OPTIMIZATION OF INTER TUBULAR DISTANCE OF SHEET-TUBE HEAT TRANSFER PANELS OF FLAT PLATE SOLAR WATER HEATING COLLECTORS

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1. Introduction

The heat-mechanical optimization method of inter-tubular distance of sheet-tube light absorbing panels of flat plate solar collectors for heating of heat-transfer fluid is proposed, which is included the dependence to coefficient of convective heat exchange interior wall surface of the heat-removing channel on the optimized parameter and expenditure of heat-removing through the channels.

There is an absorption and conversion of solar radiation to the low-potential heat and transfer to the water, in heat exchanger panel of flat plate solar water heating collectors. The most common type of heat exchanger panels used in flat plate solar water heating collectors, is sheet-tube design, which is a blackened light absorbing metal plate with slots, which are stacked and secured by soldering series of heat transfer parallel tubes, connected each other by a common hydraulic feed and outlet channels, providing uniform distribution of the flow of heated water on the heating channels (Fig. 1).

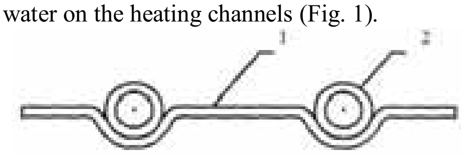


Fig.1. Principal Scheme of cross section of sheet-tube light absorbing panel of flat plate solar water collector: 1-light absorbing panel with channels; 2-tube.

As mentioned on [1,2], the thermal efficiency of solar collectors of this type mainly depends on the coefficient of thermal efficiency of heat transfer panels, which characterizes the transferring efficiency of absorbed solar radiation to the water heated in their heat transfer. In this regard, one of the major problems facing developers of a new, highly efficient and low metal-consuming generations of flat plate solar water collectors is to find ways to maximize the heat output of heat exchanger panels with minimum weight.

2. Mathematical description

Expressions for determining the specific thermal productivity and mass of sheet-tube heat exchanger panel of flat plate solar water heating collectors per unit area of the front surface, according to [2] will have the form

$$q_u = \eta_p \left[(\alpha_p \tau)_{eff} q_\perp^\Sigma - U(\overline{T}_f - T_o) \right] \tag{1}$$

$$m_p^{sp} = \frac{M_p}{A_p} = \frac{2a_{ap}\delta_{ap}\rho_{ap} + 0,25\pi(d_{ext}^2 - d_{in}^2)\rho_{ch}}{2a_{ap} + d_{ext}}, \tag{2}$$

where

$$\eta_p = \left\{ \left(2a_{ap} + d_{ext}\right)U \left[\frac{1}{(2a_{ap}\eta_{ap} + d_{ext})U} + \frac{1}{\frac{k_w}{\delta_w}\overline{\Pi}} + \frac{1}{\alpha_{in}\Pi_{in}} \right] \right\}, \tag{3}$$

- Coefficient of thermal efficiency sheet-tube panels with a perfect thermal contact between the light absorbing plate and the heat-removing channels;

$$\eta_{ap} = \frac{th\left(a_{ap}\sqrt{\frac{U}{\delta_{ap}k_{ap}}} \right)}{a_{ap}\sqrt{\frac{U}{\delta_{ap}k_{ap}}}} \tag{4}$$

- Coefficient of thermal efficiency light absorbing plate of panel; δ_{ap} - wall thickness of the light absorbing plate; k_{ap} - thermal conductivity of the material of light absorbing plate; k_w - thermal conductivity of the material of heat-removing channel's wall;

$$\overline{\Pi} = \frac{0,5\pi}{\phi}(d_{ext} - d_{in}) \quad (5)$$

- The perimeter of the average cross-section of the heat-removing channels; α_{in} - convective heat transfer coefficient of inner wall surface of the heat-removing channels;

$$\phi = \frac{(d_{ext} - d_{in})}{2(d_{ext} - d_{in})} \ln \frac{d_{ext}}{d_{in}} \quad (6)$$

- Coefficient of curvature heat-removing channel with circular form;

$$\Pi_{in} = \pi d_{in} \quad (7)$$

- The perimeter of the inner cross-section of heat-removing channel; a_{ap} - half the width of the light absorbing plate; U - coefficient of total heat loss of heat transfer panels, reduced to unit area of the front surface; d_{in} and d_{ext} accordingly, the internal and external diameters of the heat-removing channels of heat transfer panels; $(\alpha_p \tau)_{eff}$ is the effective reduced absorptivity of solar radiation, system "heat transfer panel – light transparent layer" of considered solar collector; $q_{\perp}^{\Sigma}$ - the surface flux density of the total solar radiation on the front surface of the heat exchange panels; $\overline{T}_f$ - the average temperature of heat transfer fluid by the length of heat-removing channel; T_o - the temperature of the environment; ρ_{ap} and ρ_{ch} - respectively, the density of materials of light absorbing plates and heat-removing channels of considered panel.

