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Issues of Increasing the Efficiency of Solar Air Heaters and Methods for Calculating Heat Transfer on Solar Receivers with a Discontinuous Boundary Layer

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Abstract: In the article we will consider the issues of increasing the thermal efficiency of solar air heaters by artificially interrupting the hydrodynamic boundary layer in the heater channels. Intensification models and formulas for calculating heat transfer are proposed.

Keywords: The heater of sunny air, effectiveness of heat hydrodynamics limited sphere, fasting of exchanging heat.

Introduction

Solar air heaters (SAH) are one of the promising solar-technical devices that can be used in air heating systems and solar drying devices. Such SAH are divided into two types:

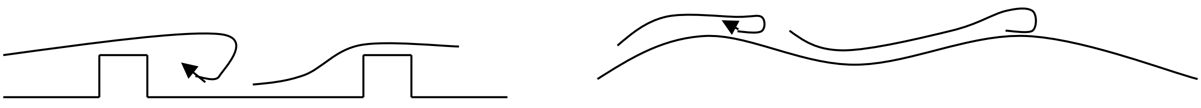
- SAHs operating under the influence of forced convection
- SAHs operating under natural convection

The second type of heating device is often used when the thermal output of the dryer is low or when heating air in rooms of small cubic capacity.

Considering that such heaters use air as a coolant, which has a small heat capacity (about 4 times less than the heat capacity of water), then these SAH s have not only increased dimensions and weight, but also low thermal efficiency. Therefore, to reduce these indicators and increase thermal efficiency in the SAH, methods must be implemented to increase heat transfer in the channels of the heating device.

To apply these methods, it is considered advisable to use methods known in industrial heating engineering to intensify heat transfer [1-5].

The most well-known methods of intensifying heat transfer processes include: periodic interruption of the hydrodynamic boundary layer formed on the solar receiving surface. In practice, such artificial updating of the boundary layer can be realized using the surfaces shown in Fig. 1.



a б

Fig. 1 Scheme of periodic renewal of the hydrodynamic boundary layer on the solar receiving surface: a – channel with roughness protrusions, b – wavy channel.

It should be noted that the near-wall flow on such surfaces is complex and currently not amenable to a theoretical solution. In this regard, semi-empirical methods for calculating heat transfer and hydraulic resistance of such a flow are considered acceptable.

To simulate the wall flow (Fig. 1a), we assume the following provisions:

1. Separated zones and reverse flows formed behind the turbulators are considered to be gradient-free, such as a boundary layer.
2. We will consider the resulting vortices in corner zones only to be sources of turbulence and quasi-stationary. Thus, the flow model can be formed as follows: model № 1.

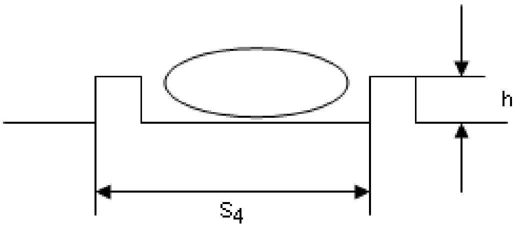


Figure 2. Formation of a single vortex

In this case, the resulting vortex and the vortex zone between the turbulators create a certain layer through which the particles of the main flow find it difficult to penetrate and impede heat transfer. Hydraulic resistance obeys the quadratic regime, i.e. a mode in which the coefficient of hydraulic resistance does not depend on the Reynolds number. It is not advisable to install such close proximity of turbulators in the air flow.

Model № 2.

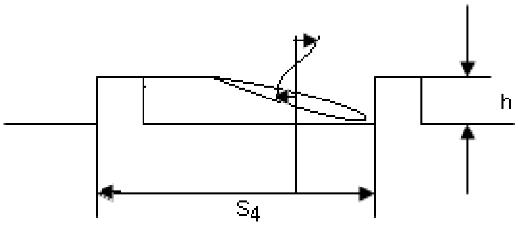


Figure 3. Reverse flows of the boundary layer type.

This type of flow can be represented as a reverse flow of the boundary layer type developing under the influence of a quasi-solid vortex. Model № 3

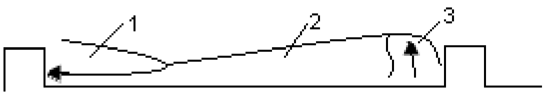


Figure 4. A chain of boundary layers formed between two turbulators

1, 2, 3 – differently directed boundary layers:

1– the boundary layer is formed in the opposite direction to the main flow after the point of reattachment;

2– the boundary layer is formed after the reattachment point in the direction of the main flow;

3– the boundary layer is formed due to a vortex arising on the front wall of the rear turbulator.

This case is optimal from the point of view of intensifying heat transfer due to the fact that throughout the entire period (the period corresponds to the distance between two protrusions) several flows are formed, which can be represented as a model.

To obtain engineering formulas for calculating heat transfer, we also accept that:

- ➢ The flow pattern between turbulators repeats from period to period [6];
- ➢ Flows in reverse directions develop under the influence of quasi-solid vortices and thus the speed in the reverse boundary layer is equal to the speed of the main flow.

For model № 1, a boundary layer type flow is not observed and, therefore, the heat exchange process between the wall and the main flow is carried out in a similar way to a rough wall.

