use of heat pipes with a given combination of materials of the wall, capillary-porous structure and heat-transfer agent, under constant operating conditions and under variable conditions.

NOTATION

pH, hydrogen indicator; c_{O_2}, concentration of oxygen in the heat-transfer agent; Re_r, radial Reynolds number; Re_a, axial Reynolds number; U, dimensionless axial velocity component of the vapor flow; $\tilde{B}$, geometric parameter; C, dimensionless axial coordinate; Y, dimensionless radial component; $\tilde{X}$, axial coordinate; $\tilde{Y}$, radial coordinate; d, r, inside diameter and radius of the heat pipe, respectively; L, length of the condensation zone; L_p, length of the plug of noncondensible gas; L_{act}, length of the active part of the condensation zone; τ, time; V, volume; X*, coordinate of the start of reverse flow; S, area of the corroding surface; t, temperature; m_{H_2}, mass of hydrogen liberated. Subscripts: cyl, cylinder; vgf, vapor—gas front.

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RATE OF SURFACING OF A GAS PLUG IN ANNULAR AND RECTANGULAR VERTICAL CHANNELS

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UDC 532.529

Relations are proposed for the limiting surfacing velocities of plugs used within a broad range of geometric parameters.

When constructing the diagram depicting two-phase flow regimes for calculation of plug flow in vertical channels of different shapes, it is necessary to know the rate of surfacing of a single gas plug in a capped pipe or the rate of descent of fluid at which the plug is suspended in the channel.

The limiting (steady-state) surfacing velocity of a gas plug is determined by the hydrodynamics in the flow of the fluid around the frontal part and does not depend on the length of the plug. When inertial and buoyancy forces predominate, this velocity is determined by the relation [1]:

$$\mathrm{Fr}(l) = \frac{U_\infty \sqrt{\rho'}}{\sqrt{g(\rho' - \rho'')l}} = A_1, \tag{1}$$

where l is a characteristic linear dimension of the channel, equal to the diameter for a pipe, the external diameter for an annular channel, the diameter of the shell for a rod assembly, and the width (of the larger side) for a rectangular channel; A_1 is an empirical coefficient determined from Fig. 1a.

The data in [1] for A_1 in rectangular channels was used in [2] to propose the linear approximation

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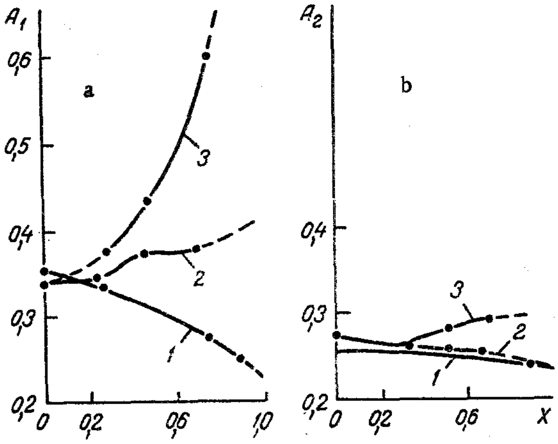

Fig. 1. Dependence of the coefficients A_1 and A_2 on the channel geometry: 1) rectangular channel, $X = 1 - \delta/b$; 2) annular channel, $X = D_2/D_1$; 3) rod bundle, $X = 1 - D_{eqv}/D_{sh}$.

$$A_1 = 0.23 + 0.13\delta/b. \tag{2}$$

Similarly, the constant A_1 for annular channels, represented in the form of a function of the ratio D_2/D_1:

$$A_1 = 0.345 + (0.23\sqrt{\pi} - 0.345)\,D_2/D_1, \tag{3}$$

also agrees well with the data in [1].

The values 0.23 and 0.345 were obtained on the basis of an approximate analytical solution of the problem of potential flow about an infinite plug in plane and circular vertical channels [3, 4].

In a rectangular channel with $\delta/b \ll 1$, the plug surfacing velocity will be the same as in an annular channel if the geometric dimensions of the two channels are related as follows:

