HEAT TRANSFER DURING NONLINEAR OSCILLATIONS OF GAS IN A HALF-OPEN TUBE

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We present results of a study of heat transfer during nonlinear oscillations of a gas in a half-open tube.

In resonance oscillations in a half-open tube the amplitude of the velocity pulsations is determined by the amplitude of the displacement of the piston operating at the closed end, and by the conditions at the open end. The amplitude of the velocity pulsations at the open end of the tube can reach values of 100-150 m/sec, and the oscillation process is accompanied by various nonlinear effects, including the emission of shock waves [1-4]. One should also expect that the thermal situation will be different from that which prevails when pressure and velocity pulsations can be considered continuous functions of the time [5, 6].

We have attampted to study heat transfer in a half-open tube containing gas oscillating nonlinearly with the emission of shock waves. Longitudinal oscillations of the column of gas in the half-open tube were produced by the harmonic motion of a flat piston at the closed end. A compressor with a piston stroke $2l_0 = 0.086$ m and a diameter $2R_0 = 0.077$ m was used to maximize the amplitude of the oscillations. The crankshaft was driven by a dc motor. The tube had an inside diameter $2R = 0.04$ m and an overall length $L_0 = 5.485$ m. It consisted of five removable sections, which permitted experiments with systems of various lengths. The tube was connected to the compressor through a conical reducer of height $h = 0.1$ m.

The frequency of the oscillations was measured by passing light from an incandescent lamp through a hole in the rotating pulley and onto a photoresistor whose output signal was recorded on a ChZ-33 frequency meter.

The mean flow temperature was measured with a resistance thermometer, and the pressure by a type LKh-610 water-cooled piezoelectric transducer whose signal was fed into one of the inputs of an S1-18 double electron oscillograph. The pulsating velocity was measured with a constant temperature hot-wire anemometer by the method described in [1]. The signal from the hot-wire anemometer was fed into the second input of the oscillograph. The dynamic calibration of the wire was extended over the frequency range 0-40 Hz.

The oscillograms of the pressure and velocity oscillations were recorded by a photographic attachment. The oscillograms obtained are complex curves containing discontinuities. Since the hot-wire anemometer records the "rectified" signal, the velocity oscillations have a frequency twice that of the pressure pulsations. The velocity oscillograms were processed under the assumption that the mean flow velocity is zero. Therefore mirror reflection can restore the form of the velocity oscillations. The "range" — the difference between the maximum and minimum values of the pressure or velocity oscillations — was measured from the curves. For linear resonance half the range corresponds approximately to the amplitude of the oscillations. In the remaining cases it was assumed that the semirange was proportional to the amplitude.

Heat measurements were made with a probe (Fig. 1) whose working portion 1 consists of a section of tubing 0.1 m long with flanges 2 to help minimize heat fluxes in the axial direction. They also served to support the bracket 3 with the traversing device 4 and the velocity and temperature gauges 5.

The heating coil 6 was insulated from the housing by a thin layer of mica 7, and from the external medium by a layer of asbestos 8 flush with the flanges. The velocity and

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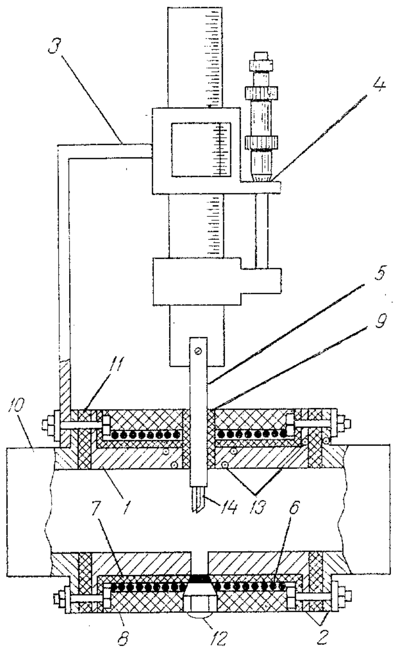

Fig. 1. Heat flux probe.

temperature gauges were insulated from the housing by Textolite 9. The working portion was attached to the main tube by adapters 10. Asbestos board disks 0.01 m thick were placed between the flanges of the adapter and the working portion. An opening was provided in the probe housing for the insertion of the velocity and temperature gauges 14, a thermocouple, and the pressure transducer assembly 12. Thermocouples 13 were mounted on the inner and outer walls of the housing of the working portion, on the outer surface of the asbestos layer, and on the flanges on both sides of the thermal insulation disks.

