uniform deformation of the drop-size spectrum, but without any very definite extremum in the drop-size distribution, depending on the gas velocity and being less noticeable with distance from the film surface.

The mean drop sizes for all the liquids studied are shown in Fig. 4 as a function of the gas velocity. The dependence of the mean drop size on the gas velocity is weaker for high-viscosity liquids than for low-viscosity liquids, where the mean drop size of the glycerin solutions is almost the same as for alcohol at low gas velocities.

In the case of a viscous liquid, thinner jets are formed and the drops obtained during dissociation of these jets will be finer. For low-viscosity liquids, the spray and secondary breakup of detached drops in the boundary layer intensify with the increase in gas velocity. The strong dependence of the mean drop size of a low-viscosity liquid on the gas velocity is evidently due to this. Hence, the drop sizes of a viscous liquid vary slightly with the increase in velocity, and the drops of a low-viscosity liquid turn out to be finer at relatively high gas velocities than for a high-viscosity liquid.

NOTATION

d, diameter; a, characteristic geometric size; A, wave amplitude; n_A, number of waves in the interval ΔA; n_{d_D}, number of drops in an interval Δd_D; u, velocity; τ, time; ν, coefficient of kinematic viscosity; ρ, density; σ, coefficient of surface tension.

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PROBLEM OF DESCRIBING THE FLOW FIELD IN AN INJECTOR MIXING CHAMBER

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The use of a three-component flow model is proposed. The results of numerical calculations are presented and compared with the experimental results.

In order to describe the one-dimensional flow field in the mixing chamber of a condensing injector, the model of a continuous liquid jet has been used [1, 2], where it was supposed that the jet of liquid issuing from the liquid nozzle is not disintegrated, but remains complete in the entire part of the mixing chamber investigated. The numerical calculations, carried out on the basis of this model, have provided satisfactory agreement with experiment only in the case of heat-transfer coefficients from vapor to liquid which are orders of magnitude higher than the well-known values for film condensation. The latter means that the continuous liquid-jet model significantly reduces the surface of interaction of the phases. Since, in fact, a significant atomization of the liquid jet almost always is observed, it will be advantageous to revert to flow models where atomization is taken into account.

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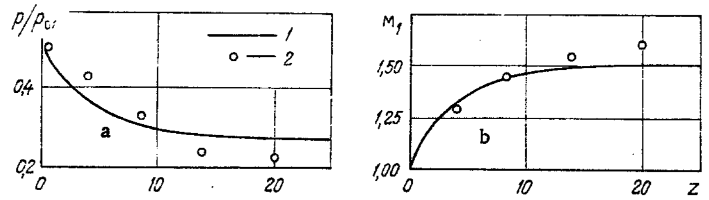

Fig. 1. Variation of static pressure ($p_{01} = 1.54$ bar) (a) and of Mach number of the vapor phase (b): 1) calculation; 2) experiment [2], z, mm.

In [3], a model of continuous atomization of a liquid jet was proposed, based on experimental investigations [4-6]; here a model of two-component flow has been used – vapor and liquid; there was no differentiation mode between the parameters of the liquid in the mixing zone and those in the core of the liquid jet. More strictly, the parameters of the liquid phase in the core and those of the liquid phase in the mixing zone were unconditionally differentiated, which is done also below.

We shall assume the following:

a) The flow field can be divided arbitrarily into the "core" of the flow of liquid – a continuous liquid jet – and the "mixing zone" – the region of two-phase dispersion-droplet flow;

b) continuous atomization of the core of the liquid jet and entrainment of the drops in the mixing zone take place.

We shall assume the flow to be steady and one-dimensional. We shall distinguish three component flows, designating them by the following subscripts: 1 – the vapor phase; 2 – the liquid in the core; and 3 – the liquid in the mixing zone. We shall suppose that these components are continuously distributed along the length and that their parameters are differentiable functions. It is possible that the viscosity and thermal conductivity are significant only in phase-interaction processes.

