HEAT EXCHANGE IN THE CHANNEL OF A FLAT SOLAR COLLECTOR

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ABSTRACT

The article studies the issue of calculating the efficiency of heat transfer in solar air heaters operating in the condition of free convection. Using the theory of the boundary layer of a heater absorber developing on a vertical surface, it is also possible to use the Reynolds analogy formula obtained for forced convection when flowing around flat surfaces in the case of a laminar regime.

Keywords: solar collectors, convection, heat transfer, hydrodynamics, Reynolds analogy, equation of motion, natural circulation.

INTRODUCTION

The operation of solar collectors and their heat transfer characteristics significantly depend on the hydrodynamics and heat exchange processes occurring in the channels, and hydrodynamic conditions almost completely determine the heat exchange ones, since the temperature profiles in the flow depend on the emerging velocity fields.

An analysis of hydrodynamics and heat transfer processes in the channels of a solar collector shows that the condition for optimal operation of a solar flat collector, consisting of flat solar receiving channels, is not only a stable natural upward movement of the coolant along the surface of the collector absorber, but also effective heat exchange between the collector wall and the air. Such a condition can be realized only if a certain connection is satisfied between important hydrodynamic characteristics such as shear stress and heat transfer coefficient. This relationship in theoretical heat engineering and solar engineering is called the Reynolds analogy [1].

RESEARCH METHOD

An analysis of literary sources [2] shows that, despite the large number of publications devoted to the study of solar collectors, there is still a certain gap in the study of the laws of friction and heat transfer during free convection in collector channels. The problem lies in studying the action of internal friction forces that determine the velocity and temperature fields in ascending currents. This heterogeneity causes convective accelerations in the liquid, when volumetric forces and inertial effects begin to dominate over viscous forces, in this case this influence can be taken into account by the criteria Gr and $\Pr$ [4]. Let us consider the analogy between high-speed and heat flow during natural convection, known in forced flows as the Reynolds analogy. Considering that during natural convection the boundary layer differs in structure from the hydrodynamic boundary layer during forced flow, we believe that the main driving force of the upward movement is only the difference in the densities of the liquid layer near the reservoir wall and at a distance from it. The influence of viscosity and, as a consequence, braking of the wall layer leads to the fact that the speed of fluid movement due to the volumetric Archimedean force first increases to a maximum value and then decreases to zero. We will assume that a laminar boundary layer develops on the wall. Let the channel temperature be equal to t_c. In accordance with [3], using known assumptions for free convection, we write the equation of motion, which for the stationary case has the form:

$$\mu(d^2\omega_x / dy) = -g(\rho_0 - \rho) \qquad (1)$$

With a linear dependence of density on temperature

$$\rho = \rho_0(1-\beta\vartheta) \qquad (2)$$

Where is $\beta -$ the temperature coefficient of volumetric expansion of the liquid [1/ C ^{0}]. ρ_0 and ρ are the liquid densities corresponding to the liquid temperatures t_0 and t; t_0 - some fixed temperature. In accordance with [3], we solve the equation of motion (1), taking into account that

$$\rho_0 - \rho = \rho_0\beta\vartheta \qquad (3)$$

Where $\vartheta = \vartheta_c(1-y/\delta)^2$ (4)

$$\vartheta = t - t_0 \text{ And } \vartheta_c = t_c - t_0$$

We get

$$d^2\omega_x / dy^2 = (\rho_0 g\beta\vartheta_c / \mu)(1-y/\delta)^2 \qquad (5)$$

Integrating the equation of motion (5) gives:

$$d\omega_x / dy = -(\rho_0 g\beta\vartheta_c / \mu)[y^2 - (1/3\delta)y^3 + (1/12\delta^2)y^4) + c_1 x + c_2 \qquad (6)$$

Border conditions

$y = 0$ $\quad \omega_x = 0$

$y = \delta$ $\quad \omega_x = 0$

Under accepted boundary conditions

$c_1 = (\rho_0 g\beta\vartheta_c / 4\mu)\delta$ And $c_2 = 0$

Thus we have

$$(d\omega_x / dy)_{y=0} = (\rho_0 g\beta\vartheta_c / 4\mu)\delta = (g\beta\vartheta_c \ell^3 / \nu^2)\nu\delta / 4 = (Gr\,\nu / \ell^3)(\delta / 4) \qquad (7)$$

Taking into account the adopted boundary layer model, we will assume that, to a first approximation [3], the thickness of the boundary layer can be expressed as

$$\delta = 5\ell / Gr^{1/4} \quad (8)$$

We get

$$(d\omega_x / dy)_{y=0} = Gr(\nu\delta / 4\ell^3) = Gr(\nu / \ell^3)(5\ell / 4Gr1/4) = 1{,}25Gr^{3/4}(\nu / \ell^2) \quad (9)$$

METHODS

The friction coefficient on the wall is equal to

$$c_f = \tau / \rho\omega^2 = \mu(d\omega_x / dy)_{y=0} / \rho\omega^2 = \nu(d\omega_x / dy)_{y=0} / \omega^2 = 1{,}25Gr^{3/4}(\nu^2 / \ell^2) / \omega^2$$
$$= [1{,}25Gr^{3/4}\nu^2][\ell^2 g\beta\vartheta_c\ell] = 1{,}25Gr^{3/4} / Gr = 1{,}25 / Gr^{1/4}$$

So

$$c_f = 1{,}25 / Gr^{1/4} \quad (10)$$

In order to analyze the intensity and efficiency of heat transfer, we use the Reynolds analogy equation [4]

$$2St / c_f = 2Nu / \mathrm{Re}\,\mathrm{Pr}\, c_f \quad (11)$$

Considering that $Nu = 0{,}555Gr^{1/4}$ (12)

We get

$$2St / c_f / c_f = (1{,}1Gr^{1/4}) / (Gr^{1/2}\,\mathrm{Pr}\,1{,}25Gr^{-1/4}) = 1{,}1 / 1{,}25\,\mathrm{Pr} = 1 / \mathrm{Pr} \quad (13)$$

In a boundary layer with natural circulation, the Reynolds analogy formula looks like this

$$2St / c_f = 1 / \mathrm{Pr} \quad (14)$$

CONCLUSION

The resulting formula [6] coincides in structure with the Reynolds analogy formula for forced flow. Note that at $\mathrm{Pr} = 1$ the intensity of friction and heat transfer on the surface of the absorber are comparable, which confirms the correctness of the obtained formula (14). For at its average temperature equal to $t_0 = 20C^0$ $(\mathrm{Pr} = 7)$ Reynolds' analogy is biased towards reducing the intensity of heat transfer.

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