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ENHANCEMENT OF HEAT AND MASS TRANSFER IN A PLANE CHANNEL WITH TURBULIZERS ON ONE OF THE WALLS

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The effect of turbulizing asperities positioned on one of the walls of a plane channel on the intensity of exchange processes near a smooth wall with injection of different working media is studied experimentally.

In a number of practical problems one must enhance heat and mass transfer processes on smooth, inaccessible fins of moving or fixed surfaces, e.g., in heating-cooling or chemical treatment of surfaces of shafts, strips, ropes, etc.

To achieve this task one can use active means (an increase in the velocity of the washing flow, jet input of the working medium, acoustic effects, etc.) associated with the supply of additional energy, and passive means – without energy supply – by selecting the shape of the surfaces of the channel in which heat and mass transfer processes occur.

In what follows we consider problems of enhancement of heat and mass transfer processes in a plane channel on smooth moving or motionless walls by positioning turbulizing asperities on an opposite wall with jet input of the working medium into the channel.

In this case, enhancement is complicated by the need to transfer turbulent vortices generated by the turbulizers to the smooth wall through the main flow and transverse to it.

Vortices generated by asperities affect heat and mass transfer in the boundary layer behind the asperity as turbulence of an external flow. It is very difficult to obtain the dependence of the rate of heat and mass transfer on turbulence of an external flow, since the parameters characterizing transfer of momentum, heat, and convective diffusion are in the boundary layer in a certain correlation and are found depending on the boundary conditions and the intensity of the processes of generation of turbulent energy and its dissipation in the main flow.

Therefore, in designing heat and mass transfer apparatuses use is made of the results of experimental determination of the effect of design and flow parameters on the required characteristics.

A large number of publications dealing with enhancement of exchange processes in channels by mounting turbulizing asperities on heat-transfer walls are known (see, e.g., [1] and references therein).

A limited number of works [1-4] are devoted to the study of these processes on smooth moving or motionless walls with finning on an opposite wall that provides additional flow turbulization. A small height of turbulizing asperities, which can be treated as artificial roughness, is a specific feature of these studies.

It is known [5] that in flow in a long plane channel with smooth walls and smooth flow input the level of turbulent pulsations is of from 2.5 to 9%.

For the case of smooth input of an artificially turbulized flow in a short plane channel, the initial degree of turbulence attains 30-50% [6]. An empirical dependence of the efficiency of heat transfer on the degree of flow turbulence is obtained in [6] in the form

$$\mathrm{Nu}_x = 0.132\,\mathrm{Re}_x^{0.83}\,\mathrm{Pr}^{0.4}\,\varepsilon^{0.3}\,, \tag{1}$$

where Nu_x, Re_x, Pr are the Nusselt, Reynolds and Prandtl numbers, respectively; ε is the degree of turbulence, %.

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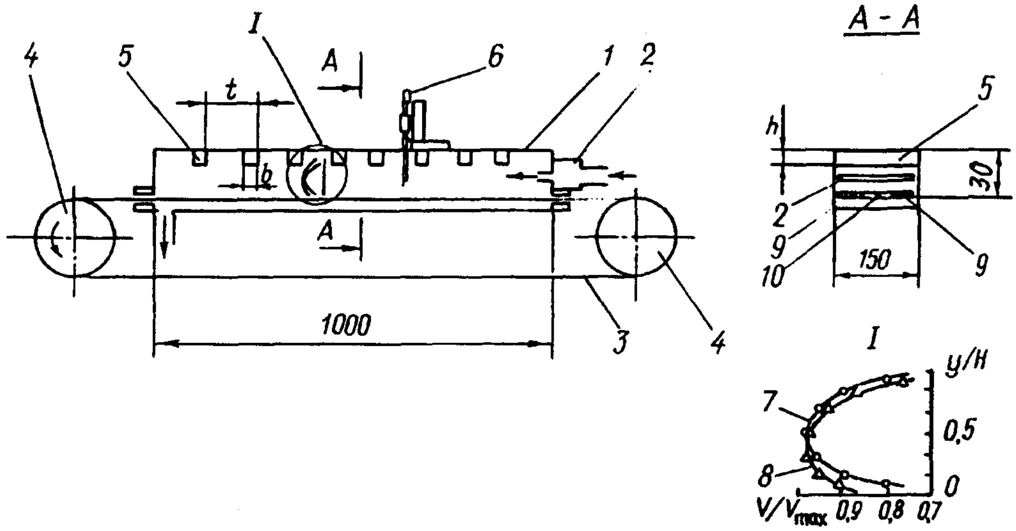

Fig. 1. Schematic diagram of experimental setup: 1) channel body, 2) slot-like nozzle, 3) channel movable wall, 4) drive drums, 5) turbulizers, 6) thermoanemometer pickup, 7, 8) characteristic velocity profiles in smooth channel and in channel with turbulizers, respectively, 9) rubber strips, 10) washed coating.

