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THERMAL OPTIMIZATION THE WIDTH OF LIGHT ABSORBING PLATE OF SHEET-TUBE SOLAR ABSORBERS FOR PREHEATING OF FEED WATER IN COMBINED SOLAR-FUEL SYSTEMS OF HOT WATER SUPPLY

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Abstract

In the current paper presented the results of computational studies on the thermal optimization the width of the plate of sheet-tube solar absorber to preheat feed water to the combined solar-fuel hot water systems, in which their maximum heat with minimum weight.

Keywords: Solar absorber, combined solar-fuel boiler, hot water consumption systems

Introduction One of the promising areas of practical application of solar energy, which currently has the greatest level of technological readiness of the Republic of Uzbekistan, is conversion of solar energy into a low-potential heat and use it as a source of heat supply systems in residential, municipal and public buildings that are major consumers of the same temperature potential.

The basic element of solar heating systems is a flat plate collector, where absorbing of solar radiation and converting it to low-potential heat then transferring to a heated water, flowing through heat transferring channels of its light absorbing panels.

All solar collectors existing in the world differs mainly in its design, material and manufacturing technology of the absorbers. Specific (referred to $1m^2$ surface area) cost of collectors, depending on their degree of perfection of heating ranges from 200 (for heating water to 50÷55°C) to 600 (to 80÷85°C) USD, from an economic point view is the main obstacle to their widespread use. To improve the economic efficiency of flat plate solar collectors in hot water supply systems the possibility of replacing them with a flat absorber insulation in which there is no traditional housing and translucent cover. To extend the using period of solar absorbers in hot water supply systems during the year and increase their reliability proposed combined solar-thermal systems instead of only solar. In solar thermal hot water systems solar absorbers are used to preheat feed water to (40-45^{o}C). Pre-heating of water to condensing temperature (65^{o}C) is carried out in boiler rooms operating on fossil fuel.

Principal Scheme and thermal characteristics of flat plate solar absorbers for preheating of feed water

Flat plate sheet and tube solar absorber is a blackened metal plate with half-grooves, which are stacked and secured a number of parallel pipes, heat-removing channels connected by a common hydraulic channels, providing a uniform distribution of coolant flow through the pipes.

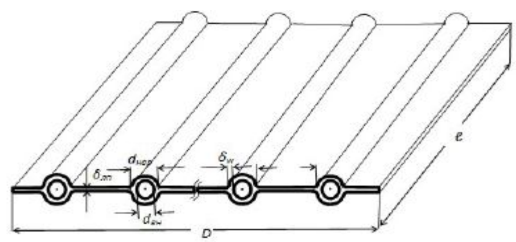


Fig.1. Principal Scheme of flat plate sheet-tube solar absorber for preheating of heat transfer liquid, prepared by the method of extrusion: *D* and ℓ - width and length of panel.

Due to the lack of housing and translucent coating heat loss planar absorbers with thermally insulated bottom, while in 2,0÷2,5 times more than conventional solar collectors of average quality, by eliminating the optical losses due to reflection of sunlight from the surface of the translucent cover, thermal efficiency of flat absorbers with other conditions being equal, for example, by heating water to the normalized temperature for hot water at 55°C, almost identical, and the value of 2.0-2.5 times cheaper than traditional collectors. If you use these absorbers to preheat feed water in the solar-fuel hot water systems up to 35 ÷ 45 °C, by reducing the heat loss of thermal efficiency in the warm season in the climatic conditions of Uzbekistan can be increased up to 0,80÷0,85, which is almost 2 times more than conventional reservoirs [1].

As the results of studies [2,3], the thermal efficiency of flat solar absorber depends mainly on their overall heat loss coefficient ($K_{np_{p-o}}$) and thermal coefficient, transfer efficiency of absorbed characterizing them to the flow of radiation heat-removing heated water channels (η_{mn}).

In this regard, one of the major challenges facing developers of new high-efficiency solar absorbers generations and on the basis of solar collectors is to find ways to ensure their maximum thermal performance with minimal weight.