Substituting (2) and (3) into (1), we obtain

$$\frac{q_u}{m_p^{sp}} = \frac{(2a_{ap}\eta_{ap} + d_{ext})\pi d_{in}\alpha_{in}\left[(\alpha_p\tau)_{eff} q_{\perp}^{\Sigma} - U(\overline{T}_f - T_o)\right]}{\left[(2\alpha_{ap}\eta_{ap} + d_{ext})\mathrm{U} + \pi d_{in}\alpha_{in}\right]\left[2a_{ap}\eta_{ap} + 0,25\pi(d_{ext}^2 - d_{in}^2)\right]\rho_{ap}} \quad (8)$$

As follows from relation (8), when given d_{in}, d_{ext}, ρ_{ap} and ρ_{ch} the task of ensuring maximum value of $\frac{q_u}{m_p^{sp}}$ for the sheet-tube heat exchanging panels of flat plate solar water heating collectors is reduced to determining the critical value of half the width of the light absorbing plate $(a_{ap})_{cr}$ depending on the expected mode of operation.

Due to the fact, that the expression (5), as the objective function to determine the critical values a_{ap} with taking into account of the influence of α_{in} (for given values of the flow of coolant through the heat-removing channels - *G*) too difficult to differentiate, value of a_{ap} depending on the value of *G* could be determined by its graphical solution.

3. Results

As shows the results of preliminary calculations to determine the possible values of the rate of heat transfer fluid (coolant) in the heat-removing channels of sheet-tube heat exchanging panels of flat plate solar water heating collectors in the range of G from 10 to 100 *l/h*, and d_{in} - from 0,008 to 0,015 *m*, the flow mode of the coolant in heat-removing channels is laminar, and character – viscous - gravitational. In this regard, the calculations on determination the values of α_{in} dependently to *G* and average temperatures of the inner wall surface of the heat-removing channels of considering panel ($\overline{T}_{w_{in}}$) and coolant in it ($\overline{T}_f$) when it is expedient to perform on universally accepted criterial equations of M.A. Mikheev [3]

$$Nu_{fd} = 0{,}17 Re_{fd}^{0,33} Pr_f^{0,43} Gr_{fd}^{0,1} \left(\frac{Pr_f}{Pr_w}\right)^{0,25} \overline{\varepsilon}_e, \quad (9)$$

Where, taken into account the influence of free convection to coefficient of convective heat transfer by forced laminar viscous-gravitational flow of coolant in the heat-removing channels.

In criterial expression (9)

$$Nu_{fd} = \frac{\alpha_{in} d_{in}}{k_f} \tag{10}$$

$$Re_{fd} = \frac{v d_{in}}{\nu_f} = \frac{4G(2a_{ap} + d_{ext})}{\pi D d_{in} \nu_f} \tag{11}$$

$$Pr_f = \frac{\nu_f}{a_f} \tag{12}$$

$$Gr_{fd} = \frac{\beta_f g(\overline{T}_{w_{in}} - \overline{T}_f) d_{in}^3}{\nu_f^2} \tag{13}$$

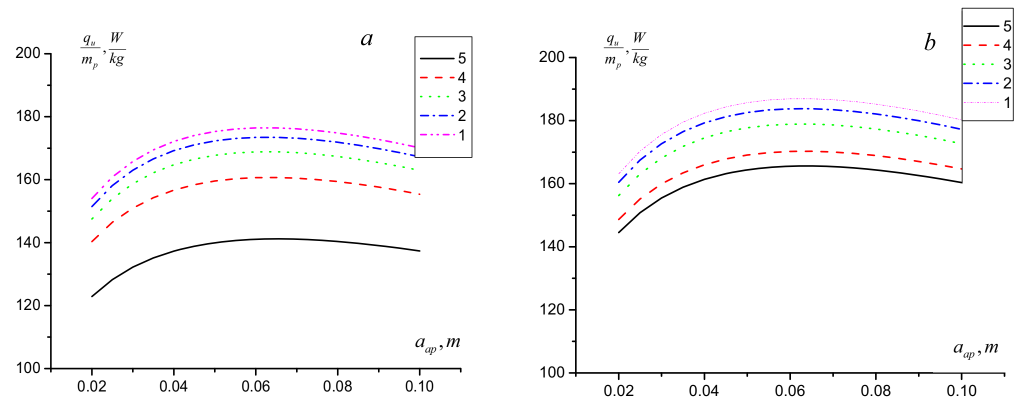
$$Pr_w = \frac{\nu_w}{a_w} \tag{14}$$

where, $\overline{T}_w$; k_f, v, ν_f, a_f and β_f - are respectively the coefficients of thermal conductivity, the rate of the coolant in heat-removing panels, the kinematic viscosity, temperature conductivity and thermal expansion of coolant; $\overline{T}_f$; ν_w and a_w - are respectively the kinematic viscosity and temperature conductivity of the coolant; g - acceleration of free fall; $\overline{\varepsilon}_e$ - coefficient of the average change rate value of α_{in} by the length of heat-removing channel of panel (at $(e/d_{in}) \geq 50$, $\overline{\varepsilon}_e = 1$).

