TRANSIENT CONVECTIVE HEAT TREANSFER DURING VARIOUS MODES OF COOLING A HOT GAS IN PIPES

G. A. Dreitser, É. K. Kalinin, and V. A. Kuz'minov

UDC 536.244

Results are shown of an experimental study concerning the transient heat transfer in pipes with hot gas, and data pertaining to various modes of cooling are generalized.

Considerable attention is paid nowadays to research concerning the transient heat transfer processes in channels. Studies have shown that, when gases are heated in channels [1, 2], the transient heat transfer process is significantly affected by the anisothermality of the stream in terms of the temperature factor T_w/T_b. For this reason, the authors measured the transient heat transfer coefficient during the cooling of a hot gas in pipes.

Transients were produced by either stepwise or smooth increases in the temperature of a hot gas at the pipe entrance while the flow rate was maintained constant (G = const). The gas was beginning to cool, while the pipe was beginning to heat at a variable rate $\partial T_w/\partial \tau$. The experiments were performed with three pipe pieces of grade Kh18N10T steel over the following ranges of parameter values: Reynolds number Re from $3 \cdot 10^3$ to $2 \cdot 10^5$, temperature factor T_w/T_b from 0.3 to 1.0, pressure p from 0.4 to 2.0 MN/m^2, and temperature of the hot gas at the pipe entrance up to 1170°K. Pipe No. 1 had an inside diameter d = 8.65 mm, wall thickness δ = 0.186 mm, and length L = 1241 mm; pipe No. 2 had an inside diameter d = 42.8 mm, wall thickness δ = 0.6 mm, and length L = 2900 mm; pipe No. 3 had an inside diameter d = 9.82 mm, wall thickness δ = 1.15 mm, and length L = 1508 mm.

Air entering the test zone had been preheated in an electric furnace (for pipes Nos. 1 and 2) or in a combustion chamber operating on alcohol (for pipe No. 3). Throughout the testing time we recorded the variations in the air flow rate (with diaphragms operating at a supercritical pressure ratio), in the air temperature and pressure at the entrance and at the exit (with Chromel—Alumel thermocouples using wires 0.1 and 0.05 mm in diameters, and with model ID-2I inductive pressure transducers), and in the temperature of the outside surface of a pipe at 8-11 sections (with Chromel—Alumel thermocouples using wires 0.05 mm in diameter along pipes Nos. 1 and 2 but 0.1 mm in diameter along pipe No. 3). All instrument readings were transmitted to a model N-004 oscillograph.

All instruments to be used for transient measurements had been tared and checked for each transient mode against steady-state conditions before and after the test. The inertia of all instruments had been estimated and found within permissible limits to ensure reliable recording of data during transients. Heat leakage through the outside pipe surfaces was determined from calibration tests.

The transient heat transfer coefficient

$$\alpha(x, \tau) = \frac{q_w(x, \tau)}{T_b(x, \tau) - T_w(x, \tau)} \tag{1}$$

was determined according to a procedure analogous to that used in the heating tests [3]. The necessary values of $q_w(x, \tau)$ and $T_w(x, \tau)$ were found by solving the reverse heat conduction problem on the basis of temperature and thermal flux readings at the outside surface of a pipe wall, while the necessary values of $T_b(x, \tau)$ were found by solving the one-dimensional equation of energy. The Nusselt number was calculated according to the formula

Sergo Ordzhonikidze Institute of Aviation, Moscow. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 25, No. 2, pp. 208-216, August, 1973. Original article submitted March 20, 1972.

$$Nu_0 = 0.023Re^{0.8} \cdot Pr^{0.4} \cdot F\left(\frac{x}{d}\right), \qquad (2)$$

with [4]

$$F\left(\frac{x}{d}\right) = 3.115\,Re^{-0.07}\left(\frac{x}{d}\right)^{-0.54Re^{-0.167}}$$

for $0.85 \le x/d \le 50$ and $F(x/d) = 1$ for $x/d > 50$. All criterial groups here were defined in terms of the mean-calorimetric stream temperature T_b across a given pipe section.

The maximum relative error in the determination of the transient transfer coefficient was 10-20%.

The Nusselt number Nu, representing the instantaneous local heat transfer rate along a pipe, was 2.5-3.0 times higher than the quasisteady Nusselt number Nu_0 during the initial stages of heating with $\partial T_w / \partial \tau$ = 100-500°C/sec. During subsequent heating stages the pipe temperature T_w stabilized, with the factor K approaching unity. The K profile along x/d is interesting: the highest values of K are noted at the beginning and at the end of a pipe.