$$b = \pi(D_1 + D_2)/2,\ \delta = (D_1 - D_2)/2. \tag{4}$$

The surface forces change U_∞ by no more than 3% for circular pipes (in the absence of the effect of viscous forces) if $Bo(D_0) > 6.2$ [2]. We will determine the corresponding condition for rectangular and annular channels. We will assume that the characteristic linear dimension of the channel λ, determining the effect of the surface forces, is a linear function of D_1 and D_2 in an annular channel. Then three variants are possible for λ, giving the asymptotic for the characteristic pipe dimension: $\lambda_1 = D_1 - D_2$, $\lambda_2 = D_1$ and $\lambda_3 = D_1 + D_2 = \Pi/\pi$. For a rectangular channel, accordingly, we have $\lambda_1 = D_{eqv}$, $\lambda_2 = b$ and $\lambda_3 = \Pi/\pi$. Considering that the condition of the absence of surface forces (<3%) is determined by the inequality $Bo(\lambda) > 6.2$ for any channel geometry and taking into account that the effect of the surface forces on U_∞ was not indicated in experiments conducted in [1] (3% error) for the numbers $Bo(\lambda_1) \geqslant 2.6$, $Bo(\lambda_2) \geqslant 4.8$ and $Bo(\lambda_3) \geqslant 11.9$, we find that $\lambda \equiv \lambda_3$. Then Eqs. (1)-(3) are valid if

$$Bo(\Pi/\pi) > 6.2. \tag{5}$$

Let us examine another restriction on the use of Eqs. (1)-(3). In accordance with Eqs. (1) and (2), an increase in the width of a rectangular channel should be accompanied by an infinite increase in the surfacing velocity of the plug. It is natural to assume that this is impossible and that with an increase in channel width, the effect of the latter on U_∞ will diminish and the plug surfacing velocity will no longer depend on the clearance. To describe this transition, we will use experimental data from [5] on the rate or surfacing of single large bubbles in a channel with plane-parallel walls filled with a fluid which wets the walls. According to this data, the bubble surfacing velocity U_∞^b increases with an increase in its diameter in proportion to V (V is the volume of the bubble) for bubbles with $D/\delta \ll 40$. At $D/\delta > 40$, the effect of bubble diameter diminishes, while the authors propose the following relations for U_∞^b

$$Fr(\delta) = \begin{cases} 6.2\,Bo(\delta) & \text{at} \quad Bo < Bo_0, & (6)\\ 6.2\,Bo_0 & \text{at} \quad Bo \geqslant Bo_0, & (7)\end{cases}$$

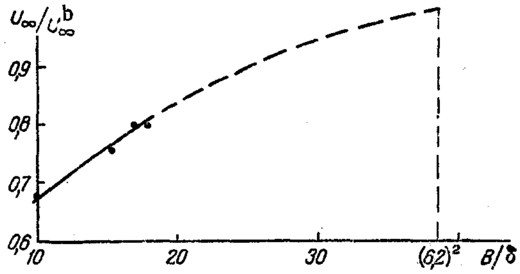

Fig. 2. Comparison of the plug surfacing velocity in a rectangular channel and the bubble surfacing velocity in a plane-parallel channel.

where $Bo_0 = 0.23$.

The comparison shown in Fig. 2 was obtained on the basis of data from [1, 5] under the condition of equality of the width of the rectangular channel to the bubble diameter. Equation (1) was validated by the author of [1] for $b/\delta \leqslant 18$, and the dashed line in Fig. 2 was constructed on the basis of the assumption that Eq. (1) remains valid for $b/\delta > 18$. It can be seen from the figure that the plug surfacing rate is less than the bubble surfacing rate, which is related to the effect of smaller lateral walls. The effect of the walls decreases with an increase in b/δ, and the ratio of the velocities approaches unity at the same values of b/δ at which the effect of bubble diameter on the rate of its surfacing decreases. This allows us to propose that the plug surfacing velocity in a ractangular channel for $b/\delta > 40$ will be determined by Eqs. (6) and (7).

Then reducing the expressions determining U_∞ in a rectangular channel to a form analogous to (7), we obtain the following formulas from (1), (2), (6), and (7) for U_∞:

$$\mathrm{Fr}(\delta) = \begin{cases} \sqrt{b/\delta}\,\mathrm{Bo}(\delta) \text{ at } \sqrt{b/\delta} < 6{,}2;\ \mathrm{Bo} < \mathrm{Bo}_1, & (8) \\ \sqrt{b/\delta}\,\mathrm{Bo}_1 \text{ at } \sqrt{b/\delta} < 6{,}2;\ \mathrm{Bo} \geqslant \mathrm{Bo}_1, & (9) \\ 6{,}2\,\mathrm{Bo}(\delta) \text{ at } \sqrt{b/\delta} \geqslant 6{,}2;\ \mathrm{Bo} < \mathrm{Bo}_1, & (10) \\ 6{,}2\,\mathrm{Bo}_1 \text{ at } \sqrt{b/\delta} \geqslant 6{,}2;\ \mathrm{Bo} \geqslant \mathrm{Bo}_1, & (11) \end{cases}$$

where $Bo_1 = 0.23 + 0.13\delta/b$.