Calculations

According to [4,5], the expression for determining the thermal efficiency of sheet-tube absorbing panels (i.e. solar absorbers) with ideal thermal contact between absorbing plate and heat-removing channels as following

$$\eta_{hp} = \left\{ \left(2a_{bp} + d_{out}\right) K_{norm_{p-o}} \times \left[\frac{1}{\left(2a_{bp}\eta_{bp} + d_{out}\right) K_{norm_{p-o}}} + \frac{1}{\frac{\lambda_w}{\delta_w} b_3} + \frac{1}{\alpha_{c_{in}} b_4} \right] \right\}, \quad (1)$$

where

$$\eta_{bp} = \frac{th\left(a_{bp}\sqrt{\frac{K_{norm_{p-o}}}{\delta_{bp}\lambda_{bp}}} \right)}{a_{bp}\sqrt{\frac{K_{norm_{p-o}}}{\delta_{bp}\lambda_{bp}}}} \quad (2)$$

the thermal efficiency coefficient of the absorbing plate of the panel; λ_{bp} is the thermal conductivity coefficient of the material of the absorbing plate; λ_w is the thermal conductivity coefficient of the material of heat conducting channel's wall;

$$b_3 = \frac{0,5\pi}{\varphi}\left(d_{out} - d_{in}\right) \quad (3)$$

the perimeter of the middle cross section of the heat-removing channel; $\alpha_{c_{in}}$ the coefficient of convective heat exchange of the inner surface of the wall of the heat-removing channel's;

$$\varphi = \frac{\left(d_{out} - d_{in}\right)}{2\left(d_{out} - d_{in}\right)} \ln\frac{d_{out}}{d_{in}} \quad (4)$$

the curvature coefficient of heat-removing channel of round shape;

$$b_4 = \pi d_{in} \quad (5)$$

is the perimeter of the inner cross-section of heat-removing channel; a_{bp} half of the width of the absorbing plate; $K_{norm_{p-o}}$ is the coefficient of the total heat losses of absorbing heat-exchange panel normalized to the unit area of frontal surface; d_{in} and d_{out} are the inner and outer diameters of the heat-removing channel of absorbing heat-exchange panel, correspondingly.

One of the most important problems in the production of high-efficiency and less metal using solar absorbers for heating of heat transfer fluid is the search for the method of achieving the maximal thermal productivity of their absorbing heat-exchange panels at minimal mass.

For absorbing heat-exchange panels whose absorbing plates and heat-removing channels are made of metals with thermal conductivity coefficient. $\lambda_w = 40 - 350\ W/\left(m^2{}^\circ C\right)$ and width $\delta = \left(0.5 - 1.0\right)\times 10^{-3} m$ at $a_{bp} = 0.05m$, $\lambda_{bp} = 400 - 350\ W/\left(m^\circ C\right)$, $\delta_{bp} = \left(0.5 - 1.0\right)\times 10^{-3} m$, $K_{norm_{p-o}} = 7.5\ W/\left(m^2{}^\circ C\right)$, $\eta_{bp} = 0.89 - 0.97$ and $\alpha_{c_{in}} = 150 - 600\ W/\left(m^2{}^\circ C\right)$, $d_{in} = 0.010m$, $d_{out} = 0.012m$.

According to the results of simple thermal - technical calculations, the value $\frac{K_{norm_{p-o}}\delta_w\left(2a_{bp} + d_{out}\right)}{\lambda_w b_3}$ constitutes $\left(69.4 - 486.1\right)\times 10^{-6}$, which is 2480-16610 times less than the value of the sum $\frac{2a_{bp} + d_{out}}{2a_{bp}\eta_{bp} + d_{out}} + \frac{\left(2a_{bp} + d_{out}\right)K_{norm_{p-o}}}{\alpha_{c_{in}} b_4}$, which equals 1.153-1.206. Therefore, expression (1), which takes into account the value of b_4, can be denoted as follows:

$$\eta_{hp} = \frac{\left(2a_{bp}\eta_{bp} + d_{out}\right)\pi d_{in}\alpha_{c_{in}}}{\left(2a_{bp}\eta_{bp} + d_{out}\right)\left[\left(2a_{bp}\eta_{bp} + d_{out}\right)K_{norm_{p-o}} + \pi d_{in}\alpha_{c_{in}}\right]}. \quad (6)$$

An analysis of expression (6) implies that, to achieve high values of η_{hp} at the same other conditions, it is necessary to achieve the highest possible values of η_{bp} and $\alpha_{c_{in}}$.

