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STABILITY OF A LOW-TEMPERATURE HELIUM FLOW IN HEATED CHANNELS

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The stability of a turbulent helium flow under forced convection is investigated in channels with different hydraulic characteristics.

The stability of a helium flow in heated channels must be investigated to determine the stable cooling conditions for different cryoenergetic apparatus in the 4-10°K temperature range at pressures up to 1.5 MPa.

The spontaneous origination of wall pressure and temperature fluctuations was observed in fluid heating for sub- and supercritical pressures in many experimental papers [1-3], say.

Among the different kinds of vibrational processes in fluids, the so-called density wave fluctuations are most widespread. Characteristic for them is the propagation of density, enthalpy, and mass-flow perturbations along a channel at the fluid flow velocity, which can damp out or grow with time for a definite relationship between the mass-flow and the thermal load.

The results of a number of theoretical and experimental investigations of such fluctuations in cryogenic systems cooled by supercritical-pressure helium have recently been published. The Nyquist frequency criterion based on the principle of an argument was used in [4] to analyze stability, and permitted estimation of a number of versions of cooling with the thermodynamic properties of the gas taken into account.

On the basis of a simplified model using an approximate description of the thermodynamic properties of helium in the near-critical region, an equation was obtained in [5] for the stability boundary in two dimensionless parameters $\psi_d = \Delta P_1/\Delta P_2$ is the ratio of the pressure drops in the input and output chokes, and $R = v_2/v_1$ is the degree of gas expansion. The pressure over the channel length was assumed constant in the computations.

The boundary of the beginning of the origination of vibrations by using analogous criteria was constructed earlier on the basis of experimental investigations [1]: the relative hydraulic drag

$$\psi = \frac{\Delta P_1 + \Delta P_{1K}}{\Delta P_{2K} + \Delta P_2} \qquad (1)$$

and the degree of expansion $R = (v_2 - v_1)/v_1$. Here, ΔP_1 is the pressure drop at the input throttle; ΔP_{1K} is the channel hydraulic drag between the input throttle and the section with the pseudocritical temperature T_m of the flow; ΔP_{2K} is the channel hydraulic drag between the section with temperature T_m and the output throttle, and ΔP_2 is the hydraulic drag of the output throttle.

A disagreement between the results of a computation [4] and the stability boundary obtained on the basis of experiments was noted in [1]. A discrepancy is also noticeable in comparing the results of computations [5] and test data, especially in the domain of small values of ψ.

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It must be noted that the problem of determining the stability boundary of systems cooled by fluids at near-critical pressure is multiparametric since the fluid properties in this domain depend substantially on the coolant pressure and temperature. The temperature and pressure distributions along the channel length depend, in turn, on the local hydraulic drags, the channel orientation in the gravity field, and the flow direction, the thermal load, the relationshp between the thermal load and the mass flow, the helium temperatures and pressure at the input. The complexity of the interrelated processes predetermines the necessity for a sufficiently exact description of the thermal and hydraulic characteristics of the flow with the change in the real thermophysical properties taken into account in the near-critical region.

The stability of a heated helium flow in a tube bounded by two throttles at the inlet and outlet is analyzed in this paper. The influence of the channel hydraulic drag and the flow parameters is examined with helium compressibility in the near-critical region taken into account. The classical method of the theory of linear automatic control systems [6] is used to investigate the dynamic processes.

The following equations

$$\frac{D\rho}{Dt} = -\rho\frac{\partial u}{\partial x}, \tag{2}$$

$$\rho u\frac{D}{Dt}\left(h + \frac{u^2}{2}\right) = \frac{q_w \Pi}{F} + \frac{\partial p}{\partial t}, \tag{3}$$

$$\rho\frac{Du}{Dt} = -\frac{\partial P}{\partial x} - \xi\frac{\rho u^2 \Pi}{8F} \tag{4}$$

are the mathematical model for the processes under consideration.

