INVESTIGATION OF LOCAL HEAT-TRANSFER COEFFICIENTS UNDER CONDITIONS OF THE RESONANCE OSCILLATIONS OF A GAS IN CHANNELS

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A physical model of the process of heat transfer during resonance oscillation of a gas in channels is proposed. The results of experiments are generalized by a criterial dependence.

We have made an attempt to generalize experimental data on the heat-transfer coefficient under conditions of resonance oscillations for gas flow in channels. The experiments were conducted in cylindrical channels with diameters $d_0 = 12$ and 19.4 mm and lengths $L = 2.045$ and 2.337 m. The tests were conducted in the following range of the basic parameters: flow temperature, $T_f = 350$–700°K; temperature of the channel wall, $T_W = 500$–800°K; temperature factor, $T_W/T_f = 1.1$–1.5; Reynolds number, $Re_0 = 10^4$–10^5; pressure in the channel, $P_0 = 3.5$–20 bar; frequency of oscillations, $f = 70$–1000 Hz; and amplitude of pressure oscillations, $\Delta P = 0.1$–3 bar. A detailed description of the experimental equipment and the procedure of the experimental investigations are given in [5, 3]. The following physical model was taken as the basis for generalizing the experimental data. Since in the stationary turbulent flow of a fluid the heat transfer is proportional to the flow velocity and the thickness of the viscous layer, under conditions of an oscillating flow the relative heat-transfer coefficient $K = Nu/Nu_0$ must depend on the relative amplitude of oscillations of the bulk velocity $\Delta(\rho u)_0/(\rho u)_0$ and on the relationship between the thicknesses of the stationary viscous layer δ_0 and the oscillating layer. The thickness of the oscillating layer $\delta_{OS} = \sqrt{2\nu/\omega}$ characterizes the gradient of the amplitude of oscillation of the velocity close to the surface; the thickness of the stationary viscous sublayer δ_0 characterizes the velocity gradient close to the surface in a stationary flow of the fluid.

According to semiempirical theories, the thickness of the viscous sublayer in the stationary turbulent flow regime is given by the formula

$$\frac{\delta_0 \sqrt{\tau_w/\rho_0}}{\nu} = 10. \tag{1}$$

The tangential stress at the wall of a cylindrical channel in the stationary flow is given by

$$\frac{\tau_w}{\rho_0} = \xi \frac{u_0^2}{8}, \tag{2}$$

where the frictional drag coefficient for $10^4 \le Re_0 \le 10^5$ is given by

$$\xi = 0.3164 Re_0^{-0.25}.$$

Using this relation we obtain

$$\frac{\delta_{os}}{\delta_0} = 0.1 \sqrt{2 \frac{\tau_w}{\rho_0} \frac{1}{\omega\nu}} = 0.0282 \sqrt{\frac{Re_0^{1.75}}{Re_\omega}}. \tag{3}$$

According to the above physical model, the criterial equation for the relative heat-transfer coefficient can be written in the form

$$K = \frac{Nu}{Nu_0} = F\left[\frac{\delta_{os}}{\delta_0}, \frac{\Delta(\rho u)_0}{(\rho u)_0}\right] = F\left[Re_0, Re_\omega, \frac{\Delta(\rho u)_0}{(\rho u)_0}\right]. \tag{4}$$

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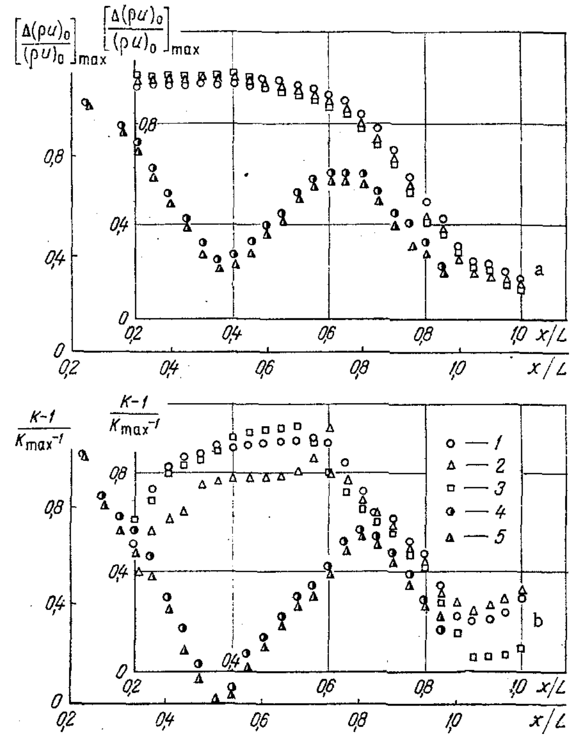

Fig. 1. Distribution of relative amplitudes of oscillations of the bulk velocity (a) and heat-transfer coefficient (b) along the length of the channel. First resonance, f = 77 Hz: 1) $P_0 = 1.5$ kg/cm^2; 2) 4.25; 3) 15; second resonance, f = 152 Hz: 4) $P_0 = 1.75$; 5) 4.25.