It is found that the value of ε along the channel length sharply decreases and at a distance of about 10–20 channel heights it reaches the value typical for a stabilized flow in a channel (ε = 5–10%), which is in correspondence with the data of [5].

Enhancement of exchange processes in a plane channel in a stabilized portion caused by installing a set of round rods transversely to the flow on one of the walls is considered in [4]. The results of experiments obtained by the method of naphthalene sublimation on finned and smooth walls indicate enhancement of exchange processes on both walls due to flow turbulization caused by a set of rods. In particular, the Sherwood number (the mass transfer analog of the Nusselt number) on a smooth wall in the presence of finning on an opposite wall increases compared to smooth channel walls by 1.3–1.8 times, and on a finned wall by 1.6– 2.4 times. If we assume that formula (1) is valid for the case of a channel with one finned wall, then such increase in the Sherwood number indicates an increase in the intensity of turbulent pulsations by 3–8 times, i.e., to values of the order of 30–70%.

The presence of jet input of a highly turbulent flow into the channel and its interaction with turbulizing asperities is a specific feature of the considered problem.

The studies were conducted on an experimental setup with a 30 × 150-mm channel having a length of 1000 mm. A schematic diagram of the setup is presented in Fig. 1. The working media were air at mean velocities of from 1 to 2 m/sec and water at a mean velocity to 0.3 m/sec.

The experiments with air were conducted on a motionless smooth wall, and with water, on a wall moving at a velocity of up to 2 m/sec. Air or water were supplied to the channel through a slot-like nozzle with a width of 3 mm located in the middle portion between the walls at the start of the channel, thus causing a high initial level of turbulence. Turbulizers in the form of rectangular changeable transverse cover-plates with a height of from 6 to 18 mm, a width of 9 mm, and a spacing of from 20 to 100 mm were positioned on one of the channel walls.

Turbulence levels and profiles of averaged velocities were determined by constant-temperature thermoanemometers using one-filament pickups with a tungsten filament. To reduce the screening effect of the wall on the readings of the thermoanemometer pickups, measurements were made at a distance smaller than 4–5 mm from the surface of the smooth moving or fixed wall [7].

The installation of turbulizers results in filling of the velocity profile in the flow cross-section and in an increase in relative averaged velocities near the smooth wall by 5–10%; the highest increment was observed at spacing t = 40 mm, which is typical for cross-sections above the asperities and between them. A similar change in the velocity profile was noted in [5].

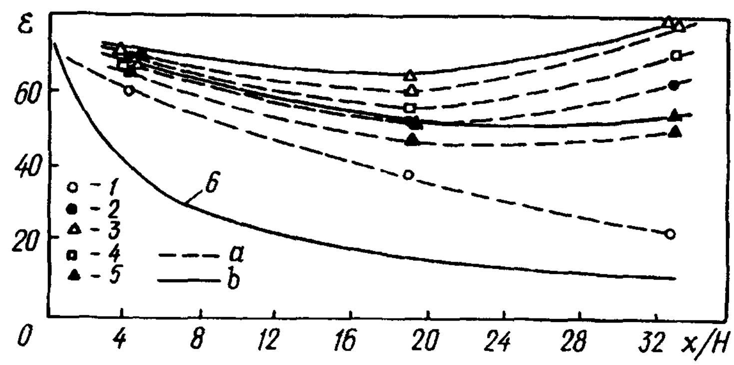

Fig. 2. Change in intensity of turbulence along channel length ($Re_d = 1.4 \cdot 10^4$, $h = 6$ mm): 1) without turbulizers, 2) $t = 20$ mm, 3) 40, 4) 60, 5) 80, 6) calculation by [6]; a) working medium-air; b) same – water. ε, %.