The equation of state [7] for a system in thermodynamic equilibrium relates the gas density to the enthalpy and pressure

$$\rho = f(h, P). \tag{5}$$

The hydraulic drag of the throttles in the self-similar flow regime is defined by the equations

$$\Delta P_1 = k_1\rho_0 u_0^2,\ \Delta P_2 = k_2\rho_{out} u_{out}^2. \tag{6}$$

It can be assumed for sufficiently slow vibrations of the density wave type ($\omega << a/L$) that the hydraulic drag coefficients k_1 and k_2 for the throttles in the developed turbulent flow domain equal their values for the stationary flow regimes. Then the pressure drop perturbations at the inlet valve are represented in the form

$$\delta(\Delta P_1) = 2k_1\rho_0 u_0 \delta u_1. \tag{7}$$

It is assumed in deriving (7) that there are no enthalpy and pressure perturbations before the inlet throttle, and the velocity perturbations are $\delta u_0 \approx \delta u_1$.

For the outlet throttle

$$\delta\Delta P_2 = \frac{-k_2(\rho_0 u_0)^2\left(\frac{\partial v}{\partial h}\right)_p \delta h + 2k_2\rho_0 u_0 \delta u}{1 + k_2\rho_0 u_0\left(\frac{\partial v}{\partial P}\right)_h}. \tag{8}$$

The stationary flow regime was analyzed by means of the equations

$$\frac{\partial \rho u}{\partial x} = 0, \tag{9}$$

$$\rho u\frac{\partial}{\partial x}\left(h + \frac{u^2}{2}\right) = \frac{q_w \Pi}{F}, \tag{10}$$

TABLE 1. Comparison between the Results of Computations and Experiments in [1]

Experiment			Computation			Experiment			Computation		
test No.	P_1 bar	flow	flow	ψ	R	test No.	P_1, bar	flow	flow	ψ	R
5—3	2,87	Unstable	Unstable	0,029	4,81	5—25	2,44	Critical	Stable	0,132	11,7
5—5	3,96	»	»	0,024	4,2	6—14	2,98	»	»	0,11	3,62
5—5	3,90	»	»	0,041	5,56	6—15	5,94	»	»	0,012	3,11
5—5	3,94	»	»	0,034	4,98	4—20	2,87	Stable	»	0,072	3,17
5—11	3,78	»	»	0,055	10,23	4—21	2,89	»	»	0,069	4,93
5—12	5,99	»	»	0,03	5,57	5—3	2,87	»	»	0,037	3,57
5—25	2,45	»	»	0,11	15,6	5—5	3,93	»	Unstable	0,031	4,2
5—26	3,20	»	Stable	0,127	16,1	5—11	3,67	»	Stable	0,137	10,49
6—14	2,95	»	Unstable	0,024	4,43	5—12	5,84	»	»	0,017	4,48
6—15	3,99	»	»	0,019	3,59	5—13	14,81	»	»	0,011	2,0
6—15	5,96	»	»	0,01	3,85	5—25	2,39	»	»	0,168	9,82
4—20	2,87	Critical	Stable	0,069	4,13	5—26	3,17	»	»	0,139	15,8
4—21	2,91	»	Unstable	0,054	6,13	6—14	2,98	»	»	0,035	2,78
5—12	6,01	»	»	0,016	5,73	6—15	3,92	»	»	0,020	2,83

Remark. The test numbers are given in conformity with [1].

$$\rho u \frac{\partial u}{\partial x} = -\frac{\partial P}{\partial x} - \xi \frac{\rho u^2 \Pi}{8F}. \tag{11}$$

Stability of the cooling system described by (2)-(5) was analyzed by means of the relation to the "small" helium velocity (mass flow) perturbations at the inlet to the channel.

After linearization of the system (2)-(5) and taking the Laplace transform with respect to the time for zero initial conditions, the system of equations obtained was analyzed for the enthalpy (δh), pressure (δP), and velocity (δu) perturbations analogously to what was done in [4].

