LITERATURE CITED

1. R. G. Galiullin and G. G. Khalimov, "Nonlinear oscillations of a gas in open tubes," Inzh.-Fiz. Zh., 37, 1043-1050 (1979).
2. G. G. Khalimov, R. G. Galiullin, and I. P. Revva, "Large-amplitude oscillations of a gas in an open tube," in: Scientific Research in Mathematics and Physics [in Russian], Nizami Tashkent Pedagogical Inst. (1979), pp. 3-12.
3. R. G. Galiullin, G. G. Khalimov, and V. N. Podymov, "End effects in nonlinear oscillations of a gas in open tubes," in: Scientific Research in Mathematics and Physics [in Russian], Nizami Tashkent Pedagogical Inst. (1979), pp. 12-21.
4. L. Van Wijngaarden and J. H. M. Disselhorst, "Resonant gas oscillations in open pipes," Arch. Mech. Stosow., 31, 115-124 (1979).
5. O. E. Karamercan and J. L. Gainer, "The effect of pulsations on heat transfer," Ind. Eng. Chem. Fundam., 18, 11-15 (1979).
6. M. Faghri, K. Jovdani, and A. Faghri, "Heat transfer with laminar pulsating flow in a pipe," Lett. Heat Mass Transfer, 6, 259-270 (1979).
7. Heat Engineering Handbook [in Russian], Vol. 2, Énergiya, Moscow.
8. R. G. Galiullin, V. B. Repin, and N. Kh. Khalitov, Flow of a Viscous Fluid and Heat Transfer of Bodies in a Sound Field [in Russian], Kazan Univ. (1978).
9. B. S. Petukhov, Heat Transfer and Resistance in the Laminar Flow of Fluid in Pipes [in Russian], Énergiya, Moscow (1967).
10. M. A. Isakovich, General Acoustics [in Russian], Nauka, Moscow (1973).
11. C. R. Illingworth, "Effects of a sound wave on a compressible boundary layer," J. Fluid Mech., 3, 471-493 (1958).

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

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UDC 532.517.4

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

Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 44, No. 2, pp. 205-210, February, 1983. Original article submitted September 29, 1981.

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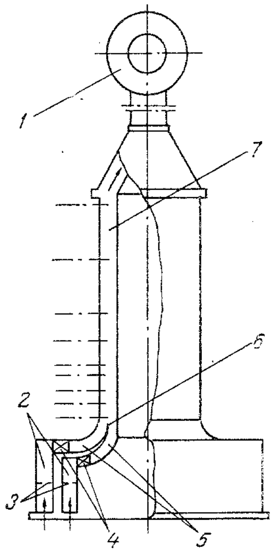

Fig. 1. Experimental unit.

swirl vanes 4. In the experiments, the angle of installation of the vanes was varied from 0 to ±60°. The air flow proceeded from the vanes into channels 5 and thence into the working part 7, which was an annular channel with an inside diameter of 0.22 m, an outside diameter of 0.34 m, and a length of 0.5 m. The ratio of the areas of the outer and inner channels at the inlet to the working part was roughly two. The dividing wall 6 was 1.2 mm thick.

Measurements were obtained with head-type flow meters and hot-wire anemometers which could be moved radically and rotated axially with the aid of remote-control traversing equipment.

The heat meter was a 6-mm-diameter cylinder with three 0.5-mm-diameter holes located 3 mm from its end. The holes were positioned over a 20° angle about the cylinder circumference. Such a design allowed us to determine the angle of the flow direction to within roughly 1°. The total and static pressures were determined on the basis of calibration of the meter in a nozzle.

The turbulence characteristics were measured with a constant temperature hot-wire anemometer with a bandwidth of 20 kHz. The probe of the anemometer had a single platinum filament 2-3 mm long and 7 μm in diameter located in a plane parallel to a tangent to the channel wall.

Measurements were made at nine **sections** along the channel and at 19 points over the radius. These sections are denoted in Fig. 1 by dot-dash lines. The distance from the end of the dividing wall to the measurement sites (probe locations) was 5, 25, 48, 81, 105, 155, 205, 305, and 405 mm.

