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HYDRODYNAMICS OF PIPELINE FILLING WITH CRYOGENIC FLUID

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A developed physicomathematical model of pipeline cooling by a liquid cryoagent is presented; some results of mathematical simulation of the process are given; the results are compared with experimental data to determine the adequacy of the description of real physical processes by the model.

At present superconducting devices with circulation cooling of windings by liquid helium and extended communications are being developed in different branches of science and technology. The conditions of operation of superconducting devices presuppose regimes of cooling of magnets by circulating helium and its evacuation in emergencies associated with substantial sudden heat releases to the flow. Moreover, to reduce heat inflows to the magnets, shields are used that are cooled by liquid nitrogen along the channels on their surfaces. In all cases the cryoagent moves along extended channels with a relatively high wall temperature and external heat inflows. Practice has shown that in cases of cooling by liquid nitrogen pressure peaks originate in the channel accompanied by a reverse ejection of the cryoagent, whereas in helium cooling the cooling time increased sharply and in some cases cooling completely stopped.

Similar processes arise in transport of liquid cryoagents along extended pipelines in systems of spacecraft refueling by fuel components (hydrogen, oxygen) [1] and also in their prompt evacuation in emergency.

The problem of cooling of extended pipelines by two-phase cryoagents is a special case of a general conjugate thermohydrodynamic "flow–wall" problem with a liquid–vapor phase transition and different flow conditions for each phase in the presence of a nonstationary distributed thermal load and variable boundary conditions.

Similar processes associated with transport of liquid along pipelines possessing a comparatively high temperature and external thermal load have been considered in reactor building for a relatively long time. This was mainly associated with emergency regimes in heat releasing assemblies with sharp boiling-up of a heat carrier [2, 3]. With its own specifics (the type of heat carrier, temperature level, relatively small pressure drop, etc.), the boiling process in steam-generating channels is in the large similar to cooling of cryogenic pipelines, and mathematical models developed for its description can serve as a basis for creating a model of filling of cryogenic pipelines.

The studied process is the following. A two-phase cryoagent (subcooled liquid) with a constant pressure difference between inlet and outlet is supplied into an extended nonadiabatic pipeline at the ambient temperature. On flowing along the channel the cryoagent is heated due to heat exchange with the wall and then it sharply boils-up. As a result a large amount of vapor is formed, which is not evacuated from the channel, thus leading to a pressure increase (in dynamics) in it. If the pressure exceeds the inlet pressure, then a reverse outflow of the cryoagent takes place. In this case the pressure in the channel decreases and the process of cooling is resumed. As a result the liquid gradually moves to the outlet with cyclic oscillations of flow rate. Under certain conditions the process reaches a self-oscillating mode with stoppage of cooling.

There exist two approaches to the solution of the formulated problem. The first is based on an attempt to account for all factors and physical effects in their interrelation with all possible modes of cryoagent flow, its heat exchange with the wall, and the external heat influx [4-6]. In this case the thermodynamic nonequilibrium of the

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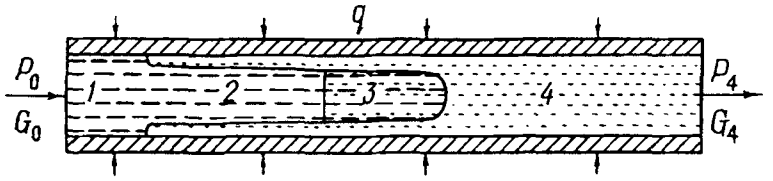


Fig. 1. Cryoagent flow structure: 1) one-phase liquid, 2) reversed plug mode with film boiling, 3) boiling of equilibrium liquid, 4) one-phase gas.

phases and their separate flows are taken into account. This two-component (heterogeneous) model is considered in a one-dimensional nonstationary nonlinear formulation with distributed parameters. The second is based on a substantial simplification of the model with a number of general and particular constraints. The general constraints include the assumption of a one-dimensional problem and thermodynamic equilibrium of phases. The particular constraints introduced by different authors lead either to purely one-component (homogeneous) models [3, 7, 8] or to models which in a one-dimensional formulation attempt to allow for the flow structure of a two-phase flow and the motion of separate phases of it [9-11].

Finite-difference models developed on the basis of the first approach are very complex in both development and application. Their employment leads to excessive computer time expenditures for calculating nonstationary processes in extended objects. There is no confirmation of the possibility of their use for modeling extended pipelines.