Because of the substantially nonlinear dependence of the thermophysical properties of helium in the near-critical domain on the temperature and pressure, the equations were represented in difference form in x, the channel was divided into N = 100 segments within whose limits $\delta f = (\delta f_j + \delta f_{j+1})/2$, $(\delta f)' = (\delta f_{j+1} - \delta f_j)/\Delta x_j$, where f = h, u, P.

After all the manipulations, we obtain a system of 3N linear algebraic equations whose coefficients form a three-diagonal matrix. The solution of the obtained system of equations is sought by the matrix factorization method [8].

The Nyquist frequency criterion is used as stability criterion and permits estimation of the stability of a closed system with lag by means of the location of the amplitude—phase characteristic of an open system relative to the singularity (—1, j0) [6].

The transfer function of the direct coupling is determined from (7) for the inlet throttle $J(s) = 0.5(k_1\rho_0 u_0)^{-1}$. The feedback transfer function $H(s) = \delta P_1/\delta u_1$ is the ratio between the pressure perturbations and the velocity perturbations of the fluid at the inlet to the tube. The transfer function of the open loop for the construction of the amplitude—phase characteristic equals J(s)H(s).

The time needed to compute one version on the electronic computer ES-1033 did not exceed 1.2 min.

Presented for comparison in Table 1 are results of computations and experimental investigations in [1] for the stability in a helium flow at supercritical pressure in a 3.99-mm diameter tube with 185-m-long working (heated) section in the parameter range P_0 = 2.7-15 bar,

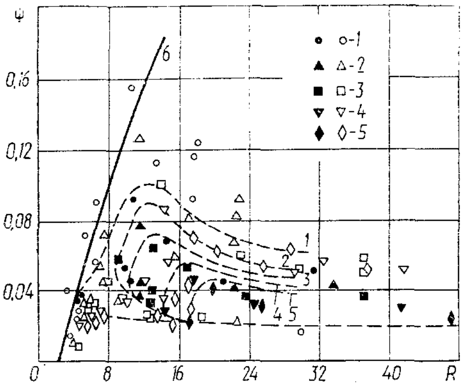

Fig. 1

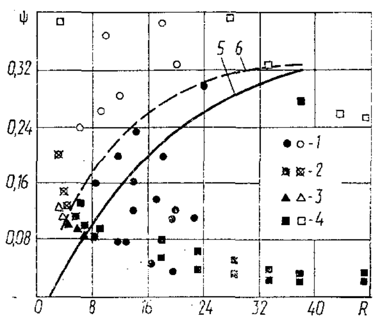

Fig. 2

Fig. 1. Influence of hydraulic drag on supercritical helium flow stability: 1) $\xi = 0.04$; 2) 0.06; 3) 0.08; 4) 0.10; 5) 0.12; 6) stability boundary according to [1]. Flow parameters and boundry conditions from [1]. Numbers of the dashed curves bounding the unstable regime domains correspond to the numbers of regimes 1-5. Open symbols stable and darkened symbols unstable flows.

Fig. 2. Stability boundary for small size tubes: 1-4) $D = 1.4 \cdot 10^{-3}$ m, $L = 0.545$ m, $P/P_{cr} = 1.13$ (1, 3), 0.91 (2), 1.36 (4); $T/T_{cr} = 0.808$ (1, 4), 0.914 (2), 1.04 (3); $G = 0.2\text{-}0.4 \cdot 10^{-3}$ kg/sec; $q_w L = 0.1\text{-}3$ W/m; 5) stability boundary from [1]; 6) stability boundary from the authors' computations. Open symbols, stable, and darked symbols unstable, flows.

$T_0 = 4.6\text{-}6$°K, $G = (0.32\text{-}0.58) \cdot 10^{-3}$ kg/sec, $u_{in} = 0.21\text{-}0.38$ m/sec, $Re_{in} = (0.2\text{-}6.4) \cdot 10^4$. The tube was constrained at the inlet and outlet by two throttles with variable resistances.