In analyzing the test data, we ignored the radial component of mean velocity v_r compared to its axial v_z and tangential v_φ components. We also assumed that the flow was axisymmetric in all regimes. Checking showed that the total pressure head deviated from its mean about the circumference by no more than 5%.

The experiments showed that a fairly high level of turbulent pulsations is seen in the investigated stream at the boundary of the flows. For example, $\varepsilon = \langle v'^2 \rangle / v_0^2 \approx 40\%$ [5] with a twist of 37°/—64° (37 and 64° are the mean angles of twist of the internal and external flows at the inlet, respectively). This circumstance fundamentally complicates theoretical analysis of the measurement errors. The accuracy of the experiment was therefore checked by comparing the hot-wire anemometer measurements with the head meter findings. The discrepancy between the velocity profiles did not exceed 5-6% in any of the experiments. We also checked the accuracy of the experiments on the basis of the condition of mass conservation by the flows. This quantity changed by 5-10% along the channel, with the deviation from the mean values monotonically increasing with an increase in the twisting of the flows and having been maximal in the initial section. At the remaining stations, the deviations in flow mass were generally 1.5-2 times lower than the maximum. To reduce the effect of the error associated with inaccurate determination of the mass of the flow, the flow characteristics for each station were converted to dimensionless form relative to the local mean-flow-rate velocity $v_0(z)$.

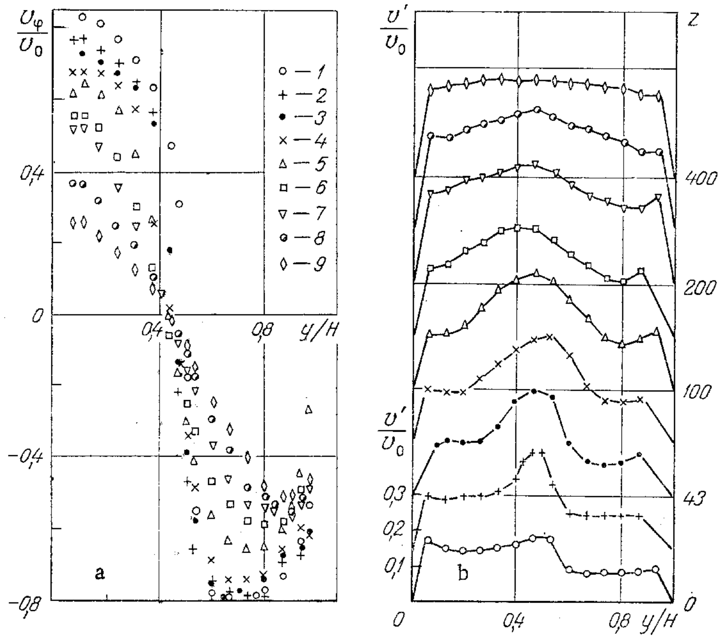

Fig. 2. Experimental profiles of tengential v_φ (a) and pulsative v' (b) components of velocity (v_0 = 26 m/sec, Re = 10^5): 1) z = 0; 2) 20 mm; 3) 43; 4) 76; 5) 100; 6) 150; 7) 200; 8) 300; 9) 400. z, mm.

The main goal of our investigation was to explain the effect of the tangential component of mean velocity v_φ on mixing of the flows. For this purpose, we regulated the air flow rates through the channel 5 and established a flow regime corresponding to m ≈ 1, where $m = v_2/v_1$. It turned out that the profiles of the axial component of mean velocity v_z at the inlet to the working part were close to uniform. The measurements also showed that the level of turbulence at the inlet was small compared with the turbulence generated in the flow and that the effect of this parameter on flow development could be ignored. Thus, the investigated flow is characterized by the following parameters: Reynolds number Re = v_0H/ν; twists of the internal and external flows, determined as the means of the angles between the direction of the velocity vectors and the channel axis; the form of the v_φ profiles at the channel inlet.

