Thermal performance and efficiency of extruded absorber solar collector

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Abstract. Thermal characteristics of a solar collector are presented in the article. A collector with extruded profiles is considered as the solar collector studied model, which allows increasing the efficiency of heat transfer and reducing heat loss to the environment. The calculation showed that with the intensity of solar radiation from 450 W/m² or more, the efficiency of using solar energy of the collector is more than 50%. The heat-storage medium heating degree in it should not be more than 30-35 °C to obtain the maximum heat output of the collector with extruded profiles. The coefficient of heat removal from the collector with a change in the arrival of solar radiation and with a change in the temperature of the heat-storage medium in the collector ranges from 0.854 to 0.877.

1. Introduction
Solar radiation, as an alternative energy source, is available in any part of the Earth's surface and has a variable intensity from 0 to 1000 W/m². In the Volgograd region, the potential of solar energy ranges from 33 kW·h/m² per month in winter and up to 280 kW·h/m² per month in summer, with more than 2000 hours of sunshine per year. This makes it attractive to use solar energy for agricultural needs in solar water heating installations.

Like any other heat exchange device, a solar collector cannot have 100% efficiency, since there are two types of losses during energy conversion: optical and thermal ones. The optical losses depend on the insulation reflectivity, the absorption capacity of the absorber, and the efficiency of the absorbing panel. Optical losses determine the optical efficiency of the solar collector and are expressed by the optical efficiency coefficient (efficiency). Heat losses directly depend on the difference in temperature on the surface of the collector and the ambient air. Thermal insulation and various coatings are used to reduce them.

The efficiency of a solar collector is determined by the overall efficiency, based on its optical and thermal efficiency. At the same time, the optical efficiency is the main determining component. Open-type solar collectors have the highest optical efficiency, since they do not have transparent insulation. But at the same time, collectors of this type also have the greatest heat losses. Vacuum-type collectors have the lowest heat losses. However, they have a small optical efficiency, as its design uses a two-layer transparent insulation and a cylindrical absorber.
The widespread use of solar collectors and their use in everyday life and in industry sets the task of improving the collectors’ design, which would lead to an increase in the overall efficiency of the collector, high heat conductivity, low energy losses.

2. Materials and methods

A collector with extruded profiles representing main liquid pipes, the ends of which are connected to collector pipes and absorbing sheets, was considered as the studied model. The absorbing sheets and all pipes are made of aluminum alloy AD31T1. Each main liquid pipe is made in the form of a profile with side fences along the entire length; the profile height is less than 1/3 of the profile end part width. The profile is divided by partitions inside, and absorbing sheets [1] are laid and welded together along the entire length on the profile side fences. The absorbing sheets and pipes are coated with a selective coating—a hot Cu(NO$_3$)$_2$-KMgO solution. At the bottom, the collector is insulated with 25 mm glass fiber. The upper part of the collector is covered with protective polycarbonate glass 3 mm thick. This design allows increasing the efficiency of heat transfer and reducing heat loss to the ambient medium.

A mathematical model based on [2-5] was developed to determine the efficiency of the collector. The equation of the energy balance of the solar collector is:

$$S_c \left[ IR_{(\tau a)b} + IR_{(\tau a)d} \right] = Q_u + Q_L + Q_s,$$

where $S_c$ is the area of the solar collector, m$^2$; $IR_{(\tau a)b}$ is the intensity of direct and scattered radiation of solar energy on the surface, W/m$^2$; $IR_{(\tau a)d}$ is the intensity of solar energy diffusive radiation on the surface, W/m$^2$; $Q_u$ is the heat flow transmitted in the collector to water, W; $Q_L$ is the heat losses in the collector to the ambient medium, W; $Q_s$ is the heat flow accumulated by the collector, W.

When the collector is stationary, the heat flow accumulated by the collector at the initial time can be ignored [6], i.e. $Q_s = 0$.

To simplify calculations, the concept of total loss coefficient $U_L$, W/(m$^2$·K), is introduced, and the collector itself is considered in the form of the following thermal scheme (Fig. 1).

![Figure 1. Thermal diagram of a solar collector: IR is total solar radiation input; IR$_{R1}$ and IR$_{R1}R2$ are energy loss on reflection; $R_1$, $R_2$, $R_3$, $R_4$ are total thermal resistances of heat transfer and thermal conductivity.](image)

As can be seen from Figure 1, total losses are made up of losses through the lower part of the collector, through the upper part and through side one:

$$U_L = U_b + U_t + U_h$$

where $U_b$ is the coefficient of heat losses through the lower part of the collector;
is the coefficient of heat losses through the upper part of the collector;

is the coefficient of heat losses through the side part of the collector.

The heat losses determining through the side parts of the collector is quite a difficult task, and based on the conditions of the collector efficiency, it is assumed that .