Thus, the problem of experimental investigation of the relative heat-transfer coefficient reduces to the determination of the functional dependence (4). For computing the distribution of the relative amplitude of oscillation of the bulk velocity along the length of the channel in a nonisothermal gas flow we made use of the technique proposed in [5].

A typical variation of the relative amplitude of oscillation of the bulk velocity along the length of a channel with diameter $d_0 = 19.4$ mm and close to the first and second resonances is shown in Fig. 1a.

As follows from this graph, the shape of the standing wave is distorted due to the nonisothermal nature of the flow and the friction; there are no well-defined nodes of the velocity of the standing wave and the amplitude decreases along the length of the channel.

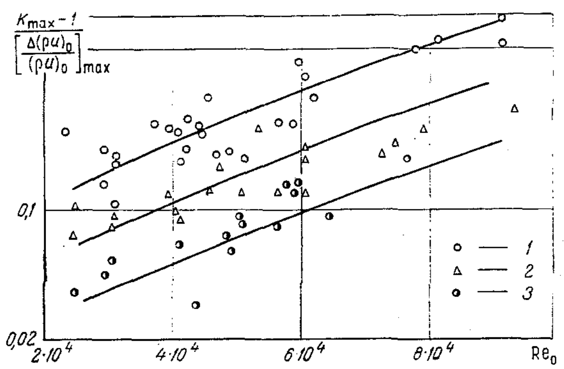

Fig. 2. The relative heat-transfer coefficient as a function of Reynolds number: 1) first resonance; 2) second resonance; 3) third resonance.

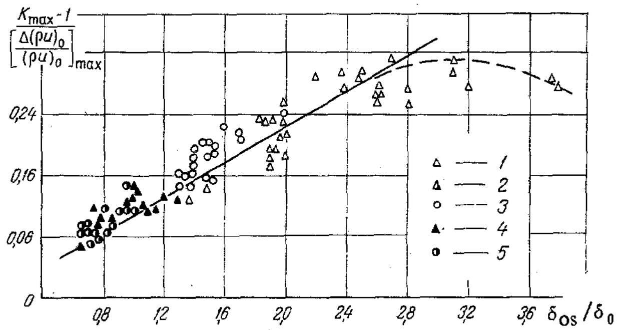

Fig. 3. Generalized dependence of the maximum heat-transfer coefficient on the ratio of the oscillating and stationary layers. First segment: 1) first resonance; 2) second resonance; second segment: 3) first resonance; 4) second resonance; 5) third resonance.

The distribution of the local heat transfer along the length of the channel near the first and the second resonance harmonics in a channel with diameter $d_0 = 19.4$ mm is shown in Fig. 1b in relative coordinates

$$\frac{K-1}{K_{max}-1} = F\left(\frac{x}{L}\right).$$

The form of the distribution of heat transfer along the length of the channel corresponds to the nature of the distribution of the relative amplitude of oscillation of the bulk velocity. The relative heat-transfer coefficient along the length of the standing wave changes in proportion to the relative amplitude of oscillation of the bulk velocity according to a law which is close to linear:

$$\frac{K-1}{K_{max}-1} = \frac{\Delta(\rho u)_0}{[\Delta(\rho u)_0]_{max}}. \quad (5)$$

Thus, the experimental dependence of the distribution of heat transfer along the length of the standing wave shown in this figure can be regarded as universal. Therefore, for generalizing the experimental data it is sufficient to determine the criterial dependence for the maximum of the heat transfer (heat transfer at the antinode of the velocity of the standing wave). An experimental investigation of this dependence is the most difficult stage of the investigation, since, as a rule, all three criteria of similarity Re_0, Re_ω, and $\Delta(\rho u)_0/(\rho u)_0$ are interrelated in the experimental equipment and a sufficiently large number of tests are required in order to separate out the effect of each individually.