In [5] the authors showed a partial generalization of those tests performed at a constant initial wall temperature T_w equal to the ambient temperature. It had been revealed there that K varies with time in a manner which depends on the gas flow rate as well as on the law according to which the gas temperature at the entrance varies, but does not depend on the gas pressure. Analogous results were obtained in the heating tests [2]. An analysis of test data pertaining to heating of gas [6] or cooling of gas and their comparison with calculations for quasisteady turbulence profiles have shown that an appreciable departure of K from unity is not due to the superposition of transient heat conduction on convective heat transfer, but due to changes in the turbulence structure of the stream. The main role in these changes is played by the transiency of boundary conditions with regard to temperature (constraints on $\partial T_w/\partial \tau$) which, for the case of cooling, have been represented in [5] by the criterial group†

$$K_{\tau g} = \frac{\partial T_w}{\partial \tau} \cdot \frac{d}{T_w - T_{w_0}} \sqrt{\frac{\lambda}{c_p g G}}, \qquad (3)$$

independent of the pressure. The appearance of T_{w_0} in the expression for K_{Tg} makes the latter a function of the initial conditions. For this reason, it becomes worthwhile to take a different approach in setting up a criterion of thermal transiency and to use the results in [7, 8] pertaining to the mechanism of turbulence generation.

It has been shown in [8] that the turbulence characteristics change with the distance from the wall. Within the viscous sublayer ($0 \le y^+ \le 5$) the flow is not laminar. Velocity fluctuations of small amplitude and large amounts of fluid mass from adjacent regions penetrate into it. Within the $5 \le y^+ \le 15$ layer there periodically appear vortices which spurt to more remote regions. The interaction between these vortices and the mainstream, especially within the $7 \le y^+ \le 30$ layer produces turbulence which would usually be concentrated within a layer reaching only to $y^+ = 70$. The vortices, their buildup and spurts from that layer are random in time and depend on local conditions, but their intensity and average frequency of appearance are functions of the modal parameters of the average flow. According to [8], the average spurt frequency is

$$\bar{\omega} = \omega_0 Re^{1.75}, \qquad (4)$$

where $\omega_0 \approx 10^{-7}$ sec^{-1}. The turbulence in the mainstream at $y^+ > 70$ is transferred to the boundary layer by convection and diffusion. Here it is characterized by large-scale displacements but smaller velocity fluctuations.

An analysis of this mechanism leads to the conclusion that under transient conditions, apparently, the decisive role is played by the local variation of temperature within the $5 \le y^+ \le 30$ layer of the stream during an average period of time between two consecutive vortex formations at a given point. According to (4), this average time can be estimated as

$$\Delta\tau^* = \frac{1}{\bar{\omega}} = \frac{1}{\omega_0 Re^{1.75}}. \qquad (5)$$

It is sufficiently short and, therefore, the rise in wall temperature during time $\Delta\tau^*$ can be estimated as a linear increment

†This group is the Predvoditelev number (editor's note).