Since the temperatures and practically do not differ, it was assumed that the thermal resistance of heat transfer at the lower surface of the collector is .

Thus, the coefficient of heat loss through the lower part of the collector is:

where is the thermal conductivity of the insulation;

is the thickness of thermal insulation.

The coefficient of heat losses through the upper part of the collector was determined based on an empirical expression obtained by Kleinom [7]:

where is the temperature of the absorbing surface, K;

is ambient temperature, K;

is the coefficient;

is the coefficient of convective heat transfer of the collector upper surface;

is velocity of the ambient medium over the surface, m/s;

is Stefan-Boltzmann constant;

is the emissivity factor of the absorber;

is the emissivity factor of the glass coating.

The solar collector can be considered as a sheet-profile system (Fig. 2).
\( \lambda_s \) is the thermal conductivity of the sheet, \( \lambda_s = 192 \text{ W/(m-K)} \); 
\( \delta_s = 0.0008 \text{ m} \) is the thickness of the sheet.

The net energy absorbed by the collector is transferred to the liquid. Thermal transfer resistances consist of the thermal resistance of the profile wall contact connection thermal conductivity and the heat transfer resistance of the liquid flowing in the collector.

The thermal resistance of the profile is determined by using the superposition method [8]:

\[
R_p = \frac{1}{2\pi\lambda_p} \cdot \ln \left[ \frac{2s}{\pi d_e} \cdot \text{sh} \left( \frac{2\pi h}{s} \right) \right], \tag{6}
\]

where \( \lambda_p = 192 \text{ W/(m-K)} \) is the thermal conductivity of the profile;

\[
d_e = \frac{2 \cdot a \cdot b}{a + b} = \frac{2 \cdot 0.015 \cdot 0.006}{0.015 + 0.006} = 0.00857 \text{ m}
\]
is the equivalent diameter of the profile holes;

\( a = 0.015 \text{ m} \) is the hole length;

\( b = 0.006 \text{ m} \) is the hole width;

\( s = 0.00825 \text{ m} \) is the distance between profile holes;

\( h = 0.0053 \text{ m} \) is the distance from the hole center to the profile surface.

The thermal resistance of the heat transfer of liquid flowing in the collector is:

\[
R_\alpha = \frac{1}{\alpha_f \pi d_e}, \tag{7}
\]

where \( \alpha_j \) is the heat transfer of the liquid in the collector pipes, determined by the criterion equations [8] depending on the liquid flow velocity in the channels of the solar collector profile.

The thermal resistance of the contact connection with a reliable contact is no more than

\[ R_c = 0.03 \text{ m}^2\cdot\text{K}/\text{W}. \]

Thus, the collector efficiency coefficient can be found:

\[
F' = \left[ \frac{1}{W + \frac{W}{L}} + \frac{W \cdot U_L \cdot R_p + W \cdot U_L \cdot R_\alpha}{(W - L) \cdot F} \right]^{-1} \tag{8}
\]

For this collector, the heat transferred to the liquid is determined by the formula [9]:

\[
Q = S_c F_i \left[ IR \cdot \eta_0 - U_L \left( T_{f0} - T_e \right) \right],
\]

where \( S_c \) is the area of the absorbing surface, \( m^2 \), for the considered collector \( S_c = 1.808 \text{ m}^2 \);

\( F_i \) is the coefficient of heat removal from the collector;

\( IR \) is the total intensity of solar energy radiation on the collector surface;

\( U_L \) is the total heat losses coefficient, \( \text{W/(m}^2 \cdot \text{K)} \);

\( T_{f0} \) is the temperature of the liquid at the inlet to the collector, \( \text{K} \);

\( T_e \) is the ambient temperature, \( \text{K} \);

\( \eta_0 = 1 - (1 - d) - (1 - d)(1 - a) = 1 - (1 - 0.88) - (1 - 0.88)(1 - 0.85) = 0.862 \) is the optical efficiency;

\( d = 0.88 \) is the light transmission coefficient of the polycarbonate coating of the collector;

\( a = 0.85 \) is the absorption capacity of the selective coating.

The coefficient of heat removal from the collector can be found by:
\[ F_i = \frac{Gc_p}{U_L} \left(1 - \exp\left[-\frac{U_i F'c_p}{Gc_p}\right]\right), \]

where \( G \) is the flow rate of the liquid per unit area of the collector, kg/(s\cdot m^2); \( c_p \) is the heat capacity of the liquid J/(kg\cdot K); \( F' \) is the coefficient of the absorbing surface that takes into account the geometry of the absorber.

