THE PROBLEM OF THE EFFECT THAT THE TURBULIZER CONFIGURATION EXERTS ON THE THERMAL EFFICIENCY OF A CHANNEL WALL SURFACE

V. G. Pavlovskii UDC 536.244

We discuss the results from an experimental comparison of the thermal effectiveness of the heating surface in a flat channel with turbulizers of various configurations positioned on that wall.

The imposition of artificial roughness on the heat-exchanger surface to increase the heat-transfer coefficient results in an intensive increase in the hydraulic resistance, and the thermal effectiveness of the heating surface is determined by the ratio between the quantity of transmitted heat and the expenditures of energy on resistance within the channel. As this criterion we can use the energy coefficient, provided that the comparison of the thermal effectiveness of the rough surface is accomplished for the case in which $\Delta t = 1°C$ and the expenditures of power in the channel are referred to a unit of the heating surface [1]:

$$E_0 = \frac{\bar{\alpha}}{9.807\,\mathrm{Eu}_f \rho u_0^3}\,\frac{F}{f}. \qquad (1)$$

However, the data existing in the literature with respect to the effect of the configurations of roughness protuberances are most frequently assoicated with a rise in the heat-transfer coefficient and have virtually no effect on the thermal effectiveness of the heating surface. This applies particularly to two-dimensional roughness protuberances on heating surfaces positioned at equal intervals along the channel length across a flow of a cooling liquid, and these protuberances are referred to in the literature as "turbulizers" (spoilers of the wall layer). In this case, the configuration of the turbulizers exerts various forms of influence of the intensity with which the coefficients of heat transfer and hydraulic resistance increase [2-6].

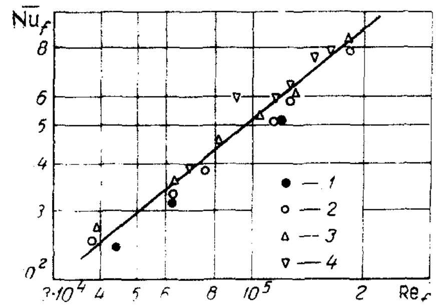

Fig. 1

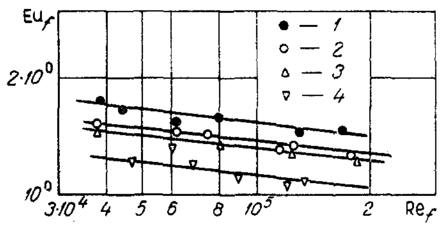

Fig. 2

Fig. 1. Determination of the correlation $\overline{\mathrm{Nu}}_f = f(\mathrm{Re}_f)$ for the wall of a flat channel with turbulizers of various configurations: 1) triangular; 2) semicircular; 3) rectangular; 4) dropshaped profile.

Fig. 2. Effect of turbulizer configuration on the resistance of a flat channel: 1-4) see Fig. 1.

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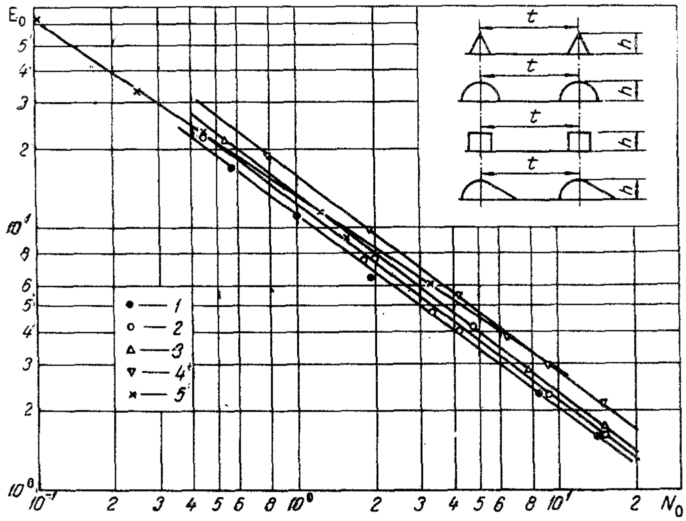

Fig. 3. Comparison of the thermal effectiveness of a channel surface with turbulizers of various configurations, based on the energy coefficients: 1-4) see Fig. 1; 5) results for a smooth channel.

The explanation should be sought in the physical concept of the hydrodynamic conditions of streamlining of the roughness protuberances on the channel walls by the cooling liquid. Thus, for example, the increase in heat transfer is associated with a change in the structure of the flow behind the turbulizer, when the energy redistribution in the case of separation streamlining exert a favorable influence on the thermal characteristics of the convection heating surface. The decisive characteristic of such a process is the turbulizer height h, and its efficiency depends on the condition $h > \delta$ and the interval t.

At the same time, the increase in the resistance is described not only by the quantity h, but by the turbulizer configurations as well. This is explained by the fact that the channel containing turbulizers at the heating surface exhibits a resistance that is composed of the friction within the very section of the channel wall and the local turbulizer resistance within that section, and this in turn is expressed in terms of the coefficient of frontal resistance.