The mean flow velocity in the test ranged from 20 to 35 m/sec, which corresponded to Reynolds number of $(0.8\text{-}1.4)\cdot 10^4$. The twist angle ranged from 0 to 64°, which corresponded to angles of vane setting from 0 to 60°. The form of the v_φ profiles was left as the variable parameter.

The experiments showed that the maximum values of the tangential velocity component monotonically decrease along each of the channels. Meanwhile, the greater the angle of twist at the inlet, the greater the decrease in the tangential component. This situation characterizes the reinforcement of turbulent transfer between the flows. In fact, an increase in the level of turbulent pulsations was seen in the tests when the angle of twist was increased. As an example, Figure 2 shows radial profiles of the tangential velocity component v_φ and the pulsative component v' at different channel stations with twist angles of 37°/—33° in the internal and external flows, respectively.

It is apparent from the graphs that similitude of the velocities v_φ is not maintained along the channel. It is also apparent that the turbulent pulsations are reinforced at the boundary of the flows.

To explain the effect of the twist parameter on the form of the v_φ profiles, we will analyze the experimental data with respect to local values of velocity in each section.

Figure 3 shows graphs of the change in Δv_φ along the channel, plotted with respect to the "natural" coordinate ξ, for a twist of 37°/—33° (see Fig. 2):

$$\Delta v_\varphi = \frac{v_\varphi - v_{\varphi 1,m}}{v_{\varphi 2,m} - v_{\varphi 1,m}}, \tag{1}$$

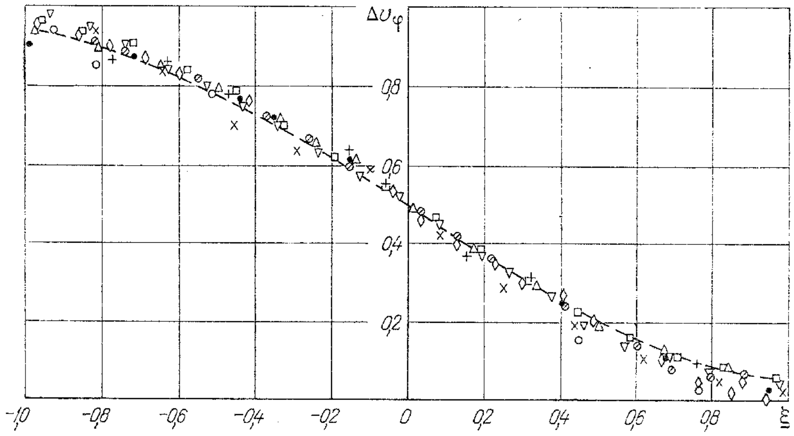

Fig. 3. Profiles of dimensionless excess velocity Δv_φ in relation to the "natural" coordinates ξ.

$$\xi = \frac{r - r_{0.5}}{r_{0.2} - r_{0.8}}. \tag{2}$$

Here $r_{0.2}$, $r_{0.5}$, and $r_{0.8}$ are coordinates of the points at which Δv_φ is equal to 0.2, 0.5, and 0.8, respectively; $-1 \leqslant \xi \leqslant 1$. It should be remembered that $v_{\varphi,m}$ and r_i depend on the distance along the channel axis.

It follows from the figure that the form of the Δv_φ profiles changes little along the channel and that these changes can be ignored within the limits of accuracy of the experiment. The experimental points for other twist angles also lie within the empirical spread. This permits the conclusion that, within the limits of accuracy of the experiment, the form of the $\Delta v_\varphi(\xi)$ profiles is independent of both the twist parameter and of the distance along the channel axis z, i.e., the profiles have a general form.

The dashed line in Fig. 3 shows the general excess-velocity curve for normal stream mixing on the initial section [6]. It is apparent that this profile approximates the resulting empirical data well. This is evidence of the fact that the mechanisms of formation of the profiles in the mixing zone are the same for the twisted flow examined here and a normal flow.

