HYDRODYNAMIC CHARACTERISTICS OF A HORIZONTAL CHANNEL WITH THE MOTION OF A TWO-PHASE HELIUM FLOW

I. S. Mamedov, V. M. Miklyaev,
V. I. Pryanichnikov, and Yu. P. Filippov

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Experimental data is presented on the pressure drop in the motion of a two-phase helium flow in a horizontal pipe under adiabatic conditions and with a supply of heat.

It is necessary to know the hydrodynamic characteristics of horizontal channels containing helium flows in order to design extensive superconducting magnet systems, such as in an accelerator-storage complex [1]. However, very little experimental work has been done on this problem [2]. A relatively few works, similar in theme [3-6], have investigated these characteristics under different conditions — in vertical channels and spiral tubes. In connection with this, we undertook studies having the aim of obtaining the hydrodynamic characteristics of horizontal channels under adiabatic conditions and with a supply of heat. Another aim was to compare these results with well-known data for vertical and spiral-tube analogs.

The test stand is shown in Fig. 1. A flow of helium from a KhGU-250/4.5 unit [7] at a pressure of $1.2 \cdot 10^5$ Pa or more passes in succession through throttle valves 1 and 2, supply main 3, heat exchanger 4, three-way valve 5, one of the devices for measuring flow rate (diaphragm) 6, input vapor-content generator 7, transparent element 8 for visual observation of the structure of the flow [8], experimental section 9, ceramic electrically insulating insert 10, return main 11, and throttle valve 12. After this, the helium enters a KhGU-250/4 5 vessel 13, where it is separated into vapor and liquid. The vapor, in the form of a reverse flow, enters the KhGU-250/4.05, while part of the liquid from the vessel 13 is sent through the make-up main 14 to cryostat 15 with nitrogen shield 16. For the vapor, the cryostat can be connected with a gas holder either directly or through a vacuum pump 17. Due to the difference in pressures in the heat exchanger 4 and the cryostat 15, the helium flow condenses and subsequently cools to the state of a liquid subheated to saturation.

The experimental unit 9 includes a tube of steel 12Kh18N10T with $\varnothing 5.03 \times 0.21$ mm and a length of 2.5 m. Two meters of the length are taken up by the heated section, while 0.25 m are accounted for by the pre- and post-connection sections. The heated part is equipped with three chambers for sampling the pressure, the chambers being located 1 m apart. A line for electrical heating is connected to the two outermost chambers. The tube for the helium is surrounded by a multiple-layer insulation of metallized lavsan and a copper shield cooled with liquid nitrogen. The tube with its insulation is enclosed in an evacuated housing.

The helium temperatures ahead of the diaphragms T_1 and in the cryostat 15 were measured with carbon resistors made by the "Allen Bradley" company, while helium temperatures at the inlet T_2 and outlet T_3 of the experimental section were measured with germanium resistance thermometers [9]. Pressure at the inlet was determined with a standard manometer. The pressure drops on the test section were measured with an OM-2 optical manometer. Helium flow rate was determined with graduated diaphragms having limits of $0.65\text{-}4.2 \cdot 10^{-3}$ and $3.5\text{-}22.0 \cdot 10^{-3}$ kg/sec respectively. The parameters of the helium ahead of the test section were changed by means of the input vapor content generator 7 installed in the cryostat 15. Heat supply to the main 3 with the shield cooled with liquid nitrogen, which connects the cryostat 15 and the test section 9, was determined from the measurements of flow rate, the temperatures T_1 and T_2, and the pressures P_1 and P_2 for the case of flow of a liquid subheated to saturation.

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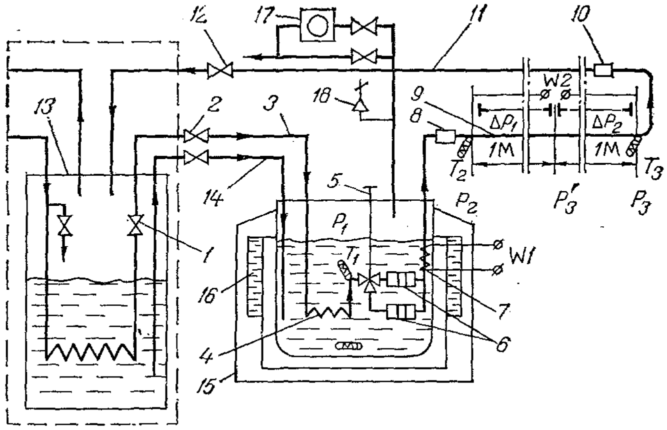

Fig. 1. Diagram of test stand.

