Results of CFD-simulation of a solar photovoltaic-thermal module

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Abstract. Currently, in the energy supply of agroindustrial complex facilities there is an increasing interest in the development of structures and engineering systems using renewable energy sources, including combined photovoltaic-thermal modules (PVT-modules), combining photovoltaic modules (PV-modules) and solar collectors in one design. The use of PVT-modules technology makes it possible to increase the electrical performance of solar cells due to their cooling during operation, while the use of PVT-modules integrated into the roofs and facades of buildings can significantly reduce the need for centralized electricity and heat supply. The purpose of this work is to develop a CFD model of a solar PVT module for producing an estimate of the thermal performance of a PVT module. Using the ANSYS Fluent software package, a CFD model of the PVT module was developed. The contours of the temperature distribution, the coolant pressure in the PVT module channel at different values of the coolant flow at the entrance to the PVT module were obtained. The paper presents the daily variation of thermal power and total insolation calculated using the solar calculator built into ANSYS. The verification of the developed CFD-model of the PVT-module showed the comparability of the calculated values with the data obtained analytically and experimentally - the relative error of the results was no more than ± 1%. ANSYS is an effective software package that allows simulating thermal processes in PVT-modules.

1. Introduction

Converting solar energy into electricity and heat with a single device (PVT-module) is currently a promising area of research and development. PVT-modules integrated into the roofs and facades of buildings and operated as part of a common power system can cover a significant part of the electricity and heat consumption of buildings.

In [1], a new design of a PVT-system is presented, in which solar cells are glued to the bottom of a glass cover in order to realize higher power generation taking into account a lower temperature of the glass cover compared to the absorber temperature. A numerical analysis model is developed for comparison of the characteristics of the traditional and new PVT-system. Article [2] presents the development of an integrated photovoltaic/thermal system (BIPV/T) in a building. Four numerical models: a three-dimensional dynamic model and three stationary models are 3D, 2D and 1D built in [3]. Based on control-volume finite-difference approach, an explicit dynamic model for a flat single-glazed PVT module was developed in [4]. The articles [5, 6] provide an overview of research and development on the photovoltaic/thermal (PVT) technology. The results of modeling a hybrid PVT-
module based on CdTe are presented in [7]. CFD-analysis solar air heater using ANSYS Fluent software was performed in [8, 9].

The goal of this article is to perform numerical modeling of thermal processes occurring in a solar PVT module. To achieve this goal, the presented article solves the following tasks: to develop a CFD model of the PVT module; to obtain the contours of the distribution of temperature, coolant pressure in the PVT-module channel at different values of the coolant flow at the entrance to the PVT-module; to verify the developed CFD model of the PVT-module based on analytical and experimental data.

2. Materials and methods
The work [10] presents the design of the developed solar PVT-module containing a protective glass coating, connected solar cells (SC), placed between the glass and the body with a coolant channel having a rectangular cross-section. As an electrical insulating material, a polysiloxane two-component organosilicon optically transparent low-modulus gel with a thermal conductivity coefficient of 0.2 W/(m·K) was used, the layer thickness is 0.1 ... 2 mm [11]. The absorber tank (coolant channel), in which the coolant is heated, is made of steel. The walls of the coolant channel are steel sheets 1.5 mm thick over the entire area. Two sheets of transparent honeycomb polycarbonate, 4 mm thick each, are used as insulating material. Figure 1 shows a cross-sectional diagram of a PVT-module with an air glass unit, indicating the thicknesses of all layers.

![Figure 1. A cross-section scheme of the PVT-module: 1 – single-chamber double-glazed window with an air gap of 8 mm (glass thickness 4 mm), 2 – layer of polysiloxane gel, 3 – solar cells, 4 – coolant channel wall (steel 1.5 mm thick), 5 – coolant (water, layer 7 mm thick), 6 – honeycomb polycarbonate sheets 4 mm thick](image)

From the side of the photo-receiver, the heat flux passes through the layer of polysiloxane gel and the steel channel of the module is equal to:

\[
q_{sc} = \frac{T_{sc} - T_t}{\delta_g \lambda_g + \delta_t \lambda_t} \tag{1}
\]
where $T_{sc}$, $T_{tank}$ – temperature of the solar cells and the wall of the coolant channel in this section, °K; $\lambda_g$ – thermal conductivity coefficient of the polysiloxane gel layer, W/m·°K; $\lambda_r$– the coefficient of thermal conductivity of steel, W/m·°K.

