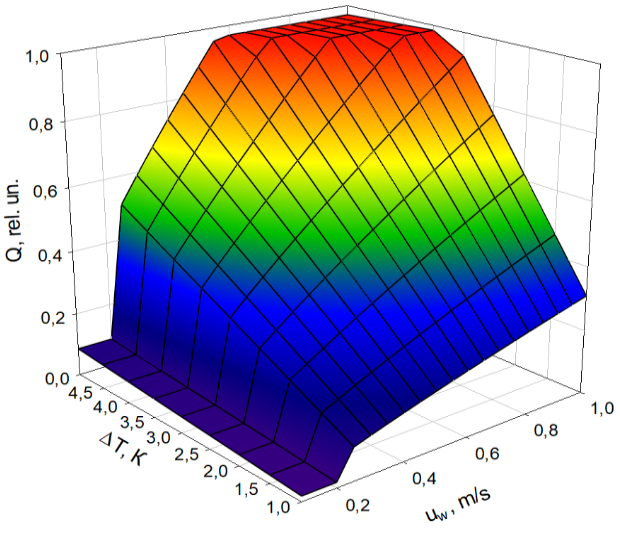


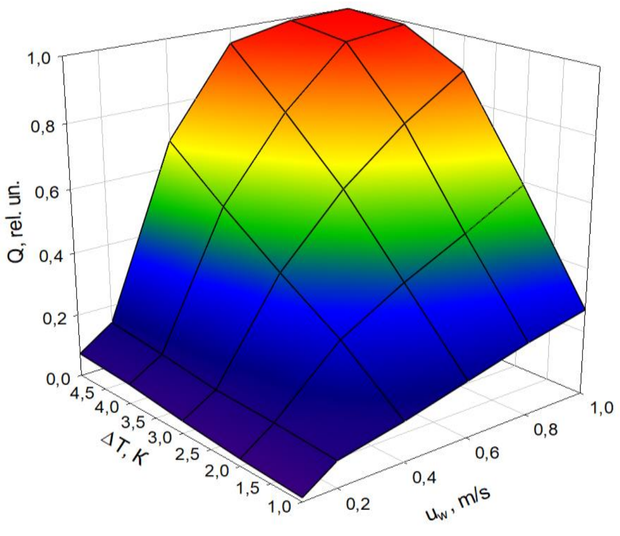
Fig. 3 – A fragment of the solar collector for testing the model (a) and a scheme of the measuring stand (b)

Studies have allowed a comparison of simulation results with the field values. The temperature difference corresponding to the simulation was achieved by forcibly changing the light intensity. The experimental dependence $Q(u_w, \Delta T)$ is shown in Fig. 4b for comparison with theoretical values on the same scale (Fig. 4a).

From the above dependences, one can see a fairly exact coincidence of the results of laboratory tests with the proposed model. During the tests, a characteristic small power take-off is also observed with a laminar flow at coolant flow rates up to 0.2 m/s, as well as reaching the maximum value at coolant velocities of more than 0.8 m/s and temperature gradient of more than 3-4 K, characteristic limit power withdrawn. The aforementioned confirms the correspondence of the model to the physical mechanism of the collector's operation and will allow further modeling of real systems, in particular PVT systems [2], where there are additional thermal interfaces influencing the heat flows.



a



b

Fig. 4 – Comparison of the theoretical (a) and experimental (b) dependences of the amount of power taken by the collector on the flow rate of the coolant and the temperature gradient between the input and output

In practice, for a real system with a constant intensity of the light flux, the temperature gradient is directly related to the flow rate of the coolant through the design parameters of the collector. For example, the analytical dependence $\Delta T(u_w)$ or the laboratory test performed in the work has the following form:

$$\Delta T = \begin{cases} 4,3 \quad at \quad u_w < 0,2 \\ 5,31 \cdot e^{-1,05 u_w} \quad at \quad u_w \geq 0,2 \end{cases} \tag{9}$$

Using the given dependence will allow to replace variables in the proposed model and obtain a dependence with one variable, which will be characteristic of a particular reservoir design. Subsequently, when using this model, the task of optimizing the reservoir parameters and the task of finding the optimal flow rate of the coolant, including an algorithm for creating intelligent control coolant velocity, are greatly simplified.

4. CONCLUSIONS

To date, there is no comprehensive mathematical model for predicting the thermal and electrical efficiency of PVT systems using liquid cooling technology. This study developed a comprehensive physical model based on the well-known solar collector model, taking into account additional parameters. Through a theoretical analysis of calculations based on widely known data [20], additional influence factors, such as the parameters of the system under study, temperature gradient, and fluid flow rate, were taken into account. For this, the basic equations of hydrodynamics and heat transfer were used.

In the model under consideration, in the general case, differential continuity equations (Navier-Stokes equations), equations of motion and energy conservation are used in the material medium to describe the processes of heat removal and heat transfer that are carried out during forced circulation at the surfaces of the coolant exposed to a greater number of factors in the material medium. Additionally, during modeling, the system parameters were introduced into the calculation, which were used to determine the amount of heat removed from the surface using the Newton-Richmann law, as well as to determine the boundary value of the heat removed from the collector surface for various temperature gradients and fluid flow rates.

Laboratory testing of the proposed model allowed us to confirm its applicability to classical solutions of solar collectors, and the correct determination of the system parameters allows us to achieve the necessary accuracy of calculations. Subsequently, this model will allow us to move on to the study and theoretical justification of more complex practical models – PVT systems, as well as to develop intelligent algorithms for controlling the coolant speed taking into account operating conditions.

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Удосконалення базової моделі сонячного колектора для PVT системи

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У статті запропонована основа загальної моделі теплового обміну сонячного колектора і розглянуто найбільш важливі параметри для процесів тепловіддачі, такі як коефіцієнт теплової конвекції води, швидкість і потужності потоку води, які в основному визначаються параметрами системи та швидкістю потоку рідини, а також ґрунтуючись на експериментальних дослідженнях даних систем. У запропонованій моделі пропонується враховувати турбулізацію потоку рідини, а також вибрати найбільш ефективний режим руху рідини, а, отже, і швидкість потоку рідини для перенесення тепла. У статті коротко представлені методи збільшення кількості теплоти, що відводиться від поверхні сонячного колектора, за рахунок потоку сумарної сонячної радіації, а також методи оптимізації для різних градієнтів температур. Аналіз отриманих результатів показує, що методи збільшення ефективності відведення тепла від поверхні, яка відповідає досягненню максимальної відібраної потужності, сильно залежать від швидкості потоку рідини та градієнта температур. Визначено найбільш оптимальні значення градієнта температур на основі аналітики швидкості потоку рідини і величини відібраної потужності.

Ключові слова: Сонячний колектор, PVT система, Потужність, Швидкість потоку води, Градієнт температур, Теплообмін.