Homogeneous models typical for the second approach are rather simple and are successfully implemented for calculating a transition process in one-phase flows. However, they do not allow for the structure of a two-phase flow, the specific character of heat transfer and hydrodynamics in liquid film boiling, or for the dynamics of a fast process, thus making their suitability for calculating the process of pipeline cooling by liquid questionable.

In this case a compromise version is needed which combines the completeness of the models of the first type and the simplicity of the models of the second type.

The developed model is based on an analysis of four typical portions of cryoagent motion with different flow modes for both forward and reverse flows (Fig. 1).

Here the following assumptions were made: the flow is one-dimensional, the gas in the one-phase portion moves only toward the outlet, whereas in other portions it can move both forward and backward; for the plug mode portion the heat flux from the wall is spent to heat the liquid core and to evaporate from its surface, and all vapor formed per time unit discharges through a flat phase interface; axial heat overflows in the channel are absent; to describe the coefficients of heat transfer and drag the empirical relations for stationary conditions and those averaged over the lengths of the corresponding portions are used.

The liquid temperature in the first portion and the liquid-core temperature in the second portion (surface boiling with subcooling) are smaller than the saturation temperature, in the third portion the flow temperature is equal to the saturation temperature, and the phase interface corresponds to the maximum vapor content ($X = X_{max} < 1$).

Equations of conservation of continuity, energy, and motion are written for each of the four portions.

First portion. We assume the liquid to be incompressible and the wall temperature to be equal to the liquid temperature; then the equations take the form

$$G = G_{liq0}(\tau); \tag{1}$$

$$\frac{\partial T_{liq}}{\partial \tau} + U_{liq}\frac{\partial T_{liq}}{\partial x} = \frac{q_{liq}}{F\rho_{liq}c_{pliq}}; \tag{2}$$

$$\rho\frac{\partial U_{liq}}{\partial \tau} = -\frac{\lambda_{liq}}{2D}\operatorname{sign}(U_{liq})U_{liq}^2 - \frac{\partial P}{\partial x}. \tag{3}$$

Second portion. Here surface film boiling in the presence of a subcooled core is observed. To describe the transition process we use a two-zone model with allowance for heat and mass transfer between the liquid and vapor phases. The vapor temperature is equal to the saturation temperature, the liquid is incompressible, and the vapor film thickness is constant and equal to the integral-mean value; pressure drop is caused by motion of the flow liquid core alone, since the film thickness is much smaller than the diameter.

The equations of continuity for the liquid and vapor are

$$\frac{\partial G_{\text{liq}}}{\partial x} = -\frac{q_{\text{ev}}}{r}, \tag{4}$$

$$\frac{\partial G_{\text{v}}}{\partial x} = -\frac{q_{\text{ev}}}{r}, \tag{5}$$

where q_{ev} is the heat flux for liquid evaporation from the phase interface; G_{liq} and G_{v} are the mass flow rates of the liquid and vapor;

$$G_{\text{liq}} + G_{\text{v}} = G_{\text{liq0}}. \tag{6}$$

The energy equation for the liquid is

$$\rho_{\text{liq}} F_{\text{liq}} \frac{\partial I_{\text{liq}}}{\partial \tau} + G_{\text{liq}} \frac{\partial I_{\text{liq}}}{\partial x} = q_{\text{liq}} + q_{\text{ev}}. \tag{7}$$

Allowing for the fact that the vapor is not heated ($T_{\text{v}} = T_{\text{s}}$), the equation is written in the form

$$q_{\text{w}} = q_{\text{liq}} + q_{\text{ev}} \tag{8}$$

or

$$q_{\text{ev}} = q_{\text{w}} - q_{\text{liq}}, \tag{9}$$

where q_{liq} is the heat flux to the liquid core; q_{w} is the heat flux from the wall to the flow.

The equation of momentum conservation is

$$\rho_{\text{liq}} \frac{\partial U_{\text{liq}}}{\partial \tau} = -\frac{\lambda_{\text{liq}}}{2D} \text{sign}\ (U_{\text{liq}})\, U_{\text{liq}}^2 - \frac{\partial P}{\partial x}.$$

In accordance with the adopted assumption the pressure drop in this portion is caused by the liquid core flow alone.

Equations (4)-(9) are supplemented by the equation of state for the vapor

$$\rho_{\text{v}} = \frac{P\mu}{RT_{\text{s}}},$$

The condition of conjugation between the wall and the channel is

$$F_{\text{w}} \rho_{\text{w}} c_{\text{w}} \frac{\partial T_{\text{w}}}{\partial \tau} = \alpha \pi_{\text{w}} (T_{\text{w}} - T_{\text{s}}) + q,$$

where q is the external inflow.