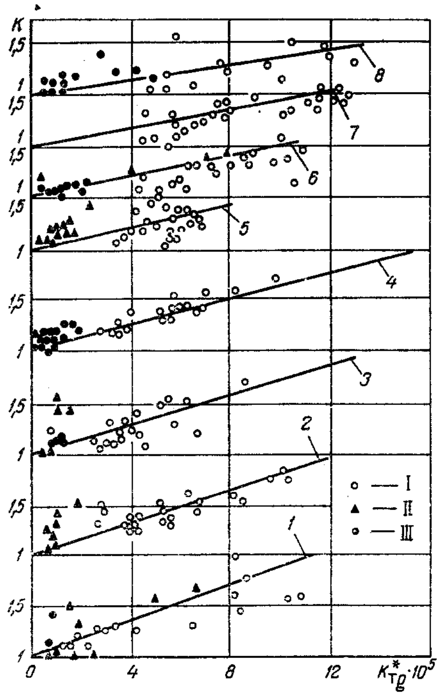

Fig. 1

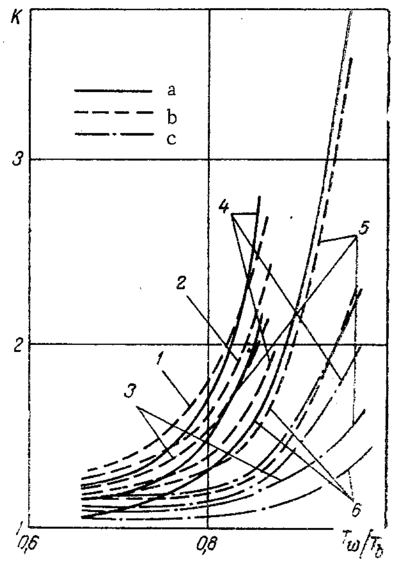

Fig. 2

Fig. 1. Relation $K = f(K^*_{Tg})$ for various values of Re and $0.7 \leq T_w/T_b \leq 0.8$: Re = $(3.2)\cdot 10^4$; $(4-5)\cdot 10^4$; $(5-6.25)\cdot 10^4$; $(6.25-7.83)\cdot 10^4$; $(7.83-9.83)\cdot 10^4$; $(9.83-12.25)\cdot 10^4$; $(12.25-1.55)\cdot 10^5$; $(1.55-2)\cdot 10^5$ (curves 1-8 respectively); I, II, and III) refer to pipes No. 1, No. 2, and No. 3 respectively.

Fig. 2. Relation $K = f(T_w/T_b)$ for various values of Re and K^*_{Tg}: a) Re = $(3.2-4)\cdot 10^4$; b) Re = $(6.25-7.83)\cdot 10^4$; c) Re = $(1.55-2)\cdot 10^5$; $K^*_{Tg}\cdot 10^5$ = 6, 5, 4, 3, 2, 1 (curves 1, 2, 3, 4, 5, 6 respectively).

$$\Delta T^* = \frac{\partial T_w}{\partial \tau}\cdot \Delta\tau^*. \tag{6}$$

After the formation and then departure of a vortex, there appears at the wall a layer of the order $y^+ \leq 30$ thick where the flow has slowed down and the velocity gradient is very small, whereupon this local slow layer interacts with a large mass of fluid moving at a velocity close to the mean overall velocity. As a result, fluid violently spurts from the slow layer into the farther regions. This spurt is the main source of turbulent energy.

One may hypothesize that, during transient cooling of gas while $\partial T_w/\partial\tau > 0$, the slow mass of gas at the wall has enough time to heat appreciably and to expand. This enlarges its surface of interaction with the larger mass moving faster than the hot gas, and thus results in a higher spurt rate. For this reason, on the one hand, the rate of turbulence generation rises while, on the other hand, a spurt of cold gas into the hot gas is ensured. As T_w/T_b decreases, however, the variations in the thermophysical properties, expecially in the gas density at the wall, will have more pronounced effects in terms of departure from isothermality.

According to the hypothetical mechanism by which transient heating of the pipe wall generates turbulence, one may expect the effect of transiency to become stronger with a higher volumetric gas expansion ratio

$$\beta = -\frac{1}{\rho}\left(\frac{\partial\rho}{\partial T}\right)_p. \tag{7}$$

The change in density during the heating period is

$$\Delta\rho = \rho_0 - \rho = \rho\beta\,(T - T_0) \approx \rho\beta\,\frac{\partial T_w}{\partial\tau}\cdot\Delta\tau^*. \tag{8}$$

It thus follows from an analysis of the mechanism by which thermal transiency affects the structure of a turbulent gas stream that the dimensionless parameter

$$K_\tau = \frac{\Delta\rho}{\rho} = \frac{\partial T_w}{\partial\tau} \cdot \frac{\beta}{\omega_0 \mathrm{Re}^{1.75}} \tag{9}$$

may be regarded as a criterion of this interaction. If one considers that $\beta = 1/T_W$ for a gas at the wall temperature T_W, then

$$K_\tau = \frac{\partial T_w}{\partial\tau} \cdot \frac{1}{T_w} \cdot \frac{1}{\omega_0 \cdot \mathrm{Re}^{1.75}}, \tag{10}$$

or, since the effect of the Reynolds number is accounted for separately,

$$K'_\tau = \frac{\partial T_w}{\partial\tau} \cdot \frac{1}{T_w} \cdot \frac{1}{\omega_0}, \tag{11}$$

where ω_0 is a dimensional constant which could be conveniently replaced by the time scale factor $d\sqrt{\lambda/c_p gG}$ just as in [2, 5]. The criterion of thermal transiency becomes then

$$K^*_{\tau g} = \frac{\partial T_w}{\partial\tau} \cdot \frac{d}{T_w}\sqrt{\frac{\lambda}{c_p gG}}. \tag{12}$$

