CALCULATING THE HYDRAULIC RESISTANCE OF FOAM FLOWS IN TUBES

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Formulas are derived for calculating the hydraulic resistance factor of a foam flow for isothermal flow in a tube with allowance for compressibility, biphasality, and a change in the structure of the foam during its movement.

In the movement of a foam flow in a horizontal tube under the influence of a longitudinal pressure gradient dp/dL, the energy involved in the displacement of the medium is expended on acceleration, changing the free phase boundary of the flow, and overcoming friction.

In the general case for a foam flow, due to the specific features of the latter (compressibility, biphasal nature, and variable specific interphase surface s), none of the terms entering into the energy balance equation can be ignored. However, there is no doubt that the energy expended on overcoming friction will be a significant portion of the total energy balance due to the high structural viscosity of a foam flow.

Determining the energy loss due to friction and the associated hydraulic resistance factor λ is an important problem in the study of foam flows, since knowledge of the hydraulic resistance is necessary in designing jet-mixing type foam generators.

The coefficient λ during foam movement was determined in [1, 2] from the Darcy—Weisbach relation without allowance for acceleration and changes in the phase boundary, since it was assumed that $dp/dL = dp_{fr}/dL$. Existing models for calculating the loss to friction for two-phase flows, such as the homogeneous model and the model of Lockhart—Martinelli [3], do not give a good agreement with the empirical data since they do not consider the features of the foams.

Compressibility was first accounted for in [4] in determining λ for a two-phase flow (water—air type) at high concentrations of the liquid phase (>10% by volume).

In the present work, we propose to experimentally substantiate calculating the hydraulic resistance factor λ in the cross section of a tube from measured hydrodynamic (p, Δp, u) and fundamental structural (K, $\bar{d}$, s) parameters of a foam flow.

Let us write the energy equation for a one-dimensional horizontal foam flow with terms which account for acceleration and the energy losses due to a change in the free phase boundary:

$$\frac{dp}{\gamma} + d\left(\frac{u^2}{2g}\right) + \frac{dE}{\gamma} + \lambda \frac{u^2}{2g} \frac{dL}{D} = 0. \tag{1}$$

It may be assumed that the phases do not slide relative to each other during movement of the foam ($u' = u'' = u$) and that the foam is quasihomogeneous and isothermal. Since the change in the free surface between the phases during the movement of the foam is considered first, let us examine the third term in Eq. (1) in greater detail.

Due to the presence of surfactants in the liquid phase, the foam flow takes the form of a highly dispersed bubbled structure with a large specific interphase surface $s \sim 10^5$–10^6 m^2/m^3. With isothermal expansion of the flow, there is an increase in the expansion ratio K and the weighted mean diameter of the bubbles $\bar{d}$. A simultaneous change in these two parameters of the flow leads to a change in s, and a certain amount of energy is expended on this change in s.

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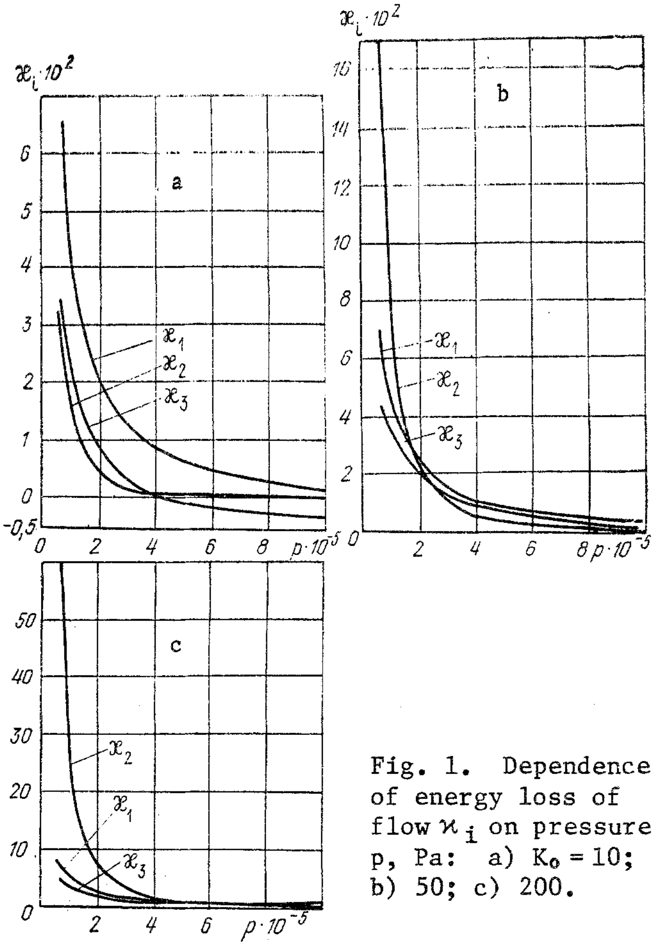

Fig. 1. Dependence of energy loss of flow $\varkappa_i$ on pressure p, Pa: a) $K_0 = 10$; b) 50; c) 200.