Using the methods of continuum mechanics for two-phase flows [7, 8], the flow being considered (in the region which does not contain discontinuities) is described by the system of equations of conservation:

$$\frac{1}{F}\cdot\frac{d}{dz}(\rho_1 w_1 F) = -g_{12} - g_{13}, \tag{1}$$

$$\frac{1}{F}\cdot\frac{d}{dz}(\rho_2 w_2 F) = g_{12} - g_{23}, \tag{2}$$

$$\frac{1}{F}\cdot\frac{d}{dz}(\rho_3 w_3 F) = g_{13} + g_{23}, \tag{3}$$

$$\rho_1 w_1 \frac{dw_1}{dz} = -\alpha_1 \frac{dp}{dz} - g_{12}(w_{12} - w_1) - g_{13}(w_{13} - w_1) - f, \tag{4}$$

$$\rho_2 w_2 \frac{dw_2}{dz} = -\alpha_2 \frac{dp}{dz} + g_{12}(w_{12} - w_2) - g_{23}(w_{23} - w_2), \tag{5}$$

$$\rho_3 w_3 \frac{dw_3}{dz} = -\alpha_3 \frac{dp}{dz} + g_{13}(w_{13} - w_3) + g_{23}(w_{23} - w_3) + f, \tag{6}$$

$$\rho_1 w_1 \frac{d}{dz}\left(i_1 + \frac{w_1^2}{2}\right) = -g_{12}\left(i_{12} - i_1 + \frac{w_{12}^2 - w_1^2}{2}\right) - g_{13}\left(i_{13} - i_1 + \frac{w_{13}^2 - w_1^2}{2}\right) + A_1 - q, \tag{7}$$

$$\rho_2 w_2 \frac{d}{dz}\left(i_2 + \frac{w_2^2}{2}\right) = g_{12}\left(i_{12} - i_2 + \frac{w_{12}^2 - w_2^2}{2}\right) - g_{23}\left(i_{23} - i_2 + \frac{w_{23}^2 - w_2^2}{2}\right), \tag{8}$$

$$\rho_3 w_3 \frac{d}{dz}\left(i_3 + \frac{w_3^2}{2}\right) = g_{13}\left(i_{13} - i_3 + \frac{w_{13}^2 - w_3^2}{2}\right) + g_{23}\left(i_{23} - i_3 + \frac{w_{23}^2 - w_3^2}{2}\right) + A_3 + q. \tag{9}$$

There are also the equations of state of the phases

$$p = R_1 \rho_1^0 T_1; \quad \rho_2^0 = \rho_3^0 = \text{const}, \tag{10)-(12}$$

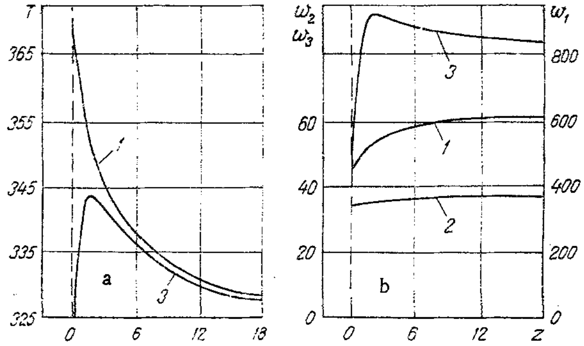

Fig. 2. Variation of temperature of components (a) and change of velocity of components (b): 1) vapor phase; 2) liquid in core of jets; 3) liquid in mixing zone; w, m/sec; z, mm.

and also

$$\alpha_1 + \alpha_2 + \alpha_3 = 1. \tag{13}$$

If the right-hand sides of Eqs. (1)-(9) are expressed in terms of the flow parameters, then system (1)-(13) can be solved by one of the numerical methods. If the principal determinant of system (1)-(13) is nonvanishing in the region being investigated, then the required solution will be unambiguous and will be found by direct numerical integration. Equating to zero the principal determinant of system (1)-(12), we obtain

$$\Delta = \rho_2^0 \alpha_1 w_3^2 (w_1^2 - a_1^2) - \rho_1^0 w_1^2 a_1^2 \left(\alpha_3 + \alpha_2 \frac{w_3^2}{w_2^2} \right) = 0. \tag{14}$$

Since $\rho_1^0(\alpha_2 + \alpha_3)/(\rho_2^0 \alpha_1) \ll 1$, it follows from Eq. (14) that $w_1^2 \approx a_1^2$. If in the region being considered (from the mixing chamber inlet to the discontinuity in the injector) there is a strictly supersonic flow of vapor, then there is no point $\Delta = 0$.