The temperature of the liquid at the collector outlet can be found by [10]:

\[ T_f = T_u + \frac{IR \cdot \eta_a}{U_L} + \left(T_{fi} - T_u - IR \cdot \eta_a/U_L\right) \cdot \exp\left[-\frac{U_L \cdot W \cdot F \cdot l_m}{Mc_p}\right], \]

where \( l_m = 1.95 \) m is the length of the collector pipes;
\( M \) is mass flow rate of water passing through the collector, kg/s.

The average temperature of the absorbing plate will be found by:

\[ T_{pi} = T_f + \frac{Q_s}{\alpha_f \cdot \pi \cdot d_f \cdot k \cdot l_m}, \]

where \( k \) is the number of pipes in the collector.

The efficiency of the collector is:

\[ \eta_a = \eta_a - \frac{U_L(T_f - T_u)}{IR}. \]

3. Results and Discussion

The calculation of the efficiency coefficient and water temperature at the collector outlet was carried out by iterations at specified ambient temperatures \( T_a = 278 \) K; at the collector inlet \( T_{fi} = 288 \) K, the absorbing surface is \( T_{pi} \) with the further refinement. The mass flow rate of water was assumed to be \( M = 1.35 \) kg/s, the atmospheric air velocity was \( v = 5 \) m/s.

To calculate the heat transfer from pipes to water the following dependences of thermophysical parameters were used:

\[ \text{Nu} = 0.15 \cdot \text{Re}^{0.33} \cdot \text{Pr}^{0.33} \cdot \text{Gr}^{0.1} \] is the Nusselt number;
\[ \text{Re} = \frac{\omega_f \cdot d_f \cdot \nu_f}{\nu} \] is the Reynolds number, where \( \omega_f \) is the water velocity in the pipes;
\[ \text{Pr} = \frac{\nu_f}{\alpha_f} \] is the Prandtl number;
\[ \text{Gr} = 9.81 \cdot \beta \cdot (T_{pi} - T_f) \frac{d_f^3}{V_f^2} \] is the Grashof number;
\[ \alpha_f = \text{Nu} \cdot \lambda_f / d_f \] is the heat transfer coefficient;
\[ \lambda_f = 0.474 \cdot (1 + 0.0043t_f) , \text{W/(m\cdot °C) is the water thermal conductivity; } \]
\[ V_f = \frac{1.78 \cdot 10^{-6}}{1 + 0.0337 \cdot t_f + 0.00221 \cdot t_f^2}, \text{m}^2/\text{s is the coefficient of kinematic water viscosity; } \]
\[ a_f = 1.32 \cdot (1 + 0.003t_f) \cdot 10^{-7}, \text{m}^2/\text{s is the water temperature conductivity; } \beta_f = 0.00013 - 5.6 \cdot 10^{-5} t_f + 3.8 \cdot 10^{-6} t_f^2, \text{°C}^{-1} \] is the coefficient of linear water expansion.

As the result of the calculation, the water temperature changes at the collector outlet, the average temperature of the heat–storage medium and the absorber temperature over time during threefold circulation (Figure 3) were determined. Further circulation to increase the temperature of the heat–storage medium is not effective, since the overall efficiency of the temperature increasing tends to zero due to increased heat losses.
As can be seen from the figure, the absorber design has a large heat transfer, which increases the collector capacity and heat production compared to other designs.

The collector efficiency depends on factors such as the intensity of solar radiation and the temperature of the heat-storage medium flowing in the collector. Results to determine the effectiveness of the solar collector allowed obtaining the collector efficiency dependence on the solar radiation intensity (Figure 4) and the efficiency dependence on the temperature changes of water heating in the solar collector (Figure 5).

**Figure 3.** The water temperature change at the collector outlet during threefold circulation

**Figure 4.** The dependence of collector efficiency on the solar radiation intensity
Figure 5. The efficiency dependence on the water heating temperature changes in the solar collector

Thus, a solar collector with extruded profiles at a solar radiation intensity of 450 W/m² or more has a solar energy efficiency of more than 50%, which indicates a high collector efficiency. At the same time, in order to achieve the collector efficiency and to obtain the maximum heat output, it is necessary to ensure the heat-storage medium heating degree in the collector of no more than 30-35 °C.

It should also be noted that with the change in solar radiation input and with the change in the heat-storage medium temperature in the collector, the heat removal coefficient changes from the collector varies little, its value ranges from 0.854 to 0.877.

4. Conclusion

The collector with the extruded absorber considered in this paper has simplicity of design elements and their fabrication, low metal intensity, higher efficiency of heat transfers and low heat losses to the ambient medium, high heat output compared to other collectors’ designs. The model has unique thermophysical and hydrodynamic characteristics. The collector mathematical model calculation showed its high efficiency; especially the best results were obtained when the heat-storage medium was heated to 50 °C, which can be used for hot water supply systems.

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