The intensity of turbulent exchange between the flows is assumed to be characterized by the angle of expansion of the mixing layer. For the stream mixing of flows, the law governing the increase in the width of this layer has the form [6]:

$$\frac{db}{dx} = c\,\frac{|v_1 - v_2|}{v_1 + v_2}. \tag{3}$$

The value of the empirical constant c for the initial section was taken as 0.27.

If the mixing of oppositely twisted flows is judged from profiles of the tangential velocity component v_φ, then, having adopted hypotheses similar to those used in deriving Eq. (3) in [6], we can obtain

$$\frac{db}{dz} = c_\varphi\,\frac{|v_{\varphi 1,m} - v_{\varphi 2,m}|}{2v_0}. \tag{4}$$

In connection with the fact that it is very difficult to empirically determine the width of the mixing layer, the distance between points at which the relative excess velocity is, for example, 0.8 and 0.2, i.e., $b_2 = r_{0.2} - r_{0.8}$, is normally used as the quantity characterizing the mixing rate.

Figure 4 shows curves of the change in $b_2(z)$ for different twists of the flows at the inlet. The nontrivial value of $b_2(0)$ has to do with the chosen method of analyzing the empirical data, as well as with the difficulty of obtaining stepped v_φ profiles at the inlet. Due to the similitude of the Δv_φ profiles in the mixing zone, b_2 and b turn out to be proportional. The proportionality factor depends on the form of the function chosen to approximate the $\Delta v_\varphi(\xi)$ profile.

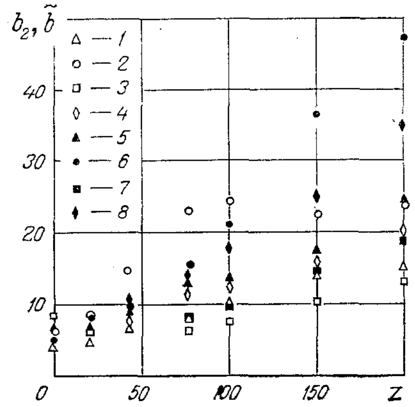

Fig. 4. Dependence of the expansion of the mixing layer on twisting of the flows: 1 3°/–52°; 2 52°/–43°; 3 16°/–21°; 4 37°/–33° (b_2); 5 3°/–52°; 6 56°/–63°; 7 16°/–21; 8 37°/–33° ($\tilde{b}$). z, mm; b_2, $\tilde{b}$, mm.

As noted above, a typical feature of the stream examined here is a monotonic reduction in the maximum values of the tangential velocity component in each of the flows. Allowing for this circumstance complicates determination of the mixing-layer expansion angle. It is therefore preferable to use a parameter connected with the change in the maximum gradient of the tangential velocity component in the mixing layer along the channel:

$$\tilde{b}(z) = \frac{|v_{\varphi 1,m} - v_{\varphi 2,m}|}{(\partial v_\varphi / \partial r)_m}. \tag{5}$$

This quantity, like b_2, is proportional to the width of the mixing layer. Figure 4 shows experimental values of $\tilde{b}$.

Analysis of the experimental data in the same manner as for stream mixing for the profile $\Delta v_\varphi(\eta) = (1 - \eta^{1.5})^2$ [6] showed that the value of the coefficient c_φ in Eq. (4) is close to the value of the corresponding coefficient for stream mixing on the initial section. Here η is a "universal" coordinate

$$\eta = \frac{r - r_1}{r_2 - r_1}, \tag{6}$$

where r_1 and r_2 are coordinates of the maximums of v_φ in the internal and external flows. It is typical that the spread of the values of both coefficients c and c_φ calculated on the basis of the empirical data with Eqs. (3) and (4) was the same. We may therefore take $c_\varphi = c = 0.27$.

This result confirms the above arguments regarding the sameness of the mechanism of formation of the velocity profiles in the mixing layer for oppositely twisted and co-current flows. Thus, in the cases examined, the development of the mixing layer is determined by the size of the shift of the tangential velocity component.