In order to achieve high values of η_{bp} with all other conditions being the same ($K_{norm_{p-o}}$ and λ_{bp}), it is necessary to decrease the ratio of a_{bp}/δ_{bp}, i.e., to increase δ_{bp} or decrease a_{bp}.

Unfortunately, the increase in η_{bp} and, consequently, η_{hp} as is implied by the expression

$$m_{hp}^{rapt} = \frac{M_{hp}}{F_{hp}^{fr}} = \frac{2a_{bp}\delta_{bp}\rho_{bp} + 0.25\pi\left(d_{out}^2 - d_{in}^2\right)\rho_{hr}}{2a_{bp} + d_{out}}, \quad (7)$$

implies leads to the growth of the partial material capacity (normalized to the unit area of the frontal surface F_{hp}^{fr}) of the panel under consideration.

In expression (7),

$$M_{hp} = M_{bp} + M_{hr} \quad (8)$$

is the mass of absorbing heat-exchange panel, which is composed of masses of absorbing plates (M_{bp}) and heat-removing channels (M_{hr});

$$F_{hp}^{fr} = Dl \quad (9)$$

is the area of the frontal surface of the absorbing panel;

$$M_{bp} = 2a_{bp}\delta_{bp} \ln \rho_{bp}; \quad (10)$$

$$M_{hr} = 0.25\pi\left(d_{out}^2 - d_{in}^2\right)\ln \rho_{hr}; \quad (11)$$

$$n = \frac{D}{2a_{bp} + d_{out}} \quad (12)$$

is the number of heat-removing channels per unit of width of sheet-tube absorbing panel; D and l are the width and length of the panel, respectively; ρ_{bp} and ρ_{hr} are the densities of the materials of the absorbing plate and heat-removing channel of the panel under consideration, respectively.

As a rule, for stamp-welded and roll-welded absorbing heat-exchange panels as well as for panels of pipes with longitudinal edges produced by extrusion $\rho_{bp} = \rho_{hr}$.

One should note that, according to (12), the decrease of α_{bp}, with all other conditions being the same (δ_{bp}), leads to the growth of a number of heat-removing channels on the unit of width of the panel under consideration (n) the expression

$$w = \frac{4G}{n\pi d_{in}^2} \qquad (13)$$

implies that the increase in n at the given value of volume flow of the heat carrier (G) through the given panel leads to decrease in its rate in heat-removing channels (w), which leads to decrease of $\alpha_{c_{in}}$ on the inner surfaces of their walls.

The increase of $\alpha_{c_{in}}$ on the inner surfaces of walls of heat-removing channels of absorbing heat-exchange panels at the same other conditions (d_{in}) can be achieved at the expense of a corresponding increase of the volume flow of the heat carrier (G). However, according to the studies of the dependence $\eta_{hp} = f(G)$ in two types of heat absorbing panels [4] increase in G in heat-exchange channels of absorbing heat-exchange panels of flat plate solar collectors is only appropriate until certain values (30-40 l/h), after which the dependence of η_{hp} on G becomes insignificant. On the other hand for increasing values of G in heat-removing channels, the corresponding increase in the flow of mechanic (electric) energy is required; this energy is used for the forced circulation of the heat carrier through the channels. In addition, despite the relatively high value of $\alpha_{c_{in}}$ at large values of G, the heat carrier on the outlet of the collector will have a relatively low temperature and multiple circulations of the heat carrier through the given collector during daylight hours is required in order to heat it to the necessary temperature.

At first the presence of these contradictory, yet mutually conditioning, formation factors of η_{hp} allows one to assume the existence of its optimal value depending on a_{bp} and G. However, the plotting and analysis of curves of the dependences $\eta_{hp} = f(a_{bp}, G)$ by the expression (6) with the corresponding consideration of dependencies of η_{bp} and a_{bp} (at the given δ_{bp}, $K_{norm_{p-o}}$ and λ_{bp}) based on the expression (2) and $\alpha_{c_{in}}$ on G (via w at the given d_{in}) based on the expression (13) shows that, when $\alpha_{bp} \to 0$ and $G \to \infty$, the quantity η_{hp} approximates its maximal (asymptotic) value, which is 1.

It can be seen that it is not sufficient to consider η_{hp} as a fully optimized parameter for developing an optimal construction of absorbing panel for a flat plate solar absorber and collector on their base and choosing the optimal flow of the heat carrier through it.