Taking $s = \frac{6}{d}\frac{K-1}{K}$ (see [5], for example) and assuming that the relation $\bar{d} = d_0\sqrt[3]{p_0/p}$ is valid for isothermal expansion, we may determine the expenditure of the energy of the flow on changing the free interphase surface of the foam structure from the following expressions

$$\frac{dE}{\gamma} = \frac{\sigma ds}{\gamma} = \frac{6\sigma}{\bar{d}_0\sqrt[3]{p_0}}\,\frac{d\left(\sqrt[3]{p}\,\frac{K-1}{K}\right)}{\gamma}. \tag{2}$$

Using the continuity equation of the flow $u\gamma F = G$ and the equation of state for the gas phase $p/\gamma'' = RT$, it is not difficult to express current values of γ, K, and u through the pressure p in a given cross section and the quantities K_0 and γ_0'' at the outlet of the flow from the tube.

Substituting Eq. (2) in (1) along with the values of γ, K, and u expressed through p, after transformation we obtain a differential equation connecting p with L:

$$C\,\frac{pdp}{\alpha+p} - 2\alpha D\,\frac{dp}{p(\alpha+p)} + B\,\frac{\sqrt[3]{p}\,(\alpha-2p)\,dp}{(\alpha+p)^3} = -\lambda dL, \tag{3}$$

where

$$C = \frac{g\pi^2 D^5}{8G^2}\,[\gamma' + (K_0-1)\gamma_0'']; \quad \alpha = (K_0-1)\,p_0;$$

$$B = \frac{g\pi^2 D^5 \sigma\alpha}{4\bar{d}_0\sqrt[3]{p_0}\,G^2}\,[\gamma' + (K_0-1)\gamma_0''].$$

Integration of Eq. (3) within the limits of the given section of tube of length L gives the following expression for λ after the appropriate transformations:

$$\lambda = \varkappa\lambda_f = [1 - \varkappa_1 - \varkappa_2 - \varkappa_3]\,\lambda_f, \tag{4}$$

where

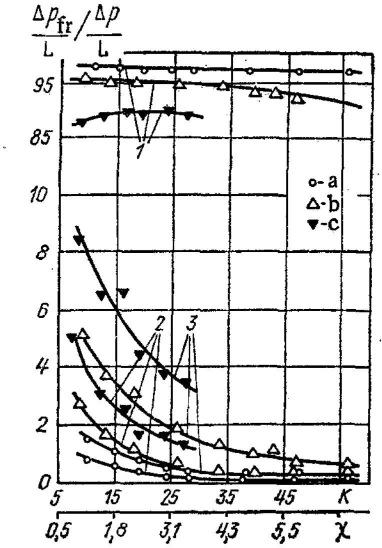

Fig. 2. Comparison of results of calculations for friction: 1) foam model; 2) homogeneous model; 3) Lockhart–Martinelli model; a) u = 1.6 m/sec; b) 4.3; c) $\frac{\Delta P_{fr}}{L} \Big/ \frac{\Delta P}{L}$, %; χ, wt. %; K) dimensionless quantity.

$$\varkappa_1 = \frac{\alpha}{2p(\alpha + p)}\Delta p; \quad \varkappa_2 = \frac{\alpha}{p(\alpha + p)}\,\frac{\gamma u^2}{g};$$

$$\varkappa_3 = \frac{2\sigma}{\bar{d}_0 \sqrt[3]{p_0}}\,\frac{\alpha(\alpha - 2p)}{\sqrt[3]{p^2}(\alpha + p)^2}; \quad \lambda_f = \frac{2gD\Delta p}{\gamma u^2 L}.$$