A similar (close to linear) law of variation of the relative heat transfer as a function of the relative amplitude of oscillation of the bulk velocity is observed at the antinode of the velocity of the standing wave.

The variation of the relative heat-transfer coefficient as a function of Reynolds number is shown in Fig. 2 for three frequencies (first, second, and third resonances). The heat-transfer coefficient increases with Reynolds number and the heat-transfer decreases with the increase of the oscillatory Reynolds number Re_ω, other conditions remaining unchanged. This nature of variation of heat transfer is explained by the variation of the quantity δ_{os}/δ_0 as a function of Re_0 and Re_ω. The analysis of the results of experiments on heat transfer at the velocity antinode of the standing wave is shown in Fig. 3 for channels with diameters $d_0 = 12$ and 19.4 mm in coordinates

$$\frac{K_{max}-1}{\left[\frac{\Delta(\rho u)_0}{(\rho u)_0}\right]_{max}} = F\left(\frac{\delta_{os}}{\delta_0}\right).$$

As follows from this graph, the results of the experiments can be satisfactorily generalized with respect to the parameter δ_{os}/δ_0. The heat transfer increases with parameter δ_{os}/δ_0 and attains a maximum for $\delta_{os} \cong 3\delta_0$; after this, for $\delta_{os} > 3\delta_0$ some decrease of heat transfer is observed.

The results obtained above can be interpreted in the following way. The maximum effect of oscillations on heat transfer is observed for such ratios of the frequency of oscillations and Reynolds number for which the thickness of the oscillating layer δ_{os} is comparable with the thickness of the stationary transitional (buffer) layer, which is approximately equal to $3\delta_0$. Since the thickness of the oscillating layer characterizes the intensity of the secondary vortices close to the surface and the maximum generation of turbulent fluctuations is observed at the boundary of the transitional layer, for $\delta_{OS} = 3\delta_0$ the maximum energy transport by turbulent fluctuations to the surface occurs through the secondary vortices, which leads to an increase of heat transfer. In the range $\delta_{OS}/\delta_0 > 3$ a decrease of the heat transfer must occur with the increase of δ_{OS}/δ_0, i.e., the increase of Reynolds number and decrease of the frequency of oscillation must lead to a decrease of the effect of oscillating flow on heat transfer, which agrees with the available experimental data. However, this assumption requires additional experimental investigation in the range of low-frequency oscillations.

According to the investigations carried out here, the results of the tests on the maximum heat transfer for $\delta_{OS}/\delta_0 \leqslant 3$ can be generalized by the criterial equation

$$K_{\max} = 1 + 3{,}13 \cdot 10^{-3} \left[\frac{\mathrm{Re}_0^{1.75}}{\mathrm{Re}_\omega} \right]^{0.5} \left[\frac{\Delta (\rho u)_0}{(\rho u)_0} \right]_{\max}. \qquad (6)$$

NOTATION

d_0, channel diameter; L, channel length; ρ, density; u, velocity; P, pressure; T, temperature; f, frequency; ω, angular frequency; τ_W, tangential stress on channel wall; δ_{OS}, thickness of the oscillating layer; δ_0, thickness of the stationary viscous sublayer; ν, kinematic viscosity coefficient; Nu, Nusselt number; $K = \mathrm{Nu}/\mathrm{Nu}_0$, relative heat-transfer coefficient; $\mathrm{Re}_0 = u_0 d_0/\nu$, Reynolds number; $\mathrm{Re}_\omega = \omega d_0^2/\nu$, oscillating Reynolds number. Indices: 0, averaged parameters; Δ, fluctuation parameters.

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INVESTIGATION OF HEAT TRANSFER IN THE ENTRANCE REGION OF A FLAT PLATE PARALLEL TO THE FLOW WITH A LEADING EDGE IN THE FORM OF A ONE-SIDED WEDGE

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Relations are given for calculating the local heat transfer in the dynamic initial section of a longitudinally washed generator of a one-sided wedge for the case of constant wall temperature and turbulent boundary layer.

Leading edges with the profile of a sharp one-sided wedge, shown schematically in Fig. 1, are frequently encountered in the natural components of power machinery and in model experiments related to the study of heat transfer and flow over longitudinally washed surfaces [1, 2, 13]. When the angle β is quite small and the velocity field in the incident stream is uniform, it is usually assumed that mixed flow is generated in the boundary layer at the edge of a wedge oriented in the direction of the velocity vector W (the edge A in Fig. 1). However, it is known that leading edges with an angle near 90° cause flow separation [4, 10, 11].

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