The test points which had been plotted for transient processes with a constant initial wall temperature were subsequently evaluated on the basis of this criterion so as to yield an empirical relation of the form K = f(K^*_{Tg}). Because all quantities here varied interdependently during the experiment, the tests points were evaluated in the form K = f(K^*_{Tg}) for various ranges of Re and T_W/T_b, whereupon this relation was referred to the average values of Re and T_W/T_b over their ranges of variation. The ranges of the Reynolds number Re were respectively between Re = $3.2 \cdot 10^4$, $4 \cdot 10^4$, $5 \cdot 10^4$, $6.25 \cdot 10^4$, $7.83 \cdot 10^4$, $9.83 \cdot 10^4$, $1.225 \cdot 10^5$, $1.55 \cdot 10^5$, $2 \cdot 10^5$ and the ranges of the temperature factor T_W/T_b were respectively between 0.6, 0.7, 0.8, 0.9, 1.0. The K = f(K^*_{Tg}) relations thus obtained are shown in Fig. 1 for one range of the temperature factor and several ranges of the Reynolds number. The corresponding test points for the three pipes are in satisfactory agreement.

Ratio K increases with increasing K^*_{Tg}, which is explained by a more thorough development of turbulence at a higher K^*_{Tg} number. According to (12), the K^*_{Tg} number decreases when the gas flow rate, or the Reynolds number, increases. This is not an adequate interpretation of how the Reynolds number affects the ratio K, however, because Figs. 1 and 2 indicate that the effect of the K^*_{Tg} number on the ratio K becomes weaker at a higher Reynolds number. When the Reynolds number becomes higher and turbulence develops to a rising overall level, the relative effect of the additional turbulence development due to transiency becomes weaker. The effect of Re on K weakens as T_W/T_b increases.

Unlike in a steady process, where the temperature factor has no effect on the heat transfer coefficient during cooling [4, 9], its effect here is significant during a transient process (Fig. 2). The effect of the K^*_{Tg} number on the ratio K becomes weaker at lower values of T_W/T_b, with K decreasing when K^*_{Tg} and Re remain constant. This trend is probably due to a transverse mass transfer across the stream, as a result of a changing longitudinal temperature profile.

During a transient process the mean temperature along the stream drops faster than the wall temperature. The gas density increases more in the mainstream than at the wall, therefore, and this is compensated by a radial mass transfer from the wall to the mainstream. The higher the mass transfer rate is, the lower is the heat transfer rate. The more anisothermal the stream is (the smaller the factor T_W/T_b is), the more differently do T_b and T_W vary longitudinally and the higher is the rate of radial mass transfer. Under transient conditions, when $\partial T_W/\partial\tau > \partial T_b/\partial\tau$ at a given section, the rate of radial mass transfer increases, because the difference between the increasing densities in the mainstream and at the wall becomes larger with rising temperatures T_W and T_b. This trend weakens at higher values of T_W/T_b.

All test data pertaining to transient heat transfer, with the three experimental pipes of different diameters and wall thicknesses, with different laws governing the variation of the hot gas temperature at the entrance, and with a constant mass flow rate, were evaluated in the form K = f(K^*_{Tg}, Re, T_W/T_b) and then generalized by a single relation