As the data of experiments given in Fig. 2 show, in the absence of turbulizers the level of turbulent pulsations decreases along the channel length. This figure also presents the results of calculation of the decrease in the turbulence level obtained by a formula suggested in [6]:

$$\varepsilon = \varepsilon_0 \left[\frac{x/H + \varepsilon_0}{\varepsilon_0}\right]^{-n}, \tag{2}$$

where ε_0 is the level of turbulence at the channel inlet; H is the channel height; x is the coordinate along the channel length; $n = 0.8$ (for $\varepsilon_0 = 60$–80%).

A substantial difference between our experimental data and the calculation curve indicates that jet input of the working medium into the channel leads to a slower dissipation of turbulent pulsations along the channel. With an increase in the relative width of the nozzle from 0.1 to 1.0 (a smooth input of the flow) in a channel without turbulizers the intensity of turbulent pulsations will change within the range bounded by curves 1 and 6 (Fig. 2).

To estimate the level of turbulence along the length of a plane channel of this setup with an initial turbulence of the flow of the order of 60–90% we can suggest the formula obtained based on the processing of empirical data

$$\varepsilon = \varepsilon_0 \exp\left[\sigma\left(1 - \frac{x/H + \varepsilon_0}{\varepsilon_0}\right)\right], \tag{3}$$

where $\sigma = 0.12$.

This figure also presents the results of experiments with turbulizers in the form of rectangular asperities of a height h positioned on one of the walls with spacing t. These data show that the installation of turbulizers allows one to compensate for the decrease in the level of turbulent pulsations along the channel length and in some cases to reach, by the end of the channel, values exceeding their initial level. This effect takes place for both working media on both moving and motionless smooth walls.

To find the optimum ratio of the geometric dimensions of turbulizing asperities and their position, at which the level of turbulence near a smooth wall reaches a maximum, the extremum value of the turbulence level was determined experimentally by successively studying the feedback surface in the factor space and advancing to an extremum [8]. The following geometric dimensions of turbulizing asperities are the factors of the experiment: asperity width b; asperity height h, distance between asperities t.

As follows from the literature on intensifying heat exchanging devices of this type [1, 9, 10] and the results of preliminary tests, the width of the asperities b, the effect of which within the limits of the experiment is slight, is considered an unimportant factor.

The studies were conducted according to the plan of a factor experiment realizing all possible nonrecurrent combinations of the levels of independent variables, each of which had to be varied on five levels; air was used as the working medium.

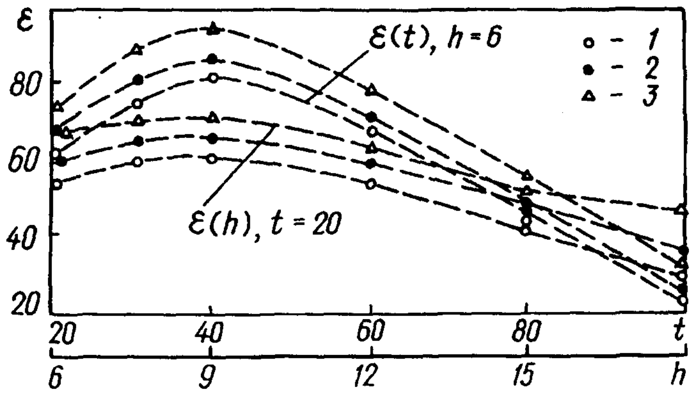

Fig. 3. Dependence of turbulence intensity $\varepsilon = \varepsilon(t, h)$ in section $x/h = 32$ for different combinations of parameters t and h: 1) $\mathrm{Re_d} = 0.8 \cdot 10^4$, 2) $1.2 \cdot 10^4$, 3) $1.4 \cdot 10^4$. ε, %; t, mm; h, mm.

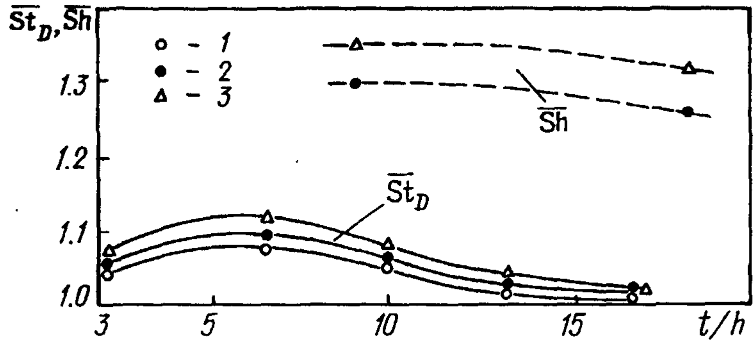

Fig. 4. Change in mass transfer mean along channel length versus t/h at different Reynolds numbers: 1) $\mathrm{Re_d} = 0.8 \cdot 10^4$, 2) $1.2 \cdot 10^4$, 3) $1.4 \cdot 10^4$.