For model № 2, wall flows are formed in the opposite direction and, to a first approximation, such flows can be expressed by the formulas of a turbulent boundary layer.

The average heat transfer can be calculated in accordance with the accepted assumptions as follows:

$$Nu_d = 0{,}0225\mathrm{Re}_d^{0{,}8}\left(\frac{S_L}{d}\right)^{-0{,}2} \qquad (1)$$

For model № 3, the formula for average heat transfer can be obtained using the superposition method;

$$\overline{Nu}_l = \frac{\overline{Nue}_1 l_1 + \overline{Nue}_2 l_2 + \overline{Nue}_3 l_3}{l_1 + l_2 + l_3} \qquad (2)$$

The length of the first layer for $S_4/h=12$ (optimal step) $L_1 = 5h$ the length of the third is $l_3 = l` = 1{,}5h$.

The length of the second layer is calculated as

$$S_4 - \ell_1 - \ell^1 = 12h - 5h - 1{,}5h = 5{,}5h$$

Substitute into formula (2) we get:

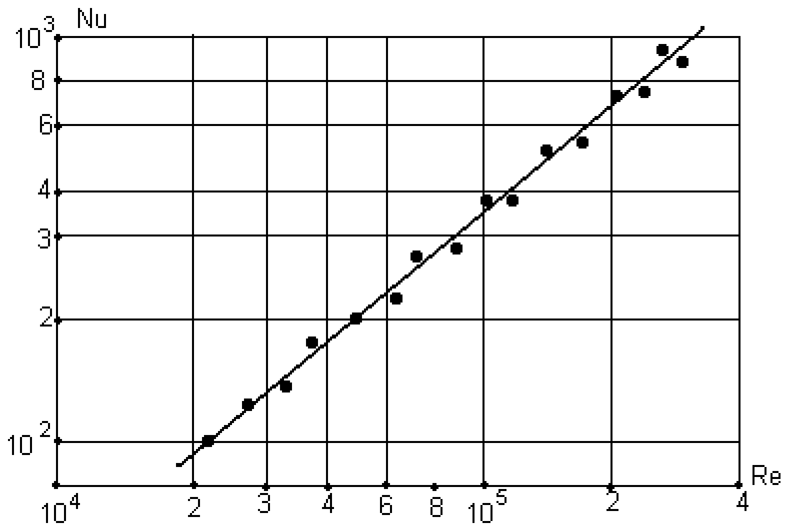
$$Nu_d = 0{,}0225\mathrm{Re}_d^{0{,}8}\frac{\left(\frac{l_1}{d}\right)^{-0{,}2} l_1 + \left(\frac{l_2}{d}\right)^{-0{,}2} l_2 + \left(\frac{l_3}{d}\right)^{-0{,}2} l_3}{l_1 + l_2 + l_3} = 0{,}0225\mathrm{Re}_d^{0{,}8}\frac{\left(\frac{5h}{d}\right)^{-0{,}2} 5h + \left(\frac{5{,}5h}{d}\right)^{-0{,}2} 5{,}5h + \left(\frac{1{,}5h}{d}\right)^{-0{,}2} 1{,}5h}{12h}$$

$$= 0{,}0225\mathrm{Re}_d^{0{,}8} * 0{,}74\left(\frac{h}{d}\right)^{-0{,}2} == 0{,}017\mathrm{Re}_d^{0{,}8}\left(\frac{h}{d}\right)^{-0{,}2} \qquad (3)$$

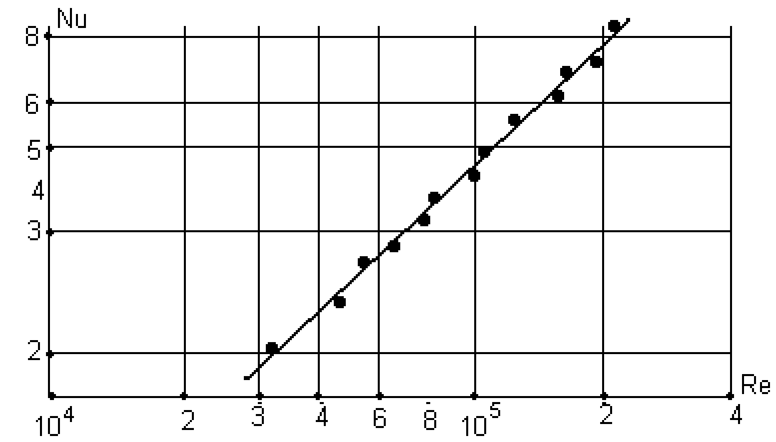
The calculation method according to the proposed formula (3) is that, first of all, the distance between the turbulators is determined; if $S_4/h<12$, then the heat transfer calculation is carried out in accordance with formula (1).

When Su/h>10, heat transfer is calculated using formula (3).

Figure 2 shows calculations using formulas (1), (2) and experimental data [1]. The graph shows their satisfactory agreement.



a) t/D=0,5; d/D=0,967



б) t/D=0,25; d/D=0,905

Figure 2. Calculation of heat transfer in a channel with protrusions. t – distance between protrusions; D – channel diameter.

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