The heat-transfer coefficient was calculated from the formula [7]

$$\alpha = \frac{q_F}{T_c - T_\infty\left(1 + \psi\frac{\gamma - 1}{2}M^2\right)}. \tag{1}$$

For turbulent flow of diatomic gases in tubes $\psi = 0.85$. The quantity q_F was determined from the formula

$$q_F = \frac{IV}{2\pi R l_p} - \frac{\lambda_a \Delta t_a}{R\ln(R_a/R_1)} - \frac{\lambda_a (R_a^2 - R^2)\Delta t_d}{R l_p l_m}. \tag{2}$$

In (2) the loss of heat from the flanges to the surrounding medium and through the housing of the velocity and pressure gauges was not taken into account. The physical parameters were taken at the film temperature $(T_c + T_\infty)/2$. All the measurements were made at a constant temperature of the inner surface of the probe wall. The total error in the measurement of the Nusselt number did not exceed 12%.

The heat probe operates in the following way. The heat flux from the inner wall of the probe housing is transmitted to the oscillating air stream which is heated to a temperature T_∞, lower than T_c. Since the rest of the tube on both sides of the probe is at a still lower temperature T_0, there is an exchange of heat between them and the air stream also. As a result, for a certain relation among T_∞, T_c, and T_0, all the heat from the probe is completely removed by the rest of the tube. The opposite case when the tube itself is heated, and the heat is removed through the probe housing, is not treated here. Thus, the heat fluxes were produced directly in the probe, and served for the detection of thermoacoustic effects.

Figure 2 shows oscillograms of the velocity and pressure pulsations for a) linear resonance with a frequency $f_1 = a/4L$ and b) the second nonlinear resonance with a frequency $f_2 = 3a/8L$ in a tube of length $L_0 = 3.485$ m, where $L = L_0 + m^2 l_0 + (m^2 + m + 1)(n/3)$ and $m = R_0/R$.

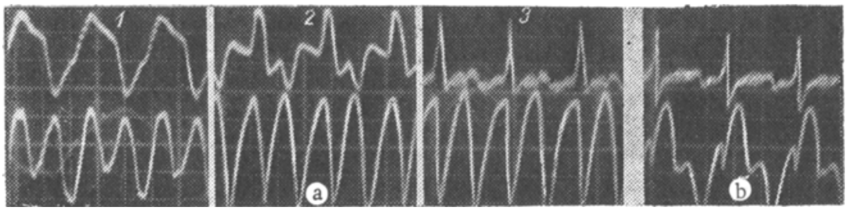

Fig. 2. Oscillograms of pressure (upper) and velocity (lower) for a) linear resonance (1, x/L_0 = 0.875; 2, 0.301; 3, 0.014; f_1 = 22 Hz) and b) the second nonlinear resonance (x/L_0 = 0.014, f_2^* = 33 Hz) in a tube of length L_0 = 3.485 m.

The oscillograms show that for the resonances mentioned shock waves are emitted from the open end of the tube. For the first nonlinear resonance the pressure pulsations at the open end of the tube retain a symmetric form, and therefore one can say that shock waves are not emitted. We note that loss of symmetry and the appearance of a discontinuity are observed near the piston (frame 1 of Fig. 2a).

Figure 3 shows the frequency distributions of a) the Nusselt numbers and b) the dimensionless semirange of velocity pulsations at the open end of the tube. All the relations have a resonance character; the resonance frequencies of the Nusselt numbers and the dimensionless semiranges of the velocity pulsations coincide. As the length of the tube is increased, the semirange of the pulsations and the Nusselt number decrease.

The solid curves in Fig. 3a represent the average Nusselt number over a period of the oscillations calculated by the quasistationary theory [8]:

$$\mathrm{Nu} = \frac{C\omega R}{\pi\lambda}\int_{0}^{2\pi/\omega}(Uf(t))^n\,dt = C_1\mathrm{Re}_k^n, \tag{3}$$

where C_1 = 0.027 and n = 0.8.

It is easy to see that the quasistationary theory gives a good description of the experimental data for the open end of the tube. The divergence of the data increases with the distance from the open end of the tube. The high value of the exponent n, characteristic of fully developed turbulent flows [9], should be noted.