Because of the relatively small thermal loads, the Filonenko [9] equation was used to calculate frictional resistance in the tube

$$\xi_T = (1.82 \lg Re - 1.64)^{-2}. \tag{12}$$

Agreement between the test data and computation results is satisfactory. Only two regimes, noted in [1] as stable (5-5) and unstable (5-26), had an opposite result in the computations. The critical modes in the tests were noted as a flow with weakly damped or low-amplitude pressure fluctuations. The regimes were separated individually into stable (4-20), (5-25), (6-14), (6-15) and unstable (4-21), (5-12) in the computations.

The empirical boundary of the beginning of fluctuations in [1], constructed on the basis of experimental data on the parameters ψ and R, turned out to be satisfactory even for the stability estimates obtained in our computations. In contrast to [1], the quantity ψ in the computations was determined here by taking account of local hydraulic losses associated with flow acceleration in the heating section. For $T_1 > T_m$ or $P < P_{cr}$, the value of P_{2k} in (1) was taken equal to the hydraulic drag of the channel between the throttles, while $\Delta P_{1k} = 0$.

It must be noted that the location of the points that characterize the stability of the system being cooled in the ψ–R coordinate system depends on many factors: the hydraulic drag coefficients of the throttles and channel, the flow parameters (the mass flow, pressure, temperature at the inlet), the heat loads, the geometric dimensions, etc. The computations of different versions in which these factors were changed showed that the parameters ψ and R are sufficiently satisfactory only for tubes with a given degree of roughness and length.

The influence of tube wall roughness on helium flow stability is shown in Fig. 1, where the quantity ξ was taken constant along the tube length. For comparison, the friction drag coefficient in a smooth tube was within the limits $(0.6\text{-}0.7) \cdot 10^{-2}$ [1]. In individual cases, even an insignificant increase in the drag coefficients as compared with a smooth tube was enough to stabilize the flow. For instance, for $\xi = 0.04$, the regimes (5-3), (5-5), (5-25), (6-15) became stable. Unstable regime domains for $\xi \geq 0.04$ are denoted by dashed curves in the figure. As the friction coefficients increase, the boundaries of these domains come together and shift toward higher R, i.e., the fluctuations start to occur for a greater heat flux or for smaller helium mass flows. The cancellation of the conditions noted in [10] for the formation of "wandering" thermal spots along the channel length is related, possibly, to the analogous influence of additional drag in the form of spiral threads.

As the tube length diminishes, the domain of unstable regimes expands. The stability boundaries for two tubes with smooth wall L = 0.545 m (L/D = 39) and L = 184 m (L/D = 46) are shown in Fig. 2. Greater throttling of the flow at the inlet is required for an identical change in the helium density from the nlet to the outlet for the channel in order to obtain stable regimes in a short tube. From an analysis of the relationships of the gas densities on the stability boundaries and those computed by using (10) it follows that an increase in channel length for identical quantities $q_w/\rho u$ for two tubes is a destabilizing factor because of a stronger reduction in the helium density at the long tube output.

The origination of density wave type fluctuations is not a singularity of just fluids at supercritical pressure. For instance, the flow in a horizontal tube of diameter $D = 1.8 \cdot 10^{-3}$ mm in unstable for P = 0.2 MPa, $G = 0.4 \cdot 10^{-3}$ kg/sec, $T_0 = 4.75$°K, the drag coefficients $k_1 = 300$ at the inlet and $k_2 = 400$ at the outlet, and the thermal load dQ/dx = 2 W/m (see Fig. 2). The tendency to helium fluctuations at a supercritical pressure is explained by its relatively strong compressibility, which, other conditions being equal, is expressed as a more significant diminution in the density of the medium along the channel length. For $P < P_{cr}$ the fluctuations occur at higher values of $q_w/\rho u$. As the flow temperature rises at the inlet to $T_0 > T_m$ the fluctuations occur for relatively greater $q_w/\rho u$ than for $T_0 < T_m$ (Fig. 2, points 3).