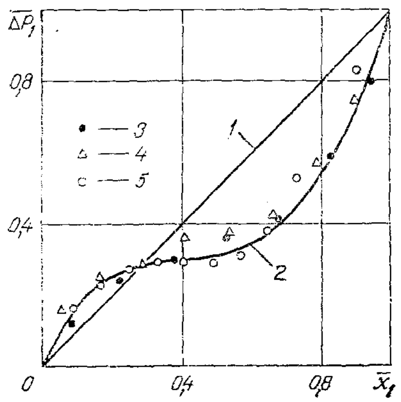

Fig. 2

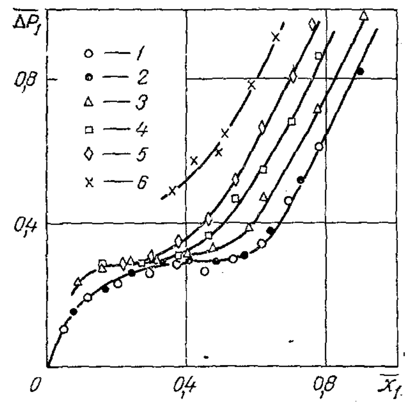

Fig. 3

Fig. 2. Comparison of the results with data for a vertical pipe [5, 6] under adiabatic conditions: 1) homogeneous model; 2) generalizing relation [6]; 3) m = 120 kg/m²·sec, P_3' = (1.31-1.33)·10^5 Pa; 4) 150, (1.31-1.35)·10^5 Pa; 5) 240 (1.34-1.41)·10^5 Pa.

Fig. 3. Comparison of the results (m = 120 kg/m²·sec) with data for a vertical pipe (m = 145 kg/m²·sec) [6] with a supply of heat: 1) q = 0, P_3' = (1.31-1.33)·10^5 Pa; 2) 360 W/m², (1.29-1.32)·10^5; 3) 720, (1.31-1.33)·10^5; 4) 1080, (1.31-1.32)·10^5; 5) 2150, (1.34-1.36)·10^5; 6) 0, data from [6]; 7) 1000 W/m², data from [6]; 8) 2000, data from [6].

Each series of tests to determine the drag of the channel was conducted with a fixed value for heat input, flow rate, and pressure at the outlet* and a variable flow-inlet enthalpy. The latter was calculated on the basis of the heat balance equation. The results obtained were represented in the form of dependences of the relative pressure drop ΔP_i = $(\Delta P_i - \Delta P')/(\Delta'' - \Delta P')$ on the mean relative enthalpy of the two-phase flow $x_i = (i_i - i')/(i'' - 1'$; with different heat fluxes q and mass velocities m. The value of $\Delta P'$ was determined experimentally, while the corresponding values of $\Delta P''$ were calculated from the relation $\Delta P'' = \Delta P'(\rho'/\rho'')$. For cases of flow of liquid helium, the experimental drag coefficients were 20% higher than the corresponding values found from Nikuradze's formula [10], which is valid for smooth tubes at $10^4 < Re < 10^8$. Comparison of the test data with the data obtained with the formula of Holbrooke and White in [6] for rough tubes showed that the equivalent surface roughness of the test section is about 1 μm, which agrees with [6]. The corresponding deviations do not exceed ±9%. The value of ΔP_i, empirically measured, consists mainly of two components — the pressure drops due to friction and acceleration of the two-phase flow.

The errors of $\bar{x}_1$ and $\Delta \bar{P}_1$ due to features of the experimental method and instruments used were about 11 and 5%, respectively. Figure 2 compares the results obtained for the first

*In tests with the above set-up, the pressure in the vessel 13 was held constant, this vessel being connected with the intake main of the compressor of the KhGU-250/4.5 units.

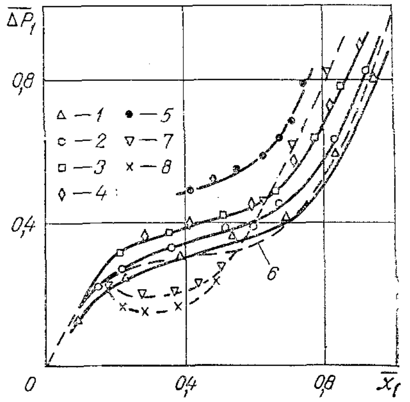

Fig. 4. Effect of the thermal load on the hydrodynamic characteristics of a horizontal pipe with m = 240 kg/m^2·sec: 1) q = 0, P_3' = (1.34-1.41)·10^5 Pa; 2) 360 W/m^2, (1.35-1.43)·10^5; 3) 720, (1.34-1.43)·10^5; 4) 1430, (1.35-1.45)·10^5, 5) 2150, (1.36-1.46)·10^5; 6) 3580, (1.40-1.48)·10^5 (the large values of P_3' correspond to large $\bar{x}_1$).

test section with the estimates from the equation for a homogeneous mixture and a relation which generalizes empirical data for a vertical pipe under adiabatic conditions [5, 6]. It follows from this figure that the relative pressure drops in the horizontal and vertical channels agree satisfactorily nearly throughout the entire range of vapor contents. Figure 2 also shows that this data is quite different from the calculated results for the homogeneous model. The agreement of the results obtained with other well-known theoretical relations can be judged by the fact that the comparison presented in [6] for a vertical pipe indicates the satisfactory agreement of the empirical data with the Martinelli–Nelson relation for water under high pressure (P = 200·10^5 N/m^2). Together with this, these empirical results are significantly less than the results calculated with the Lockhart–Martinelli equation [11].