The distributions of the Nusselt number and the dimensionless semirange of the velocity along the length of the tube are shown in Figs. 4a and b, respectively. As might be expected, the heat-transfer coefficient decreases with the distance from the open end of the tube; a decrease of the length of the tube is conducive to an increase of the semirange of the velocity pulsations and an increase in the heat-transfer coefficient. The results shown in Figs. 3a and 4a are satisfactorily described by the relation (solid curves in Fig. 4a)

$$\mathrm{Nu} = C_1\,\mathrm{Re}_k^n + C_2\left[1 - \cos\frac{\pi x^*}{L_0}\right], \tag{4}$$

where C_2 = 95.

It follows from (4) that an effect arises inside the tube leading to values of the Nu number larger than those from the quasistationary theory. The increase is larger the farther the heat flux probe is located from the open end of the tube. At the open end the effect vanishes completely.

Using a flat channel as an example, we try to explain the observed effect qualitatively, although the presence of heat transfer complicates the problem considerably.

Because of the nonuniform temperature distribution the sound speed is nonuniform over a cross section of the tube, i.e., the propagation of oscillations is nonisentropic, and in addition the sound speed is nonuniform along the length of the tube.

For relatively high-frequency oscillations an oscillating boundary layer is formed near the channel surface. If the thickness of this layer $\delta = \sqrt{2\nu/\omega}$ is much less than the radius of the tube R, i.e., $R\sqrt{\omega/2\nu} \gg 1$, the oscillations in the flow core are practically isentropic. For a tube with $R = 2 \cdot 10^{-2}$ m, L_0 = 3.485 m, ω = 138 sec^{-1}, and $\nu = 15 \cdot 10^{-6}$ $\mathrm{m^2/sec}$, we obtain $R\sqrt{\omega/2\nu} \simeq 43$, which justifies the assumption that the oscillations are isentropic.

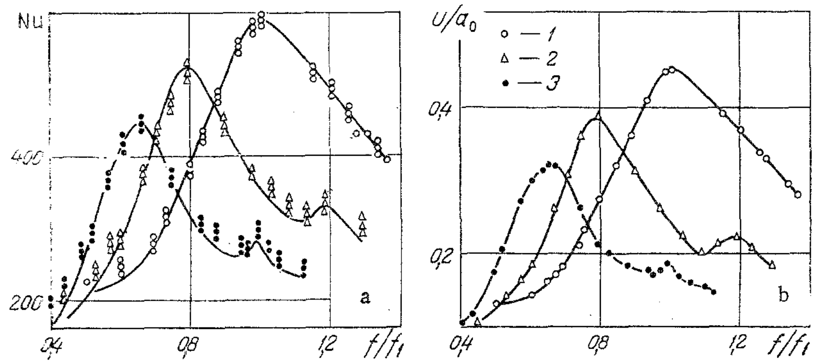

Fig. 3. a) Nusselt number Nu (1, 2, 3, experiment; solid curves calculated from Eq. (4)) and b) dimensionless semirange of velocity pulsations U/a_0 as functions of dimensionless frequency f/f_1 (f_1 = 22 Hz) in a tube with 1) L_0 = 3.485; 2) 4.485; 3) 5.485 m.

The relation between the nonuniformity of the time average of the temperature distribution and the nonuniformity of the sound speed along the length of the tube can be estimated in the following way. The one-dimensional acoustic equations in an ideal stationary medium have the form [10]

$$\rho_0 \frac{\partial U_1}{\partial t} + \frac{\partial p_1}{\partial x} = 0, \tag{5a}$$

$$\frac{\partial \rho_1}{\partial t} + U_1 \frac{\partial \rho_0}{\partial x} + \rho_0 \frac{\partial U_1}{\partial x} = 0, \tag{5b}$$

where the nonuniformity is taken into account by the term $U_1(\partial\rho_0/\partial x)$. We take account of this contribution by assuming that the process is isentropic in the flow core: $p_1 = \rho_1 a^2$. In a half-open tube with a uniform density distribution $U_1 = U_\infty \sin(kx + \alpha_0) \exp(i\omega t)$, $p_1 = i\rho_0 a U_\infty \cdot \cos(kx + \alpha_0) \exp(i\omega t)$. Then it is easy to see that for

$$(1/\rho_0)\, \partial\rho_0/\partial x \ll \frac{\omega}{a} \operatorname{ctg}(kx + \alpha_0) \tag{6}$$

the effect of the nonuniformity of the density (temperature) on resonance oscillations in a half-open tube can be neglected. Calculation shows that for nonlinear oscillations in the tubes investigated condition (6) takes the form $(1/\rho_0)\partial\rho_0/\partial x \ll (6.71\text{–}4.09)\omega/a$ near the piston, and $(1/\rho_0)\partial\rho_0/\partial x \ll (0.15\text{–}0.24)\omega/a$ at the open end. Assuming that the density (temperature) gradient for a given position of the probe does not change along the length of the tube, and T_0 = 300°K, it can be shown that with an acceptable accuracy (10%) the temperature drop should not exceed 7-11°C for the probe at the open end, and 300-190°C near the piston. Taking account of the fact that for large-amplitude oscillations there is an intense intermixing with the surrounding medium at the open end of the tube, condition (6) is easily satisfied.