The results presented in this paper refer to the case when the hydraulic drag coefficients are either independent of the regime parameters or are determined by a relationship of the form (12) for fluids with constant properties. It follows from an analysis of the test data [11] that the relationship (12) satisfactorily extends the data on the hydraulic drag for a helium flow of supercritical pressure but for small thermal loads $(q_w D^{0.2}/(\rho u)^{0.8}) < 10$ units SI. Consequently, it can be assumed that the stability boundary cited in [1] that agrees well with our computations when the friction coefficients for a smooth tube are used, has a bounded domain of application for fluids of supercritical pressure, namely, within the limits of a weak dependence of the friction on the thermal load. The necessity to take account of this constraint is associated with the fact that a significant reduction in the heat elimination and friction coefficients is characteristic for fluids of supercritical pressure as the thermal load ($q_w/\rho u$) increases, especially near the fluid pseudocritical temperature. Because of the established dependence of the flow stability on the friction, it can be assumed that the stability boundary will have a more complex form in this case than for quasiisothermal heat transfer.

NOTATION

T, temperature; P, pressure; H, enthalpy; $\rho = 1/v$, density; t, time; x, distance from the inlet; u, velocity; q_w, heat flux density at the wall; G, mass flow rate; L, length of heated tube section; Π, perimeter; F, passage section area; ξ, hydraulic drag coefficient; a, speed of sound in helium; ω, circular fluctuation frequency; δf, perturbation of the function f; s, complex variable of the Laplace transform. Subscripts: 1, at the inlet; 2, at the outlet; 0, before the inlet throttle; cr, at the critical point; j, layer designation.

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HYDRODYNAMIC CHARACTERISTICS AND EFFICIENCY OF CONTACT ELEMENTS OF WET-GAS-CLEANING EQUIPMENT

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Mathematical modeling, with the use of unidimensional equations, is employed as a basis for determining the relations between the hydrodynamic characteristics and the efficiency of wet-gas-cleaning in single-pass contact elements.

Single-pass gas—liquid units for removing impurity particles from gas by the wet method are a widely used type of modern industrial equipment [1, 2]. The inclusion of such equipment in thermally stressed power-generating systems tightens requirements in regard to their reliability and efficiency, while it is at the same time necessary to minimize energy expenditures for pumping the heat carrier being cleaned.

It was shown in several studies [1, 3] that the rate of interphase mass transfer of solid microscopic impurity particles in contact units (CU) can be described by relations of the type

$$\eta = A\Delta P^n, \quad (1)$$

where the coefficients A and n are functions only of the disperse composition of the impurities. This relation is unquestionably approximate in nature, but it adequately illustrates the above-mentioned relationship between the efficiency of a CU and the energy costs of its operation.

The presence of such a functional relation makes it possible to choose the total pressure drop as the determining parameter in the optimization of mass-transfer (connected with interphase transport of impurity particles), dynamic, and geometric characteristics of single-pass contact elements of CUs.

This optimization is done on the basis of a mathematical model of a vapor-drop flow with impurity particles in channels of variable diameter in regard to the process of cleaning of a gaseous dissociated heat carrier in contact with its condensate. The model was constructed on the basis of unidimensional equations of conservation of mass, thermal energy, and the momenta of the liquid, vapor, and solid inert phases with the use of distribution functions for the size of the drops and impurity particles [4]. The model is closed by integral relations for the interphase heat-transfer coefficient and friction coefficient.

The total pressure drop in a unidimensional approximation was determined in the form [4]

$$\frac{dP_\Sigma}{dz} = \frac{dP_{\text{fr. wa}}}{dz} + \frac{dP_{\text{fr.d}}}{dz} + \frac{dP_{\text{a}}}{dz} + \frac{dP_g}{dz}, \quad (2)$$

where

$$\frac{dP_{\text{fr.wa}}}{dz} = \xi_{\text{wa}} \frac{1}{d} \frac{\rho'' W''^2}{2}, \quad (3)$$

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