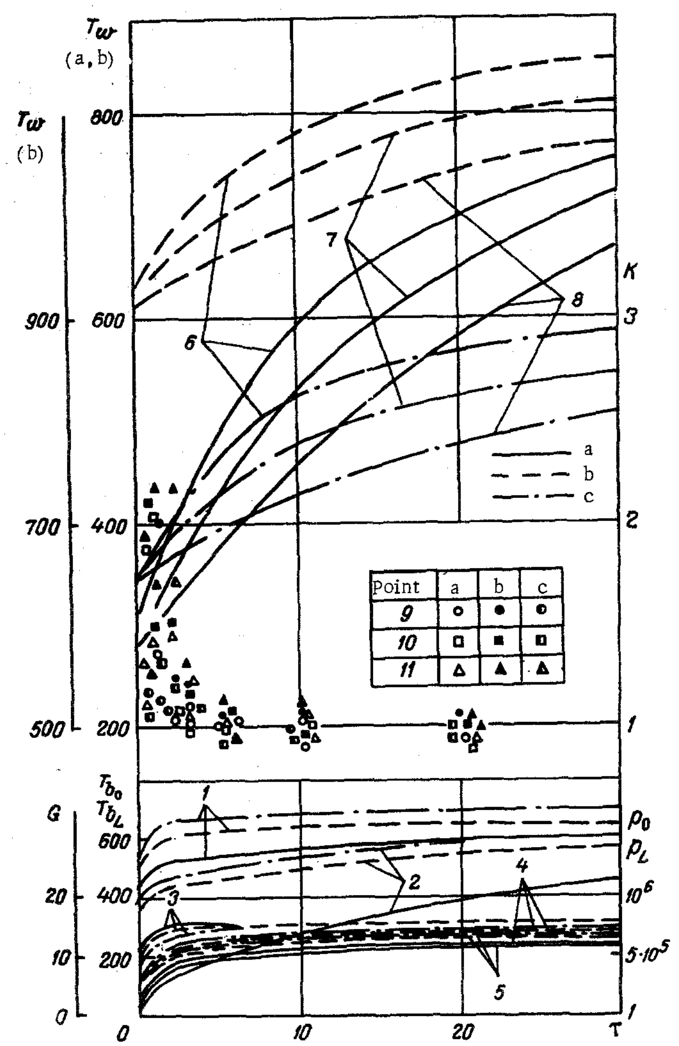

Fig. 3. Temperature of the hot gas at the entrance T_{b_0}, °C (1), at the exit T_{bL}, °C (2), gas flow rate G, g/sec (3), gas pressure at the entrance p_0, N/m^2 (4), at the exit p_L, N/m^2, wall temperature T_w, °C at sections x/d = 8.05, 53.9, and 115 (curves 6, 7, and 8 respectively), and ratio K at sections x/d = 8.05, 53.9, and 115 (curves 9, 10, and 11 respectively), as functions of time τ, sec during the transient process with cyclic discharges of hot gas: a) cycle No. 1, b) cycle No. 2, c) cycle No. 3.

$$K = 1 + \left\{\left[14.97\left(\frac{T_w}{T_b}\right)^3 - 16.07\left(\frac{T_w}{T_b}\right)^2 - 0.526\left(\frac{T_w}{T_b}\right) + 3.193\right] \times (\mathrm{Re}\cdot 10^{-5})^{1.85-3\frac{T_w}{T_b}} + 46.77\left(\frac{T_w}{T_b}\right)^3 - 119.1\left(\frac{T_w}{T_b}\right)^2 + 99.09\left(\frac{T_w}{T_b}\right) - 27.08\right\} K^*_{Tg}\cdot 10^5 \qquad (13)$$

for K^*_{Tg} from 0 to $2\cdot 10^4$, Re from $3.2\cdot 10^4$ to $20\cdot 10^4$, and T_w/T_b from 0.6 to 1.

It must be noted that relation (13) provides a realistic basis for comparing our results with known relations for steady cooling of a gas in a pipe. The quasisteady effects of Re, x/d, and T_w/T_b on the heat transfer coefficient have been accounted for in the Nusselt number Nu_0, while K being a function of Re and T_w/T_b indicates that these two quantities affect the Nusselt number Nu differently under transient than under steady conditions. It must also be noted that relation (13) refers to instantaneous local values of the heat transfer coefficient along a pipe.

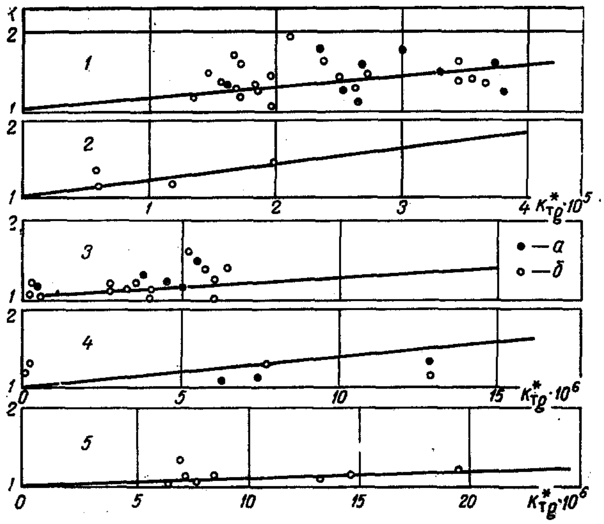

Fig. 4. Comparison of test points corresponding to a cyclic discharge of hot gas with relation (13): 1) Re = (5–6.25)·10^4; T_w/T_b = 0.9–1; 2) (4–5)·10^4 and 0.9–1 respectively; 3) (5–6.25)·10^4 and 0.8–0.9; 4) (4–5)·10^4 and 0.8–0.9; 5) (4–5)·10^4 and 0.7–0.8; solid lines represent relation (13); a) cycle No. 2; b) cycle No. 3.