The heat flux from the wall is presented as the sum of the forced and natural convection flows in the vapor film.

Third portion. Here a disperse regime of liquid film boiling takes place. A homogeneous flow model for a heated channel is used.

The equations of continuity, energy, motion, and state for a two-phase flow are:

$$\frac{\partial \rho_{\text{liq v}}}{\partial \tau} + \frac{\partial (U\rho_{\text{liq v}})}{\partial x} = 0; \quad \frac{\partial I_{\text{liq v}}}{\partial \tau} + U_{\text{liq v}} \frac{\partial I_{\text{liq v}}}{\partial x} = \frac{q_w}{F\rho_{\text{liq v}}};$$

$$\rho_{\text{liq v}} \frac{\partial U_{\text{liq v}}}{\partial \tau} = -\frac{\lambda_{\text{liq v}}}{2D} \operatorname{sign}(U_{\text{liq v}}) U^2_{\text{liq v}} - \frac{\partial P}{\partial x}; \quad \frac{1}{\rho_{\text{liq v}}} = \frac{1 - X}{\rho_{\text{liq}}} + \frac{X}{\rho_v(P)}.$$

Fourth portion. Gas with inlet flow rate G_v is heated from temperature T_0 to T_4. A zero-dimensional model is used for the continuity equation and a one-dimensionsional model is used for the energy equation. Under these conditions we have

$$F\frac{\partial \rho_v}{\partial \tau} + \frac{\partial G_v}{\partial x} = 0.$$

The energy equation is

$$F\bar{\rho}_v \frac{\partial T_v}{\partial \tau} + G_v \frac{\partial T_v}{\partial x} = \frac{q_w}{c_{pv}},$$

where $\bar{\rho}_v$ and T_v are the mean density and temperature of the gas.

The equation of motion is written in a quasistationary representation and in integral form

$$P_3 - P_4 = \frac{\lambda_v G_v^2}{2DF^2\bar{\rho}_v}(L - x_3).$$

The equation of state is

$$\rho_v = \frac{P\mu}{RT_w}.$$

In cryoagent forward flow the general boundary conditions are as follows:

$$x = 0: \; P = P_0; \; x = L: \; P = P_4; \; x = 0: \; I = I_0.$$

In cryoagent counterflow they can be written as:

$$x = 0: \; P = P_0; \; x = L: \; P = P_4; \; x = x_3: \; I = I_3.$$

In the course of the solution we must find the displacement of the boundaries of the portions during cooling with allowance for reverse flow of the liquid and also changes in the flow parameters (I, P, U, ρ).

The difficulty in realization of the mathematical model is in description of the dynamics of origination, changing the size, and joining of the mentioned portions with different flow models under conditions of variable flow rate and a reverse liquid flow. In this case in the flow itself the boundaries between the portions can move with different velocities and in different directions. Moreover, subsequent reduction of portions till complete disappearance of them due to discharge of the cryoagent through the inlet orifice during reverse flow is typical for this process.

The model was realized on the basis of a numerical analytical successive-parallel solution of the conservation equations in a one-dimensional approximation with the composition of finite-difference approximations for the numerical solution. The grids were joined by the developed method of a "running" search for bounding nodes of the wall for forward and reverse flows in the description of heat transfer. This made it possible to solve

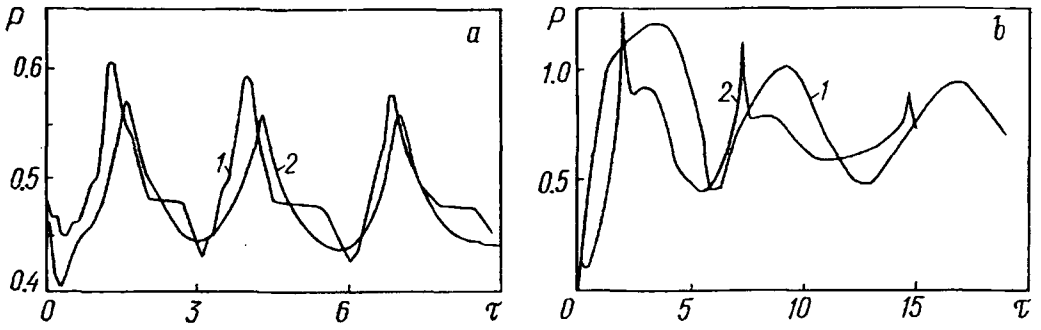


Fig. 2. Experimental (1) and calculated (2) pressures in the channel during transition process: a) ∅ 18 × 1 mm, L = 47 m, P_0 = 0.48 MPa, T_{in} = 300 K, T_0 = 78 K; b) ∅ 23 × 1.5 mm, L = 50 m, P_0 = 0.8 MPa, T_{in} = 300 K, T_0 = 78 K. P, MPa; τ, sec.

the problem of numerical simulation of the process in the description of the dynamics of the changes in the boundaries of the portions and their joining.