These formulas are valid if $Bo(\Pi/\pi) > 6.2$. This condition is incompatible with the restrictions limiting the range of application of Eq. (8), but we will keep this equation because it makes it possible to understand the principle underlying the construction of the remaining formulas.

It follows from (3), (4), and (9) that the surfacing velocity for square and circular channels will be determined by the equations:

$$\mathrm{Fr}(b) = \mathrm{Bo}_1, \tag{12}$$

$$\mathrm{Fr}(D_0) = \sqrt{\frac{\pi}{2}}\,\mathrm{Bo}_1. \tag{13}$$

To reduce these equations to forms with analogous right sides (without coefficients of the type $\sqrt{\pi/2}$), it suffices to take half the wetted perimeter as the characteristic dimension for the inertial forces. Then instead of Eq. (1) we obtain

$$\mathrm{Fr}(\Pi/2) = A_2, \tag{14}$$

where A_2 is determined from Fig. 1b. The following linear approximation is acceptable for A_2 in the case of annular and rectangular channels:

$$A_2 = 0.23 + 0.045\,(1 - D_2/D_1), \tag{15}$$

$$A_2 = 0.23 + 0.025\delta/b. \tag{16}$$

The introduction of a new characteristic linear dimension which is useable for any geometry is also justified by the fact that A_2 is less dependnet on channel geometry than A_1 and has roughly the same values for different channel geometries. The strong dependence of A_1 on X for rod bundles, constructed and applicable only for seven rods (for one rod, curve 3 in Fig. 1a should be replaced by curve 2), may be the reason for the large error in the determination of the surfacing velocity for X > 0.8. This shortcoming is eliminated with the introduction of the new characteristic dimension, and system (8)-(11) is changed to the following form:

$$\mathrm{Fr}(\Pi/2)=\begin{cases} Z\,\mathrm{Bo}(\delta) \text{ at } Z\leqslant 1;\ \mathrm{Bo}(\delta)<\mathrm{Bo}_2, & (17)\\ Z\,\mathrm{Bo}_2 \text{ at } Z\leqslant 1;\ \mathrm{Bo}(\delta)>\mathrm{Bo}_2, & (18)\\ \mathrm{Bo}(\delta) \text{ at } Z>1;\ \mathrm{Bo}(\delta)<\mathrm{Bo}_2, & (19)\\ \mathrm{Bo}_2 \text{ at } Z>1;\ \mathrm{Bo}(\delta)>\mathrm{Bo}_2, & (20)\end{cases}$$

where $\mathrm{Bo}_2 = A_2$, $Z = 6.2\sqrt{2\delta/\Pi}$.

Here, A_2 becomes the boundary value of the number $\mathrm{Bo}(\delta)$, with the effect of surface forces diminishing above this value (if condition (5) is satisfied, of course).

NOTATION

ρ', ρ'', density of liquid and gas; σ, surface tension; g, acceleration due to gravity; D_0, diameter of pipe, D_1 and D_2, external and internal diameters of annular channel; b, δ, width and clearance of rectangular channel; D_{sh}, diameter of shell of rod bundle; U_∞, limiting plug surfacing velocity; l, linear dimension; $\mathrm{Bo}(l) = l/\sqrt{\sigma/g(\rho' - \rho'')}$, Bond number.

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THEORY OF GASLIFTS

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On the basis of a simplified model, a new method is proposed for the analysis of unsteady-state gaslifts, and instabilities of steady-state gaslift processes are demonstrated.

A considerable part of the petroleum which is produced is recovered by the use of various types of gaslifts for exploiting wells, and the importance of this method in the total volume of petroleum produced shows a clear tendency to increase. In this connection, the problem of optimizing gaslifts is becoming particularly important, since this is related to increasing the production of wells and decreasing their capital and operating costs; the solution of this problem is not possible without the effective modeling of the processes occurring in gaslifted wells. However, the existing methods of modeling and calculation of these processes are unsuitable for analyzing the significantly unsteady-state phenomena which occur in gaslifts. In addition, even under steady-state conditions they are not well adapted to explaining the distributions of the gas—liquid mixture in the ascending column of the well. In fact, these methods are based on semiempirical considerations of the steady-state regime only [1], and in principle they do not extend beyond the models proposed as much as a generation ago [2].

Under the conditions occurring in practice the gas lift process often appears to be unsteady-state in nature. Unsteadiness occurs in the process of starting up a well, and may also be introduced when the gaslifts are organized to operate batchwise [3]. In addition, the steady-state regime sometimes proves to be unstable, which leads ultimately to the generation of self-excited oscillations [4]. The author knows of only a single formalized approach

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