The same amount of heat is transferred to the coolant from the channel surface:

$$q_{sc} = \alpha_f (T_t - T_f),$$

(2)

where $T_f$ – fluid (water) temperature, °C; $\alpha_f$ – fluid heat transfer coefficient, W/m²·°K.

Coolant temperature will change due to heat flow:

$$q_f = q_{sc} - c_p\rho\n(T_{in} - T_{out}),$$

(3)

where $c_p$ – heat capacity of the coolant, J/kg·°K; $\rho$ – fluid density, kg/m³; $\n$ – coolant speed, m/s; $T_{in}$– inlet coolant temperature, °K; $T_{out}$– outlet coolant temperature, °K.

Through the cover glass to the ambient air from the surface of the photo-receiver the heat flux is transmitted:

$$q_{sc} = \frac{T_{sc} - T_{gl}}{\delta_g + R_{gl}},$$

(4)

where $T_{gl}$ – glass temperature, °K; $R_{gl}$ – thermal resistance of the glass unit, m²·K/W.

In turn, the same flow spreads from glass to the environment:

$$q_{sc} = \frac{T_{gl} - T_{am}}{h_k + h_r},$$

(5)

where $T_{am}$ – ambient temperature, °K; $h_k$, $h_r$ – respectively, the heat transfer coefficients to the environment by convection and radiation, W/m²·°K.

Next, using the method of successive approximations, we found the temperature of the solar cells and adjusted the flow distribution. After the coolant temperature was determined. The calculation method is as follows: the density of the heat flux on the surface of the solar cells is determined, in the first approximation, the symmetric propagation of the heat flux is assumed, heat balance equations are drawn up on both sides of the solar cells. Further, the temperature of solar cells is found by the method of successive approximations and the flux distribution is corrected. The temperature of the absorber and the heat carrier is determined using formulas (1)-(2).

To verify the results of calculations using ANSYS Fluent software package, we developed a mathematical model of the PVT-module, which is the Navier-Stokes equation (viscous incompressible fluid):

$$\rho \frac{\partial w_x}{\partial \tau} + \rho \left( w_x \frac{\partial w_x}{\partial x} + w_y \frac{\partial w_x}{\partial y} + w_z \frac{\partial w_x}{\partial z} \right) = \rho g_x - \frac{\partial p}{\partial x} + \mu \left( \frac{\partial^2 w_x}{\partial x^2} + \frac{\partial^2 w_x}{\partial y^2} + \frac{\partial^2 w_x}{\partial z^2} \right),$$

$$\rho \frac{\partial w_y}{\partial \tau} + \rho \left( w_x \frac{\partial w_y}{\partial x} + w_y \frac{\partial w_y}{\partial y} + w_z \frac{\partial w_y}{\partial z} \right) = \rho g_y - \frac{\partial p}{\partial y} + \mu \left( \frac{\partial^2 w_y}{\partial x^2} + \frac{\partial^2 w_y}{\partial y^2} + \frac{\partial^2 w_y}{\partial z^2} \right),$$

$$\rho \frac{\partial w_z}{\partial \tau} + \rho \left( w_x \frac{\partial w_z}{\partial x} + w_y \frac{\partial w_z}{\partial y} + w_z \frac{\partial w_z}{\partial z} \right) = \rho g_z - \frac{\partial p}{\partial z} + \mu \left( \frac{\partial^2 w_z}{\partial x^2} + \frac{\partial^2 w_z}{\partial y^2} + \frac{\partial^2 w_z}{\partial z^2} \right),$$