The numerical solution has been carried out, with account taken of condensation, only in the mixing zone; condensation on the surface of the jet was neglected in view of the insignificance of the jet surface in comparison with the surface of separation of the phases in the mixing zone. The flow density of the condensing substance was determined by the relation [9]

$$g_e = \frac{2\beta}{2-\beta} \left[\frac{p}{(2\pi R_1 T_1)^{1/2}} - \frac{p_{eq}(T_3)}{(2\pi R_1 T_3)^{1/2}} \right], \tag{15}$$

where the coefficient of condensation β was assumed to be equal to unity on the basis of [10].

In accordance with [3], the interaction of the phases f and q, not associated with mass transfer, was neglected. The parameters of the masses being transferred were assumed to be as follows:

$$w_{12} = w_{13} = w_1, \quad w_{23} = w_2, \quad i_{12} = i_{13} = i_1, \quad i_{23} = i_2.$$

The system of basis equations is simplified and assumes the form

$$\frac{1}{F} \cdot \frac{d}{dz} (\rho_1 w_1 F) = -g_{13}; \quad \frac{1}{F} \cdot \frac{d}{dz} (\rho_2 w_2 F) = -g_{23};$$

$$\frac{1}{F} \cdot \frac{d}{dz} (\rho_3 w_3 F) = g_{13} + g_{23};$$

$$\rho_1^0 w_1 \frac{dw_1}{dz} = -\frac{dp}{dz}; \quad \rho_2^0 w_2 \frac{dw_2}{dz} = -\frac{dp}{dz};$$

$$\rho_3 w_3 \frac{dw_3}{dz} = -\alpha_3 \frac{dp}{dz} + g_{13}(w_{13} - w_3) + g_{23}(w_{23} - w_3);$$

$$\frac{d}{dz}\left(i_1 + \frac{w_1^2}{2} \right) = 0; \quad \frac{d}{dz}\left(i_2 + \frac{w_2^2}{2} \right) = 0;$$

$$\rho_3 w_3 \frac{d}{dz}\left(i_3 + \frac{w_3^2}{2}\right) = g_{13}\left(i_{13} - i_3 + \frac{w_{13}^2 - w_3^2}{2}\right) + g_{23}\left(i_{23} - i_3 + \frac{w_{23}^2 - w_3^2}{2}\right). \qquad (16)$$

From this last system, with account taken of the relation $i_3 = u_3 + p/\rho_3^0$, we obtain the following: $u_3 = \text{const}$ and $T_3 = \text{const}$. This would also be expected with the assumed simplification.

In order to describe g_{23}, the empirical relation from [1] was used, defining the length of the complete atomization of the liquid jet in the mixing chamber:

$$l_c = 0.0927\left(\frac{m_1 w_1}{m_2 w_2}\right)^{-0.181}, \qquad (17)$$

which is verified in the region $1 \leq (m_1 w_1)/(m_2 w_2) \leq 8$. By varying the inlet temperature of the liquid in [1], it was found that the effect of the intensity of condensation on the disintegration of the liquid jet is insignificant.

The inclusion of Eq. (17) allows a different nature of change of the transverse section of the jet from the inlet size down to zero for $z = l_c$. It was assumed for the numerical calculation that the diameter of the core of the liquid jet changes linearly in the section $[0, l_c]$.