Using the value for α_{in} obtained by equation (9) under given values of d_{in}, d_{ext}, δ_{ap} and U, and the current values of a_{ap}, η_{ap} and $\overline{T}_f$ by (3) could be determined the value of η_p.

With the obtained value of η_{ap} by the expression (14) dependently $(\alpha_p \tau)_{eff}$, $q_{\perp}^{\Sigma}$ and T_o under given value U, and also value $\overline{T}_f$, determined by iterative calculations by definition α_{in}, could be determined the value of q_u. Next for given values δ_{ap}, ρ_{ap}, d_{in} and d_{ext} by (2) could be determined the specific mass of the considered light absorbing panel dependently from a_{ap}. And finally, on the base of obtained values q_u and m_p^{sp} would be determined the useful heat output of considered light absorbing heat-transfer panel, per unit of mass (q_u / m_p^{sp}).

The results of calculations to determine the optimal value of half a distance $(a_p)_{opt}$ between the tubes of sheet-tube heat transfer panels, made from corrosion-resistant aluminum alloy with length l = 1,5m and a width D = 1,0m, wall thickness of light absorbing plate δ_p = 0,001m when d_{in} = 0,010m, d_{ext} = 0,012m, $\rho_{ab} = \rho_{ch}$=2600 kg/m^3, (k_p) = 160 $W/m°C$, T_o=25°C, q = 150 W/m^2 and $(\alpha_p \tau)_{eff}$ = 0,8 (about midday hours of daylight) for different values of G shown in Fig. 2.



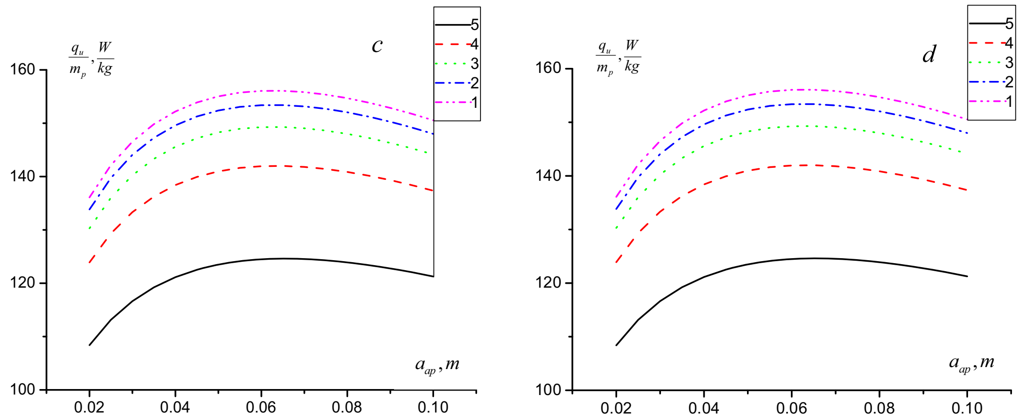


Fig.2 Dependence $\frac{q_u}{m_p^{sp}} = f(a_{ap})$ **under** $(\alpha_p\tau)_{eff}\, q_\perp^\Sigma = 600(W/m^2)$; $U = 7,5(W/(m^2\,^oC))$; $k_{ap} = 160(W/(m\cdot {}^\circ C))$; $\delta_{ap} = 0,001m$: *a,b,c,d–* respectively, under $T_{f_{in}} = 20^oC$ and $T_o = 25^oC$, $T_{f_{in}} = 20^oC$ and $T_o = 30^oC$, $T_{f_{in}} = 30^oC$ and $T_o = 25^oC$, $T_{f_{in}} = 30^oC$ and $T_o = 35^oC$; 1,2,3,4 and 5 respectively, when *G* =15, 30, 45, 60 and 75 *l/hr.*

From the analysis of the results shown, it follows that:

Increasing the volumetric flow rate of heat transfer fluid through channels of light absorbing panels will lead in a corresponding increase of q_{us}/m_p. However, the rate of increase is significantly reduced with increasing *G*. In all cases the maximum values of q_{us}/m_p occur when the value of a_p = 0,065*m*, which can be considered optimal for the given heat-exchange panel for its thermal-technical characteristics, equal *U* =7,5 *W/(m*^{2o}*C),* k_f =160W/(m^{o}C), δ_p = 0,001*m and* $\sqrt{U/\delta_p k_p}$ = 6,8465 m^{-1}. Independence of the optimal values of a_p from $(\alpha_p\tau)_{eff}\, q^\Sigma$, $T_o, T_{f_{in}}$ and G suggests appropriate independence of $(a_p)_{opt}$ from the environmental parameters and operating conditions of the proposed heat transfer panel. Due to the fact that near the critical point curve $q_{us}/m_p = f(a_p)$ has a flat character, if necessary (for technical reasons) be possible to vary the value of a_p ±0,01*m* near its critical value. When performing calculations for the generality of the solution weight of inlet and outlet hydraulic channels of the panels are not included, since they do not affect the optimal value of a_p.

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