(6)
where \( \rho g, \partial dx, dy, dz \) - gravity in projection on the x-axis applied to a selected elementary volume of fluid with rib dimensions \( dx, dy, dz \), N; \( g \) - acceleration of gravity, m/s\(^2\); \( -\frac{\partial p}{\partial x} dx dy dz \) - pressure force acting on an element in the opposite direction to the movement of the fluid, N; \( \mu \left( \frac{\partial^2 w_x}{\partial x^2} + \frac{\partial^2 w_x}{\partial y^2} + \frac{\partial^2 w_x}{\partial z^2} \right) dx dy dz \) - friction force, N; \( \mu \) - dynamic viscosity coefficient, Pa·s; \( w_x \) - fluid velocity projection on the x-axis, m/s.

Combining equations (6) into one vector equation, we obtain the Navier-Stokes equation in vector form:

\[
\rho \frac{D\vec{w}}{Dt} = \rho g - \nabla p + \mu \nabla^2 \vec{w},
\]  

(7)

where \( \nabla p \) - Hamilton operator for pressure force; \( \nabla^2 \vec{w} \) - Laplace operator of the velocity vector.

The temperature boundary conditions:

\[
T_{am} = 302^\circ K; \quad T_{in} = 298^\circ K.
\]  

(8)

Differential equations (6) are solved by the finite volume method on a three-dimensional mesh - the original geometry, divided into finite volumes - geometric primitives (hexagons, prisms, tetrahedrons). The total number of grid elements was 32576 final volumes.

3. Results

During the simulation, the insolation value was calculated using the built-in ANSYS solar calculator: clear windless day (sunshine factor is assumed to be 1, wind speed 0 m/s), Moscow, date 27.07, time 13.00, total insolation \( E = 874.86 \text{ W/m}^2 \). The coolant flow was taken equal to \( l = 3.65 \ldots 40.15 \text{ l/h} \); water was used as the coolant. The number of iterations was taken equal to 1000, while the convergence of the solution under various conditions was achieved in the range from 500 to 600 iterations. Figure 2 shows the temperature contour of the coolant along the PVT-module channel at the coolant flow \( l = 14.6 \text{ l/h} \).

Figure 2. The temperature contour of the coolant along the PVT-module channel at a coolant flow of \( l = 14.6 \text{ l/h} \), total insolation – \( E = 874.86 \text{ W/m}^2 \), ambient temperature – \( T_{am} = 302 ^\circ K \), and temperature of water at the inlet of PVT-module – \( T_{in} = 298 ^\circ K \)

Figure 3 shows the pressure contour of the coolant through the PVT-module channel at the coolant flow of \( l = 14.6 \text{ l/h} \).
Figure 3. The pressure contour of the coolant along the PVT-module channel at a coolant flow of $l=14.6$ l/h, total insolation - $E= 874.86$ W/m$^2$, ambient temperature - $T_{am}=302$ °K, and temperature of water at the inlet of PVT-module - $T_{in}=298$ °K.

For clarity, Figures 4-5 show graphs of changes in the temperature of the coolant and pressure along the entire length of the PVT-module channel.

Figure 4. The temperature distribution diagram of the coolant along the PVT-module channel at a coolant flow of $l=14.6$ l/h, total insolation - $E= 874.86$ W/m$^2$, ambient temperature - $T_{am}=302$ °K, and temperature of water at the inlet of PVT-module - $T_{in}=298$ °K.

As seen from Figure 4, the value of the coolant temperature changes along the channel length from 298 °K at the PVT-module inlet to 303.78 °K at the channel outlet, while in the zone of the coolant flow deceleration (along the edges of the PVT-module channel) regions with an increased temperature of 306 °K are observed.
Figure 5. The pressure distribution diagram of the coolant along the PVT-module channel at a coolant flow of $I=14.6$ l/h, total insolation $- E= 874.86$ W/m$^2$, ambient temperature $- T_{am} = 302$ °K, and temperature of water at the inlet of PVT-module $- T_{in}=298$ °K.