Let us consider the boundary layer equations in compressible flow without restriction (6) when the core flow is isentropic. They have the form [8]

$$\begin{gathered}\rho\left(\frac{\partial \tilde{u}}{\partial \tilde{t}} + \tilde{u}\frac{\partial \tilde{u}}{\partial \tilde{x}} + \tilde{v}\frac{\partial \tilde{u}}{\partial \tilde{y}}\right) = -\frac{\partial p}{\partial \tilde{x}} + \frac{\partial}{\partial \tilde{y}}\left(\mu\frac{\partial \tilde{u}}{\partial \tilde{y}}\right),\\ \frac{\partial \rho}{\partial t} + \frac{\partial(\rho\tilde{u})}{\partial \tilde{x}} + \frac{\partial(\rho\tilde{v})}{\partial \tilde{y}} = 0.\end{gathered} \tag{7}$$

Following [11], we introduce the transformations

$$\tilde{u} = \frac{\rho_0}{\rho}\frac{\partial\psi}{\partial \tilde{y}}, \quad \tilde{v} = -\frac{\rho_0}{\rho}\left(\frac{\partial\psi}{\partial \tilde{x}} + \frac{\partial y}{\partial t}\right), \quad y = \int_0^{\tilde{y}} \frac{\rho}{\rho_0}\, d\tilde{y}.$$

Simple calculations show that Eqs. (7) go over into the familiar boundary-layer equations for an incompressible fluid

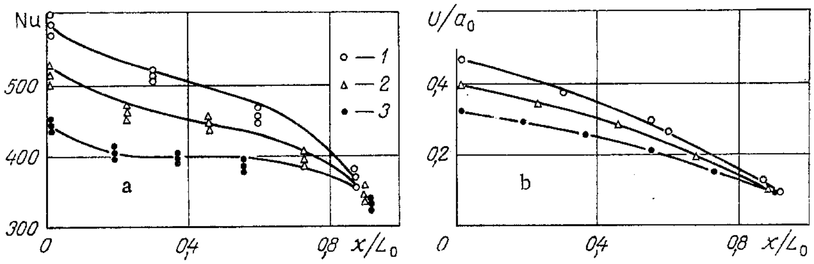

Fig. 4. Distribution of a) Nusselt number Nu (1, 2, 3, experiment; solid curves calculated from Eq. (4)) and b) dimensionless semirange of velocity pulsations U/a_0 over the length of the tube x/L_0: 1) L_0 = 3.485 m (f_1 = 22 Hz); 2) L_0 = 4.485 m (f_1 = 17.5 Hz); 3) L_0 = 5.485 m (f_1 = 14.5 Hz).

$$\frac{\partial u}{\partial t}+u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}=-\frac{1}{\rho}\frac{\partial p}{\partial x}+\nu_0\frac{\partial^2 u}{\partial y^2},\qquad \frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}=0. \tag{8}$$

To analyze heat transfer it is necessary to supplement Eqs. (8) by the equations for the boundary-layer temperature, which in the new "incompressible" variables have the form

$$\frac{\partial T}{\partial t}+u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}=\chi\frac{\partial^2 T}{\partial y^2}+\frac{\nu_0}{c_p}\left(\frac{\partial u}{\partial y}\right)^2. \tag{9}$$

We seek the solution of Eqs. (8) and (9) for $\varepsilon = U_\infty/\omega\lambda \ll 1$ and $R\sqrt{\omega/2\nu} \gg 1$ by a perturbation method.