In order to verify the feasibility of using relation (13) for calculations of the transient heat transfer in channels with initial conditions other than had been considered here, we tested pipe No. 3 with periodic discharges through the test zone. The following cyclic flow modes were realized: discharge for 90 sec with a pause 10 sec long, discharge for 50 sec with a pause 50 sec long, and discharge for 10 sec with a pause 90 sec long. The pipe cooled down during a pause, while the hot gas was diverted through a bypass, and the pipe heated up again during a next discharge of hot gas through it. Since there was not enough time for the pipe to cool down completely during a pause, each successive heating cycle began at a higher wall temperature and continued at a lower $\partial T_w/\partial \tau$ but higher T_w and T_w/T_b.

In Fig. 3 is shown the time variation of the various parameters during three cycles of one such test: 50 sec discharge and 50 sec pause. While T_w and T_w/T_b vary appreciably at the start of the first and the second cycle, they vary much less during the second and the third cycle. The test points do not seem to fit the universal relations K = f(K_{Tg}, Re, T_w/T_b) in [5] for a constant initial wall temperature T_{w_0} (with zero initial heat transfer). On the other hand, the test data evaluated in terms of K = f(K^*_{Tg}, Re, T_w/T_b) agree closely with relation (13) (Fig. 4). The K^*_{Tg} number, which is independent of the initial conditions, thus makes it possible to determine the rate of transient heat transfer at any instant of time throughout the process from the change in T_w, regardless of the law according to which the wall temperature T_w varies.

NOTATION

c_p	is the specific heat of the gas;
d	is the inside diameter of the pipe;
$g = 9.8\ m/sec^2$	
G	is the mass flow rate;
$K = Nu/Nu_0$	
K_{Tg}, K^*_{Tg}, K_T, K'_T	are the dimensionless groups describing the variation of T_w with time;
L	is the pipe length;
Nu	is the Nusselt number defined for transient conditions;
Nu_0	is the Nusselt number defined for transient conditions but on the basis of the quasisteady relation;
r	is the radius;
p	is the pressure;
q_w	is the thermal flux density;
Pr	is the Prandtl number;
Re	is the Reynolds number;

T_w is the wall temperature;
T_{w_0} is the wall temperature at time $\tau = 0$;
T_b is the mean-over-the-mass stream temperature;
w_x is the axial velocity;
x is the axial coordinate;
y^+ is the dimensionless distance from the wall;
α is the heat transfer coefficient;
β is the thermal expansivity;
δ is the wall thickness;
λ is the thermal conductivity;
ρ is the density;
τ is the time;
ω is the frequency.

LITERATURE CITED

1. V. K. Koshkin, E. K. Kalinin, G. A. Dreitser, B. M. Galitseiskii, and V. G. Izosimov, Internatl. J. Heat and Mass Transfer, 13, No. 8 (1970).
2. É. K. Kalinin, G. A. Dreitser, B. S. Baibikov, and A. S. Neverov, in: Heat and Mass Transfer, Vol. 1, Part 1 [in Russian] (1972).
3. B. M. Galitseiskii, G. A. Dreitser, V. G. Izosimov, É. K. Kalinin, and V. K. Koshkin, Teplofiz. Vys. Temp., 5, No. 5 (1967).
4. N. I. Artamonov, Yu. I. Danilov, G. A. Dreitser, and É. K. Kalinin, Teplofiz. Vys. Temp., 8, No. 6 (1970).
5. É. K. Kalinin, G. A. Dreitser, and V. A. Kuz'minov, in: Heat and Mass Transfer, Vol. 1, Part 1 [in Russian] (1972).
6. B. S. Baibikov, G. A. Dreitser, É. K. Kalinin, and A. S. Neverov, Teplofiz. Vys. Temp., 10, No. 6 (1972).
7. S. Y. Kline, W. S. Reynolds, F. A. Schraul, and P. W. Runstadler, J. Fluid Mechan., 35, No. 4 (1967).
8. E. R. Corino and R. S. Brodkey, J. Fluid Mechan., 37, No. 1 (1969).
9. A. B. Ambrazyavichyus, A. A. Zhukauskas, and P. Yu. Valyatkyaichyus, in: Heat and Mass Transfer, Vol. 1, Part 1 [in Russian] (1972).