Figure 5 shows that the coolant pressure along the length of the channel varies from 0.085 Pa at the inlet of the PVT-module to 0.005 Pa at the outlet from the PVT-module. The thermal power of the PVT-module is found by the expression (9):

$$P_T = c_p \rho c s (T_{out} - T_{in}),$$  \hspace{1cm} (9)

where $s$ – cross-sectional area of the coolant channel, m$^2$.

Figure 6 shows the diurnal variation of the thermal power of the PVT-module and the total insolation calculated for July 27, Moscow (a clear, windless day was taken).

Figure 6. Diurnal variation of thermal power of PVT-module and total insolation on a clear windless day (July 27, Moscow), temperature of water at the inlet of PVT-module $- T_{in}=298$ °K
4. Discussion
To verify the calculation results obtained using the PVT-module model developed in ANSYS Fluent, a comparison was made of the dependence of the temperature of the coolant at the outlet of the PVT-module on the coolant flow obtained analytically.

The relative error of the calculation results is determined from the expression (10):

\[ \Delta_c = \frac{T_c - T_{ANSYS}}{T_c} \times 100, \% \]

(10)

where \( \Delta_c \) – relative error of the computer model compared to the calculated results, \%; \( T_c \) – calculated value of the coolant temperature at the outlet PVT-module, °K; \( T_{ANSYS} \) – calculated using ANSYS value coolant outlet temperature PVT-module, °K.

The relative error of the results of the computer model in comparison with the experimental data is determined in a similar way:

\[ \Delta_e = \frac{T_e - T_{ANSYS}}{T_e} \times 100, \% \]

(11)

where \( \Delta_e \) – relative error of a computer model compared to experimental results, \%; \( T_e \) – the experimental value of the coolant temperature at the outlet PVT-module, °K.

A selection of experimental data and calculation results are shown in Table 1.

| Coolant flow, l/h | Calculation | Experiment | ANSYS calculation | Relative error |
|-------------------|-------------|------------|------------------|----------------|
|                   | \( T_c, ^\circ K \) | \( T_e, ^\circ K \) | \( T_{ANSYS}, ^\circ K \) | \( \Delta_c, \% \) | \( \Delta_e, \% \) |
| 3.65              | 319.09      | 319.74     | 318.42           | 0.21           | 0.41           |
| 7.30              | 309.29      | 309.48     | 309.20           | 0.03           | 0.09           |
| 10.95             | 305.70      | 306.01     | 305.64           | 0.02           | 0.12           |
| 14.60             | 303.85      | 304.05     | 303.78           | 0.02           | 0.09           |
| 18.25             | 302.71      | 302.94     | 302.65           | 0.02           | 0.10           |
| 21.90             | 301.94      | 302.09     | 301.87           | 0.02           | 0.07           |
| 25.55             | 301.39      | 301.52     | 301.32           | 0.02           | 0.07           |
| 29.20             | 300.97      | 301.12     | 300.89           | 0.03           | 0.07           |
| 32.85             | 300.65      | 300.77     | 300.58           | 0.02           | 0.06           |
| 36.50             | 300.39      | 300.35     | 300.33           | 0.02           | 0.01           |
| 40.15             | 300.17      | 300.16     | 300.13           | 0.01           | 0.01           |

The data in Table 1 show the convergence of the data obtained using computer modeling with the results of the experiment and analytical solution - the relative error is no more than ± 1%.

5. Conclusion
As a result of numerical modeling of the processes occurring in the PVT-module, using the ANSYS Fluent software package with a built-in solar calculator, the contours of the distribution of the coolant temperature and pressure along the channel are obtained. At a total insolation of 874.86 W/m², an ambient temperature of 302 °K, a coolant temperature at the PVT-module inlet of 298 °K, a coolant flow from 3.65 to 40.15 l/h, the coolant temperature at the outlet of the PVT-module changes from 318.42 to 300.13 °K. Verification of the data obtained using the developed computer model showed the comparability of the analytically calculated and experimental values of the coolant temperature at the outlet of the PVT-module.
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