For the first approximation with Pr = 1

$$\frac{\partial u_1}{\partial t}=-\frac{1}{\rho}\frac{\partial p_1}{\partial x}+\nu_0\frac{\partial^2 u_1}{\partial y^2}, \tag{10a}$$

$$\frac{\partial u_1}{\partial x}+\frac{\partial v_1}{\partial y}=0, \tag{10b}$$

$$\frac{\partial T_1}{\partial t}+u_1\frac{\partial T_0}{\partial x}+v_1\frac{\partial T_0}{\partial y}=\nu_0\frac{\partial^2 T_1}{\partial y^2} \tag{10c}$$

with the boundary conditions

$$u_1=v_1=0,\quad T_1=0,\quad y=0,\quad U_1=U_1(x)\exp(i\omega t),\quad y\to\infty.$$

Beyond the limits of the boundary layer Eq. (10a) takes the form

$$\rho i\omega U_1(x)\exp(i\omega t)=-\frac{\partial p_1}{\partial x},$$

and the solution of (10a) has the form

$$u_1=U_1(x)f_1(y)\exp(i\omega t).$$

From the equation of continuity (10b) it is easy to obtain

$$v_1=U_1'(x)f_2(y)\exp(i\omega t),$$

where the prime denotes differentiation with respect to x.

To solve (10c) it is necessary to know the explicit form of the expressions for $\partial T_0/\partial x$ and $\partial T_0/\partial y$. For high-frequency oscillations, however, conditions are satisfied which permit the determination of the form of T_1 without knowing the explicit form of these quantities. Actually, a significant change in T_0 occurs at a distance R, whereas the analogous distance for u_1 and v_1 is $\delta = \sqrt{2\nu/\omega}$. For $R\sqrt{\omega/2\nu} \gg 1$ the dependence of $\partial T_0/\partial x$, and $\partial T_0/\partial y$ on x and y can be ignored in the integration of (10c). Then we obtain

$$T_1\simeq\left[U_1(x)\frac{\partial T_0}{\partial x}f_1^*(y)+U_1'(x)\frac{\partial T_0}{\partial y}f_2^*(y)\right]\exp(i\omega t). \tag{11}$$

Retaining only stationary terms and Pr = 1, the equations of the second approximation have the form

$$\left\langle u_1 \frac{\partial u_1}{\partial x}\right\rangle + \left\langle v_1 \frac{\partial u_1}{\partial y}\right\rangle = \langle U_1 U_1' \rangle + \nu_0 \frac{\partial^2 \langle u_2 \rangle}{\partial y^2}, \tag{12a}$$

$$\frac{\partial \langle u_2 \rangle}{\partial x} + \frac{\partial \langle v_2 \rangle}{\partial y} = 0, \tag{12b}$$

$$\left\langle u_1 \frac{\partial T_1}{\partial x}\right\rangle + \left\langle v_1 \frac{\partial T_1}{\partial y}\right\rangle + \left\langle u_2 \frac{\partial T_0}{\partial x}\right\rangle + \left\langle v_2 \frac{\partial T_0}{\partial y}\right\rangle = \nu_0 \frac{\partial^2 \langle T_2 \rangle}{\partial y^2} + \frac{\nu_a}{c_p}\left\langle \left(\frac{\partial u_1}{\partial y}\right)^2\right\rangle, \tag{12c}$$

where < > denotes the time average. Omitting the sign < > from now on, $U_2 = U_1 U_1' f_3(y)$, $v_2 = (U_1'^2 + U_1 U_1'')f_4(y)$. Taking account of the expressions for u_1, v_1, T_1, u_2, and v_2, we obtain

$$\begin{aligned} \nu a \frac{\partial^2 T_2}{\partial y^2} &= U_1 U_1' \left[(f_3 + f_1 f_1^* + f_1^{*\prime} f_2) \frac{\partial T_0}{\partial x} + (f_1 f_2^* + f_1^* f_2) \frac{\partial^2 T_0}{\partial x \partial y} \right] + \\ &+ U_1'^2 \left[f_2^* f_2 \frac{\partial^2 T_0}{\partial y^2} + (f_4 + f_2^{*\prime}) \frac{\partial T_0}{\partial y} \right] + U_1^2 \left(f_1 f_1^* \frac{\partial^2 T_0}{\partial x^2} - \frac{\nu_0}{c_p} f_1'^2 \right) + U_1 U_1'' (f_4 + f_1 f_2^*) \frac{\partial T_0}{\partial y}. \end{aligned} \tag{13}$$

The integration of Eq. (13) with respect to y from 0 to ∞ would make it possible to determine the heat flux due to secondary flows, but this is not necessary to determine the essential properties of the phenomenon.