The flow in the mixing zone was assumed to be monodispersed. The closing components assume the form

$$g_{13} = g_e S_{13};\quad S_{13} = n\pi\delta^2;\quad \alpha_3 = \frac{1}{6}\, n\pi\delta^3;$$
$$\frac{\rho_1^0 (w_1 - w_3)^2 \delta}{\sigma} = \mathrm{We}_{\mathrm{cr}}. \qquad (18)$$

The critical value of the Weber number was assumed to be equal to 14, which is one of its popular values.

A comparison with experiment was carried out for a mixing chamber of constant cross section [2].

Figure 1a, b shows the calculated values of the static pressure and Mach numbers of the vapor phase, compared with the experimental values from [2]. The changes of the parameters are given here right up to the static pressure reaching a stabilized value.

Figure 2a shows the calculated values of the temperatures of the vapor and liquid in the mixing zone, and Fig. 2b shows the change of velocity of the components. The velocity and temperature of the drops do not vary monotonically: first of all, an intense increase of these parameters is observed, which then declines, changing to a decrease with subsequent arrival at stabilized values. This nature of behavior of the velocity and temperature of the drops can be explained in the following way. The droplets, separated from the core of the liquid jet, are intensely accelerated and are heated up by the vapor: the droplets are very small, with a size of the order of tens of micron; therefore, transfer processes proceed intensively. But in the mixing zone, all the large and lower-velocity, colder droplets from the core are atomized, which leads to a reduction of the average velocity and temperature of the liquid in the mixing zone.

The agreement between calculation and experiment in this case can be assumed to be satisfactory. If necessary, the calculation can be refined by taking account of condensation at the surface of the jet.

NOTATION

ρ_j^0, true density of the j-th component; α_j, volumetric content of the j-th component; ρ_j, average density, $\rho_j = \alpha_j \rho_j^0$; p, static pressure; w, velocity; T, absolute temperature; a_1, velocity of sound in the vapor phase; $M_1 = w_1/a_1$, Mach number of vapor; i, specific enthalpy; u, internal energy; f, q, g_{ij}, strengths of distributed sources of mechanical interaction and heat and mass transfer from the i-th component to the j-th component, respectively; i_{ij}, w_{ij}, parameters of transferred masses; A_j, terms reflecting the distribution of dissipated mechanical energy; F, cross-sectional area of channel; z, longitudinal coordinate; m, mass flow rate per sec; $p_{eq}(T_3)$, equilibrium pressure with respect to temperature of the liquid; δ, diameter of drop; σ, surface tension of liquid phase; n, concentration of drops.

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MEASUREMENT OF GAS-FLOW TEMPERATURES BY TWO THERMOCOUPLES

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A method for determination of the true temperature of a gas flow by two thermocouples is proposed. Results of an experimental verification are presented.

To reduce errors in measurement of gas-flow temperatures, either a thermocouple with very fine conductors (0.05 mm) or several thermocouples with conductors of different diameter are used [1]. Below, we offer a more accurate method of measurement with two thermocouples.

Let the hot junction of the thermocouple have the shape of a sphere with diameter d and with a conductor diameter equal to 2r. The two metals or alloys forming the thermocouple junction have coefficients of thermal conductivity λ_1 and λ_2, respectively. We assume that the conductors are located in an insulating medium of thickness δ, with coefficient of thermal conductivity λ_3.

Upon placement of the thermocouple junction in a high-temperature gas flow it will be heated by convective heat transfer. At the same time, the junction will be cooled by loss of heat through radiation and thermal conductivity of the free ends of the thermocouple. Obviously, for the steady state

$$P_c = P_r + P_t, \tag{1}$$

where P_c is the power conveyed to the junction by convective heat transfer; P_r is the power lost by the junction through radiation; and P_t is the power lost through thermal conductivity.

The power supplied by convective heat transfer is determined by Newton's formula

$$P_c = \alpha_1 (T_g - T_1) S, \tag{2}$$

where α_1 is the coefficient of convective heat liberation; T_g is the temperature of the gas flow, °K; T_1 is the junction temperature, according to its calibration, °K; and S, is the area of the heat-exchange surface, πd^2.

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