Let us analyze the quantitative contribution of each term entering into the factor $\varkappa$ with λ_f in Eq. (4) for specific parameters of a foam flow that might be realized in an experiment: $G = 8.16$ N/sec, $\sigma = 72 \cdot 10^{-3}$ N/m, $\bar{d}_0 = 0.04$ mm, $D = 50$ mm, $\Delta p = 9.8 \cdot 10^3$ Pa. Figure 1 shows calculated values of $\varkappa_1$, $\varkappa_2$, and $\varkappa_3$ for these flow parameters in relation to p at $K_0 = 10$, 50, and 200. It is apparent that the change in $\varkappa_1$, $\varkappa_2$, and $\varkappa_3$ is greater, the lower the pressure of the flow and the higher the value of K_0. Within the investigated pressure interval $p = 10^5$–10^6 Pa, the quantity $\varkappa_3$ changes its sign at $K_0 = 10$ (Fig. 1a). For $K_0 = 50$ and 200, the quantity $\varkappa_3$ changes sign at $p > 10^6$ Pa. The change in the sign of $\varkappa_3$ indicates that the value of s may decrease with movement of the foam, given a foam with certain hydrodynamic parameters. Thus, part of the energy of the flow is liberated, and then dissipated.

Thus, the calculations established that, generally, in empirically determining λ it is necessary to consider the features of foam flows. These features are accounted for by the terms $\varkappa_1$, $\varkappa_2$, and $\varkappa_3$, determining the portion of the flow energy expended on compression, acceleration, and changing the free surface, respectively.

Figure 2 shows experimental data for friction obtained by the proposed method (curves 1) and data calculated in accordance with well-known models of two-phase flows — the homogeneous model (curves 2) and the Lockhart–Martinelli model (curves 3) — for the same initial data.

It is apparent from Fig. 2 that there is substantial diagreement between the empirical and theoretical results. This is due to the fact that the foam is a structured two-phase system, the high viscosity of which leads to high values of friction during movement. This latter circumstance is not taken into account by the well-known theoretical models and shows that they should not be used to calculate friction in the movement of foams.

NOTATION

p, pressure in a cross section of the tube; Δp, total pressure drop on a section of the tube; Δp_{fr}, pressure drop due to friction on a section of the tube; u, mean flow rate; γ, specific gravity; K, expansion ratio; T, flow temperature; R, universal gas constant; G, weight flow rate; χ, weight gas flow rate; $\bar{d}$, weighted mean diameter of bubble; σ, surface tension coefficient; D, tube diameter; L, length of section of tube; F, cross-sectional area of tube; Indices: ', liquid phase; ", gas phase; 0 refers to atmospheric conditions for $p = 9.8 \cdot 10^4$ Pa.

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TIME TO COOL A CRYOGENIC OBJECT BY A GASEOUS CRYOGENIC AGENT

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Dependences to compute the cooling time of a single-channel object are obtained theoretically and confirmed experimentally.

The heat-transfer—flow-through part of modern cryogenic apparatus is a system of channels with $L/d > 1000$ as a rule. Cooling such apparatus, i.e., reducing the temperature from the initial to the working value, is performed by a single-phase cryogenic agent. The cooling time here can reach several days. The purpose of the present paper is to determine the cooling time and to estimate the parameters influencing its value most strongly.

The problem can be formulated as follows: A cryogenic agent whose temperature does not vary during the cooling process goes into a channel with a constant initial temperature. Determine after what time the end of the channel takes on the temperature of the cryogenic agent.

The solution applied to a steam-generating channel, obtained in [1, 2], can be used to find the cooling time. The dynamic process is described in [1, 2] by two energy equations

$$\frac{Dc_p}{\alpha F_s}\frac{\partial T}{\partial \tau}+\frac{Gc_p}{\alpha U}\frac{\partial T}{\partial z}=\Theta-T,$$
$$-\frac{Mc_m}{\alpha F_s}\frac{\partial \Theta}{\partial \tau}=\Theta-T \qquad (1)$$

with the boundary conditions

$$\Theta(z,\ 0)=T(z,\ 0)=T_0,\ T(0,\ \tau)=T_{\text{in}}=\text{const}.$$

By introducing the new independent variables

$$\zeta=\frac{z\alpha F_s}{LGc_p},\quad \eta=\frac{\tau-z/W}{Mc_m}\alpha F_s \qquad (2)$$

we obtain the solution of system (1) in the form of infinite series in ζ and η. A singularity of this solution is the determination of the finite value of the time for any point of the channel to reach the working temperature. The complex form of the solution forced the authors of [1, 2] to tabulate the dimensionless temperature T/T_{in} for a broad range of values

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