Since $l \ll L$ and $R \ll L$, the boundary conditions at the ends of the channel will affect heat transfer of the probe in the zero approximation only if the probe is located near one end. When the probe is located at a distance $\sigma = (5\text{–}10)R$ from the ends, an estimate shows that this effect is negligible, so that we can seek the zero approximation by assuming that the probe is located in an infinitely long channel. Then the problem becomes symmetrical with respect to the cross section at the middle of the probe at the point $x = \sigma$. In this case the ends of the probe are in the cross sections at $x = \sigma + l/2$ and $x = \sigma - l/2$. In order to find the heat flux transferred by the probe to the oscillating air stream it is necessary to integrate the specific heat flux of Eq. (11) over the whole heat-transfer surface of the probe. In view of the symmetry and the condition $l \ll L$, it is easy to see that the final result will not contain terms proportional to $U_1 U_1'$. Then, if we assume $U_1 = U_\infty \cdot \sin(kx + \alpha)\exp(i\omega t)$, which is equivalent to (6), the total flux q_u due to secondary flows will contain a sum of terms proportional to $\sin^2 kx$ and $\cos^2 kx$, i.e.,

$$q_u = A + B\cos 2(kx + \alpha_0). \tag{14}$$

By using $\omega = \pi a/2L$, $k = \omega/a$, $x^* = x - L_0$ as $\alpha_0 \to 0$, the appearance of the second term on the right-hand side of Eq. (4) can be explained completely.

Thus, heat transfer in a half-open tube containing gas undergoing nonlinear oscillations with the emission of periodic shock waves is satisfactorily described by the quasistationary theory including thermoacoustic effects.

NOTATION

l_0, amplitude of piston displacement; R_0, radius of piston; R, inside radius of tube; R_1, inside radius of asbestos layer; R_a, outside radius of flange; l_p, length of working portion of probe; l_m, thickness of thermal insulation disks; x, distance from piston; x^*, distance from open end of tube; L_0, length of tube; h, height of conical reducer; y, distance from wall; α, heat-transfer coefficient; q_F, specific heat flux; T_c, temperature of inner surface of wall; T_∞, temperature of oscillating air stream; Δt_a, temperature drop across asbestos layer; Δt_d, temperature drop across thermal insulation disks; $Nu = 2\alpha R/\lambda$, Nusselt number; $Re_k = 2U_1 R/\nu$, Reynolds number; λ, thermal conductivity of air; λ_a, thermal conductivity of asbestos; $\nu = \mu/\rho$, kinematic viscosity; μ, dynamic viscosity; ρ, density of gas; χ, thermal diffusivity; ψ, temperature recovery factor; γ, adiabatic exponent; I, current; V, potential drop; A, B, C, C_1, C_2, constants; f(t), time dependence of velocity oscillations; p, pressure; t, time; λ^*, wavelength; $k = \omega/a$, wave number; ω, cyclic frequency of oscillations; f, excitation frequency; f_1, f_2^*, natural frequencies for linear and second nonlinear resonances; a_0, speed of sound in unperturbed gas; a, speed of sound at T_∞; $M = U/a$, Mach number; U, semirange of velocity pulsations; U_∞, amplitude of velocity pulsations at open end of tube; U_1, velocity pulsations in flow core; u, v, longitudinal and transverse velocity components in boundary layer; T_0, temperature of cooling part of the tube wall; δ, boundary layer thickness; σ, longitudinal coordinate of center of probe.

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EXPERIMENTAL STUDY OF THE MIXING OF TURBULENT, OPPOSITELY TWISTED STREAMS IN THE INITIAL SECTION IN AN ANNULAR CHANNEL

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A study was made of the effect of the tangential component of mean velocity on the mixing of oppositely twisted flows. The equivalence of the mechanisms of formation of the velocity profiles in the mixing layers of oppositely twisted and co-current flows was established.

The twisting of flows is one of the most frequently used means of intensifying mixing processes. A large number of works have now been published involving theoretical and experimental investigation of twisted flows — [1-4], for example. These studies examine flows twisted in one direction and note that the swirling significantly complicates the flow pattern and analysis of the laws of its development as well as generalization of the data. The same situation holds with regard to the case of coaxial streams with opposite directions of rotation. At least two additional factors, characterizing the intensity of the twisting in each flow, must also be considered here as determining parameters.

The present work experimentally studies the laws governing the mixing of oppositely twisted flows and is a continuation of the work in [5]. In the latter, it was shown on the basis of analysis of loss for unseparated mixing of flows that the use of high degrees of twisting is best from the point of view of reinforcement of the mixing properties of the flows.

The experiments were conducted on the unit shown in Fig. 1. An air flow was created by a fan 1 installed at the outlet of a channel. The air entered annular channels 2, at the inlet of which had been installed grates 3 of variable through cross section. This allowed us to independently vary the gas flow rate in each channel. The flow was twisted by tangential

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