In connection with this, this work (which is a development of [3]) suggests that one should solve the problem under consideration and take into account the dependence of $\alpha_{c_{in}}$ on a_{bp} at the given values of G.

At the same time, the partial thermal productivity of the considered absorbing heat-exchange panel is defined from the expression 2,3

$$q_{tot} = \eta_{hp}\left[\alpha_p q_{fall}^{\Sigma} - K_{norm_{p-o}}\left(\bar{t}_f - t_o\right)\right], \qquad (14)$$

where α_p is the absorbing coefficient of total solar irradiation of the absorbing panel of the considered solar absorber; q_{fall}^{Σ} is the surface density of flow of total solar irradiation falling on the plane of frontal surface of absorbing panel; $\bar{t}_f$ is the average temperature of the heat carrier over the length of heat-removing channel; and t_o is the temperature of environment.

Substituting (6) into (7) and (14) and taking into account the value of m_{hp} according to (7), the expression for the ratio $\frac{q_{tot}}{m_{hp}^{part}}$ can be denoted as:

$$\frac{q_{tot}}{m_{hp}^{part}} = \frac{\left(2a_{bp}\eta_{bp} + d_{out}\right)\pi d_{in}\alpha_{c_{in}}\left[\alpha_p q_{fall}^{\Sigma} - K_{norm_{p-o}}\left(\bar{t}_f - t_o\right)\right]}{\left[\left(2a_{bp}\eta_{bp} + d_{out}\right)K_{norm_{p-o}} + \pi d_{in}\alpha_{c_{in}}\right]\left[2a_{bp}\eta_{bp} + 0.25\pi\left(d_{out}^2 - d_{in}^2\right)\right]\rho_{bp}} . \qquad (15)$$

Because it is extremely difficult to differentiate expression (15) as a target function for determining the critical values of a_{bp} taking into account its influence on $\alpha_{c_{in}}$ (at the given G), depending on G, the value of a_{bp} is defined based on its graphical solution.

According to the results of preliminary calculations on the determination of the rate of the heat carrier in heat-removing channels of sheet-tube absorbing panels of flat solar absorbers and collectors on their base for heating a liquid heat carrier using expression(13) in the range of change in $G_{part} = \frac{G}{F_{hp}^{fr}}$ from 10 to 100 *l/(m² h)*, change in n from 5 to 20 units/m, and change in d_{in} from 0.008 to 0.015m, the flow mode of the heat carrier in heat-removing channels is laminar and the character is viscous-gravitational. In connection with this, calculations for determining the values of $\alpha_{c_{in}}$ depending on G and average temperatures of the inner surface of the wall $\bar{t}_{w_{in}}$ of the heat-removing channel of the panel under consideration and the heat carrier in it ($\bar{t}_f$) satisfy the generally accepted criterial equations presented on M.A.Mikheev [4].

Calculation results at $t_0 = 30°C$ and $t_{in} = 20°C$ are shown in Fig.2.

As can be seen from a comparison of graphic dependences $\frac{q_{us}}{m_{hp}^{sp}} = f(a_{bp})$ on the Fig.2., maximal value of $\frac{q_{us}}{m_{hp}^{sp}}$, as expected, with increasing values of $K_{r_{p-o}}$ are shifted to the left. So, if at $K_{r_{p-o}} = 15\ W/(m^2°C)$ $\left(\sqrt{\frac{K_{r_{p-o}}}{\delta_{bp}\lambda_{bp}}} = 6.98\frac{1}{m}\right)$ and $G = 45l/h$ maximal value of $\frac{q_{us}}{m_{hp}^{sp}}$ is $185.0\frac{W}{kg}$ then at same value of G (i.e. 45 l/h) at $K_{r_{p-o}} = 20\ W/(m^2°C)$ $\left(\sqrt{\frac{K_{r_{p-o}}}{\delta_{bp}\lambda_{bp}}} = 11.18\frac{1}{m}\right)$ values maximal value of $\frac{q_{us}}{m_{hp}^{sp}}$ will be 162,78W/m^2.