Figure 3 shows the effect of heat flux on the hydrodynamic characteristics $\Delta\bar{P}_1(\bar{x}_1)$ with a mass velocity m = 120 kg/m^2·sec. It follows from the figure that the empirical relations for the heated channel are located above the curve characterizing flow under adiabatic conditions. Here, for $\bar{x}_1$ = idem, $\Delta\bar{P}_1$ increases with an increase in q. It should be noted that when q ≃ 1860-1880 W/m^2, flow rate fluctuates within ±10%. Meanwhile, the amplitude of these fluctuations increases with a further increase in q.

Figure 3 also shows a comparison of the data obtained with similar results for a vertical channel. Thus, for the latter with m = 145 kg/m^2·sec and 0 < $\bar{x}_1$ < 0.15-0.20, the effect of the thermal flow is little noticed [5], while it is more substantial for a horizontal tube. In the range 0.15 < $\bar{x}_1$ < 0.55 for a vertical channel, a hydraulic crisis is typical. This crisis is accompanied by a larger decrease in the pressure drop, the higher the heat flux. When $\bar{x}_1$ > 0.55, $\Delta\bar{P}_1$* is nearly independent of q, and the corresponding values of $\Delta\bar{P}_1$ somewhat exceed the pressure drop under adiabatic conditions.

Comparison of the data obtained with the experimental results for a heated coiled tube [4] (q = 600 W/m^2) shows a fairly significant deviation. The magnitude of this discrepancy can be judged from the fact that the data in [4] agrees satisfactorily with the estimates obtained from the equation for a homogeneous mixture.

An increase in mass velocity to 240 kg/m^2·sec leads to a situation whereby the pressure drop is nearly the same both for adiabatic flow and for q = 360 W/m^2 throughout the range of relative enthalpies (Fig. 4). In this case, a change in $\bar{x}_1$ from 0.15 to 0.55 leads to an increase in $\Delta\bar{P}_1$ by about 35%, while for m = 120 kg/m^2·sec (see Fig. 3) in the same interval of $\bar{x}_1$ the value of $\Delta\bar{P}_1$ increases almost 90% (for adiabatic conditions). This can be explained by different regimes of flow of two-phase helium [8].

*For the vertical channel, the value of $\Delta\bar{P}_1$ corresponds to the pressure drop due to friction.

At loads $q > 360$ W/m^2 and $\bar{x}_1 > 0.4$, the character of the relations $\Delta\bar{P}_1(x_1)$ for m = 240 kg/m^2·sec is qualitatively the same as for m = 120 kg/m^2·sec. However, in contrast to the results shown in Fig. 3, some of the test data for $x_1 < 0.4$, m = 240 kg/m^2·sec, q = 720 W/m^2 and q = 1430 W/m^2 (Fig. 4) is almost the same as the data for adiabatic conditions. This can be explained both by the fact that the flow regime is different at different m and by a certain difference in pressures in each series of tests, when m = 240 kg/m^2·sec and q = idem. It should also be noted that the fluctuations in flow rate for m = 240 kg/m^2·sec did not occur as heat flux was increased up to 5300 W/m^2.

Thus, in the motion of a two-phase helium flow, the hydrodynamic characteristics of a horizontal channel are quite different from those for vertical and coiled-tube channel in both the qualitative and quantitative sense. This difference becomes more pronounced as the thermal load increases.

NOTATION

m, mass velocity; P_1, P_2, P_3', P_3, pressures ahead of the flow meter, at the inlet of the test section, at the middle of this section, and at the end of the section; ΔP_i, $\Delta P''$, and $\Delta P'$, pressure drops corresponding to a two-phase flow saturated with vapor and liquid; $\Delta\bar{P}_i$, relative pressure drop, $\Delta\bar{P}_i = (\Delta P_i - \Delta P')/(\Delta P'' - \Delta P')$; i_i, i'' and i', enthalpies of a two-phase flow saturated with vapor and liquid; $\bar{x}_i$, relative enthalpy, $\bar{x}_i = (i_i - i')/(i'' - i')$; i - 1, 2, indices pertaining to the first and second sections; ρ'', ρ', densities of the saturated vapor and liquid; T_1, T_2, T_3, temperatures of the helium flow ahead of the flow meter and at the inlet and outlet of the test section; Reynolds number; q, heat flux referred to the inside surface of the channel.

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