Consequently, the turbulizer configuration may exert considerable influence on the increase in resistance within the channel. Here we should expect that the least resistance and the greatest thermal effectiveness on the part of the channel surface will be found when turbulizers are in operation, provided that their hydrodynamic profile has the minimum coefficient of frontal resistance.

Studies of a thermal model of a flat channel with a length of l = 420 mm and a height of 2s = 30 mm – on whose horizontal electrically heated walls were mounted turbulizers of rectangular, triangular, semicircular, and dropshaped profile (see Fig. 3) – were devoted to clarifying the effect of turbulizer configuration on the thermal effectiveness of the heating surface and the possibility of expressing this quantity in terms of the frontal-resistance coefficient. The geometric criteria of comparable turbulizers were kept constant during this study and equal to $h/t = 0.054$ and $h/s = 0.29$. The decisive quantity in the comparison of the experimental data was therefore the turbulizer configuration.

The coefficient for the transfer of heat at the channel wall was determined by the steady-state heat-flow method

$$\alpha(x) = \frac{Q}{\Delta t(x)\,\varphi F}. \qquad (2)$$

Integration of local heat-transfer coefficients yielded the average $\overline{Nu}_f$ whose correlation with the quantity Re_f is shown in Fig. 1. In similarity criteria we have chosen the channel length as the determining dimension, while the arithmetic mean between the sum of the average magnitudes of the wall temperature and the temperature of the flow over the channel length has been taken as the determining temperature. The effect of turbulizer configuration has little effect on the intensity of heat transfer in the channel. All of the experimental data are grouped about the curve $\overline{Nu}_f = f(Re_f)$, in whose equation the exponent with the Reynolds number is equal to 0.8, while the proportionality factor is equal to 0.05. At the same time, for the hydraulic resistance in the channel (Fig. 2) the effect of configuration is quite noticeable, and each turbulizer configuration corresponds to a unique magnitude of this resistance, determined from the following relationship:

$$\mathrm{Eu}_f = f(\mathrm{Re}_f;\ C_{\text{front}}). \tag{3}$$

Here $C_{\text{front}} = f(\mathrm{Re}'_f)$ is the coefficient of frontal resistance for the turbulizer, and $\mathrm{Re}'_f = u'h/\nu_f$ [7].

The velocity u' at the widest point of the turbulizer midsection, equal to the height h, is determined with sufficient accuracy [5] from the expression

$$\frac{u'}{u^*} = 2.5 \ln\left(\frac{h}{h_s}\right) + 8.48\ldots, \tag{4}$$

where for air the dynamic velocity $u^* = u_0 \cdot 0.7\psi^{0.5}$. The equivalent sand roughness h_s in (4) is defined [8] as follows:

$$\lg\left(\frac{s}{h_s}\right) = \frac{\psi^{-0.5} - 4.24}{4.07}. \tag{5}$$

Subsequently, for each turbulizer configuration and for each corresponding value of Re' we find from the $C_{\text{front}} = f(\mathrm{Re}')$ diagram [7] the quantity C_{front} for which the rectangular, triangular, semicircular, and dropshaped profiles are, respectively, equal to 0.9, 1.5, 0.99, and 0.2.

Consequently, we find the relationship between the quantities (3) from the following criterial equation:

$$\mathrm{Eu}_f = 3.12\, C_{\text{front}}^{0.2} \mathrm{Re}_f^{-0.1}. \tag{6}$$

Comparison of the heating surface in terms of the energy coefficient with consideration of the increase in the heating surface both for h/t = idem and h/s = idem is shown in Fig. 3, where we see that the effect of the heating surface, as well as the resistance within the channel, diminishes with an increase in the magnitude of the frontal resistance. The best heating surface in terms of the energy coefficient was achieved in this case for a channel with turbulizers of a dropshaped profile, which is characterized by the minimum coefficient of frontal resistance from among all the turbulizer configurations examined. If we neglect the insignificant change in the slope of the straight lines in Fig. 3 and if we adopt the average value, equal to 0.8, all of the results can be generalized by the following equation:

$$E_0 = 11.9\, C_{\text{front}}^{-0.2} N_0^{-0.8}. \tag{7}$$

The equations derived here, i.e., (6) and (7), for certain factors which affect the thermal effectiveness of the channel heating surface and are associated with the geometric roughness criteria $K_m = h/t$ and $K_s = h/s$, make it possible to find the most efficient turbulizer configuration. Here we determine the minimum loss to resistance to within the channel and the greatest thermal effectiveness of the heating surface.

NOTATION

u_0	is the inlet flow velocity, m/sec;
f	is the lateral cross-sectional area of the channel, m^2;
F	is the heating surface, m^2;
l	is the channel length, m;
φ	denotes the relationship between the rough and the smooth surfaces;
$\Delta t(x) = t_w(x) - \bar{t}_f(x)$	is the local temperature head, °C;
$t_w(x)$ and $\bar{t}_f(x)$	denote the wall temperature and the average temperature of the air flow through the channel cross section at a distance x from the channel inlet, °C;
N_0	is the power expended on moving the air through the channel, $W/m^2 \cdot h$;
s	is the half the channel height, m;
ψ	is the coefficient of frictional resistance for the heating surface;
δ	is the thickness of the hydrodynamic boundary layer, m;
t	is the interval between the turbulizers at the heating surface, m.

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