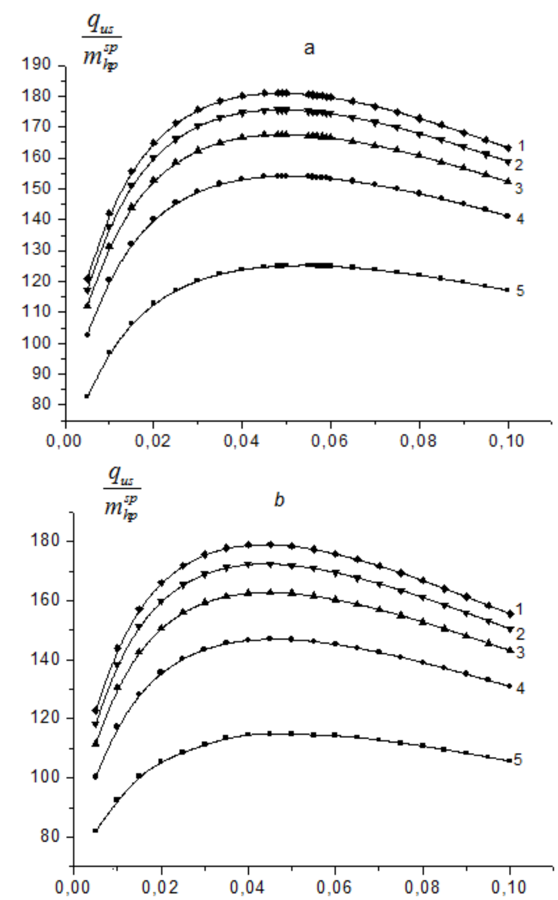
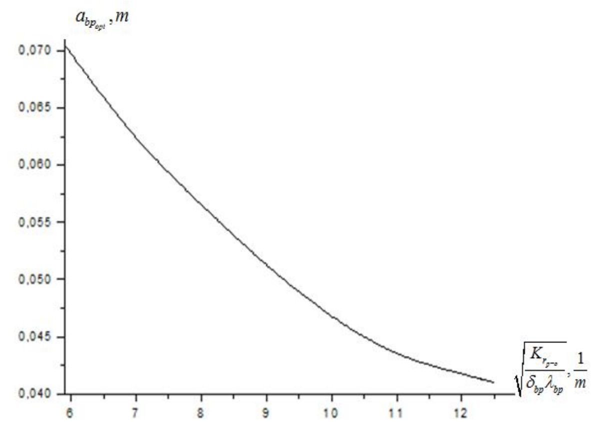


Fig.2. Dependence $q_{us}/m_{hp}^{sp} = f(a_{bp})$ at $\alpha_p = 0.8$, $q_{fall}^{\Sigma} = 750W/m^2$, $\lambda_{bp} = 160W/(m^2°C)$, $\delta_{bp} = 0.001m$, $t_0 = 30°C$ and $t_{in} = 20°C$: ***a*** and ***b*** at $K_{r_{p-o}} = 15$ and $K_{r_{p-o}} = 20\ W/(m^2°C)$, respectively 1,2,3,4 and 5 at $G = 15, 30, 45, and\ 75l/h$.

The results of calculations derived from data in Fig.2 in the form of a graphical dependence shown in Fig.3. As can be seen from Fig.2, if the solar absorber sheet-tube considered for water heating, the optimum width value of light absorber plate (i.e., half the inter-tube distance of the absorber at 15 and 20 are 0.06 and 0.043 *m*, respectively.

The results of calculations derived from data in Fig.2 in the form of a graphical $a_{bp_{opt}} = f\left(\sqrt{\frac{K_{r_{p-o}}}{\delta_{bp}\lambda_{bp}}}\right)$ dependence shown in Fig.3.

As can be seen from Fig.3, if the solar sheet-tube absorber considered for water heating, the optimum value of width of light absorbing plate (i.e., half the inter-tube distance of the absorber at $K_{r_{p-o}} = 15$ and $K_{r_{p-o}} = 20\ W/(m^2°C)$ are 0.06 and 0.043 m, respectively.



In Fig. 3. The dependence of the optimal value of half the distance between the tubes, leaf-tube solar water-heating of the absorber a parameter.

Conclusion

From the analysis of the obtained results it follows that the optimal value of the distance between the tubes leaf-tube solar heat absorber heat transfer fluid depends strongly on their thermal parameters.

The proposed methodology and the results can be used by enterprise-producers of flat solar collectors and absorbers for heating the heat transfer fluid to determine the optimal value of the distance between the tubes of heat exchangers sheet-tube absorber panels to ensure maximum thermal performance with minimal weight.

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