Figure 3 presents the range of ε variation, depending on h and t, which has a clearly expressed extremum in the factor field of the experiments for $30 < t < 50$ mm and $7.5 < h < 10.5$ mm for Reynolds numbers of from $0.8 \cdot 10^4$ to $1.4 \cdot 10^4$.

In the experiments with water the rate of the entrainment of the mass of the washed coating from a smooth moving channel wall was studied. A solution of soap in freon applied between two rubber strips glued to a smooth moving belt along its length was used as the washed coating. Coating entrainment was estimated with respect to time of complete coating washing away under different effects on the diffusion process, which is in correspondence to the change in the Stanton diffusion criterion, which in the experiments was determined by the formula

$$\mathrm{St}_D = \frac{\Delta M\,(\rho_c + \rho)}{\Delta\tau\, F V \rho_c \rho},$$

where ΔM is the entrainment of the mass of the washed coating; ρ_c is the coating density; ρ is the liquid (water) density; $\Delta\tau$ is the time of mass entrainment ΔM; F is the area of the washed coating in the channel; V is the liquid velocity in the channel with respect to the coating.

Figure 4 shows $\overline{\mathrm{St}}_D - \mathrm{St}_D/\mathrm{St}_{D0}$ versus t/h for different Reynolds numbers of the flow in the channel (here and in what follows the subscript 0 corresponds to turbulent flow in a smooth channel). The results presented indicate that at $t/h \approx 4-8$ the effect of turbulizers on the increase in the rate of mass transfer on a smooth channel flow is maximum.

To compare the results obtained with the data of [4] we take into account the relationship between the diffusion Stanton number (St_D) and the Sherwood (Sh), Reynolds (Re), and Schmidt (Sc) numbers [11]

$$\mathrm{St}_D = \frac{\mathrm{Sh}}{\mathrm{Re\,Sc}},$$

from which, assuming $Sc_0 = Sc \approx 1$ and $Re = Re_0$, we obtain

$$\frac{St_D}{St_{D0}} \approx \frac{Sh}{Sh_0}.$$

Proceeding from the above the comparison of the data of [4] ($\overline{Sh} = Sh/Sh_0$) with the results of our work ($\overline{St}_D$) is justified (see Fig. 3).

The considerably lower increase in the mentioned ratios obtained by the authors is explained by the high mean level of turbulent pulsations in a smooth channel, which is caused by jet input of the flow ($\varepsilon_0 = 40-50\%$) as compared to the values of pulsations of the order of 5–10% with smooth input of the working medium into the channel, as in [4].

Thus, we obtained experimental data which make it possible to choose the geometry of a system of turbulizers mounted on one of the walls of a plane channel that ensures conservation and an increase in the level of turbulence along the channel length with jet input of a highly turbulent flow of the working medium into the channel. It is shown that these relations are valid for fixed and moving smooth walls. They can be used to enhance heat and mass transfer processes on the surfaces of fins that are inaccessivble for engineering reasons.

NOTATION

$Nu = \alpha L/\lambda$, Nusselt number; α, coefficient of heat transfer; L, characteristic dimension (hydraulic diameter, d; coordinate, x); H, channel height; h, height of turbulizing asperities; y, coordinate over channel height; λ, coefficient of thermal conductivity; $Re = LU\rho/\mu$, Reynolds number; U, flow velocity; ρ, medium density; μ, coefficient of dynamic viscosity; $Pr = \mu/(\rho a)$, Prandtl number; a, coefficient of thermal diffusivity; $\varepsilon = \sqrt{(U')^2}/U \cdot 100\%$, degree of turbulence; U', flow velocity fluctuation; $Sc = \mu/(\rho D)$, Schmidt number; D, coefficient of molecular diffusion; $St_D = k/U$, Stanton number; k, coefficient of mass transfer; $Sh = Lk/D$, Sherwood number.

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