NOTATION

z, r, φ, cylindrical coordinate system; v_z, v_r, v_φ, components of mean velocity; v_0, mean-flow-rate velocity; v', pulsative component of velocity in the direction of the mean-velocity vector; R_1, R_2, internal and external radii of the annular channel; $H = R_2 - R_1$, radial gap of annular channel; $y = (r - R_1)/H$, dimensionless radial coordinate; ξ, "natural" coordinate; η, "universal" coordinate; b, width of mixing layer. Indices: 0, mean-flow rate; 1, internal flow; 2, external flow; m, maximal.

LITERATURE CITED

1. G. N. Abramovich et al., Turbulent Mixing of Gas Streams [in Russian], Nauka, Moscow (1974).
2. R. B. Akhmedov et al., Aerodynamics of a Twisted Stream [in Russian], Énergiya, Moscow (1977).
3. Handbook: Certain Problems of Investigating the Eddy Effect and Its Commercial Application [in Russian], Kuibyshev Aviation Institute, Kuibyshev (1974).

4. B. P. Ustimenko, Turbulent Transfer Processes in Rotating Flows [in Russian], Nauka, Alma-Ata (1977).
5. A. A. Sviridenkov, V. V. Tret'yakov, and V. I. Yagodkin, "Efficiency of mixing coaxial flows twisted in opposite directions," Inzh.-Fiz. Zh., 41, No. 3, 407-413 (1981).
6. G. N. Abramovich, Theory of Turbulent Streams [in Russian], Fizmatgiz, Moscow (1960).

HEAT EXCHANGE ON A THERMAL INITIAL SECTION IN THE EVAPORATION OF TURBULENT FALLING LIQUID FILMS

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UDC 536.242:532.62

An approximate analytical solution is obtained for a problem of convective heat transfer on a thermal initial section in the evaporation of turbulent falling liquid films.

The evaporation of falling liquid films finds application in evaporators, refrigeration systems, and various types of water distillation units [1]. The high rate of the process at low temperature heads makes it possible to process solutions of thermolabile substances, and in many cases it is the only method of doing so. It is also best to minimize the time in the heating zone for such substances, which makes it necessary to use relatively short heat-exchange surfaces. The effect of the thermal initial section becomes important in heat exchange on such surfaces. However, there are presently physical and mathematical difficulties attendant to allowing for the effect of this section on heat exchange in evaporation in falling liquid films, particularly in theoretical investigations.

Heat transfer on the thermal initial section in the evaporation of turbulent falling liquid films was studied theoretically in [2]. Here, the investigators examined several models for the turbulent liquid, being an arbitrary combination of the familiar Van Drist and Shablevskii relations. The modification consisted mainly of allowing for the decay of eddy viscosity with approach to the free surface of the film. Otherwise, the theoretical Nu numbers significantly exceed the empirical values [3, 4]. It was established that the most satisfactory data is obtained using the model in [3] for the internal region of the film and modifying this model with the relation in [5] for the external region. Thus, only a numerical solution of the energy equation was obtained for several Re and Pr numbers, which complicates further use of the results.

Obtained below is an approximate analytical solution of a problem of heat transfer on the thermal initial section in the evaporation of turbulent liquid films flowing down a vertical surface. Here, we use the eddy viscosity model of Millionshchikov [6, 7] for the entire film thickness. The model gives a zero value for eddy viscosity on the free surface of the film. As already noted, this position is in accord with current representations. It should be mentioned that the use of this model of turbulence in the case of heat exchange without evaporation [8] produces results in agreement with the experimental data.

We will assume the physical properties of the liquid to be constant. The flow is hydrodynamically stabilized, with a flat free surface. The velocity profile in the film is described by a power relation with the exponent 1/7.

Given these assumptions, the evaporation heat-transfer problem reduces to the solution of the differential equation [8]

$$\eta^{1/7}\frac{\partial\theta}{\partial\xi}=\frac{\partial}{\partial\eta}\left[\left(1+\frac{\nu_t}{\nu}\frac{\mathrm{Pr}}{\mathrm{Pr}_t}\right)\frac{\partial\theta}{\partial\eta}\right] \qquad (1)$$

with the conditions

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