It is quite clear that the developed mathematical model is complex and the notions about the physics of the process it involves and the adopted assumptions require thorough practical verification of the model's adequacy to real physical processes.

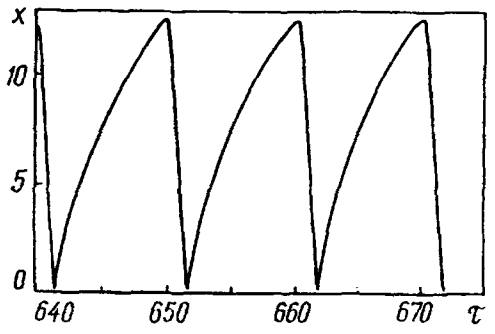
In our opinion, confirmation of the model's authenticity requires satisfactory agreement between calculated and experimental values of the local parameters (pressure, temperature, cryoagent velocity) and integral characteristics of a nonstationary process.

To check the model, an experimental pipeline was cooled by liquid nitrogen. The experimentally obtained data show that pressure variation in the channel has a noticeably expressed dynamic character. A comparison was made between experimental and calculated data on the amplitude of pressure fluctuations for an experimental element of L = 47 m, ∅ 18 × 1 mm at nitrogen-supply pressure P_0 = 0.48 MPa and initial pipeline temperature T_{in} = 300 K (Fig. 2a). Figure 2b presents the data of calculation (by the suggested technique) of the results of similar experimental works for an element with L = 50 m, ∅ 23 × 1.5 mm at T_0 = 78 K, P_0 = 0.8 MPa, T_{in} = 300 K by a group of authors [9]. The maximum discrepancy between the calculation and the experiment for these nitrogen pipelines was 12%.

The integral characteristics of the model were analyzed on the basis of a comparison of calculated and experimental data about the duration of the transition process. The studies were conducted on experimental pipelines at the following parameters: L = 47 m, ∅ 18 × 1 mm (P_0 = 0.3 MPa, T_0 = 88 K; P_0 = 0.4 MPa, T_0 = 87 K; P_0 = 0.48 MPa, T_0 = 86 K) and L = 45 m, ∅ 12 × 1.5 mm (P_0 = 0.3 MPa, T_0 = 79 K; P_0 = 0.4 MPa, T_0 = 83 K; P_0 = 0.5 MPa, T_0 = 83 K) with nitrogen cooling. To check the efficiency of the model for different cryoagents and to solve the most important problem – modeling of cryogenic-pipeline cooling by liquid helium – similar studies were conducted on an experimental pipeline [12]: L = 30 m, ∅ 6 × 0.32 mm (P_0 = 0.12 MPa, T_0 = 4.424 K; P_0 = 0.15 MPa, T_0 = 4.680 K; P_0 = 0.18 MPa, T_0 = 4.911 K).

It turned out that for different pressures of supply, geometry and cryoagents the dynamic model describes the total time of pipeline cooling with an accuracy of 20%, whereas, e.g., the "heat balance" method gives an accuracy of 50%. These data were obtained for adiabatic conditions in the absence of an external heat load.

One of the effects noticed in mathematical simulation and then confirmed experimentally is the fact that maxima of pressure peaks in the channel during cooling fall at a reverse liquid flow rather that at the moment of liquid stoppage. This is facilitated by the inertia of liquid motion and heat exchange with the wall within a small range of flow velocities. In forward cryoagent flow, as a result of heat exchange with the wall, the pressure in the channel slowly increases and the filling rate decreases. As the pressure in the channel exceeds the supply pressure, the liquid continues to move forward for some time due to inertia. This comparatively small period of time corresponds to a sharp decrease in liquid velocity till its stoppage, thus requiring an accurate description of heat transfer in the zone of liquid stoppage. This inertial delay in reverse flow together with convection and evaporation lead to insufficient excess pressure in the channel relative to the pressure at the inlet. This is enough to impart



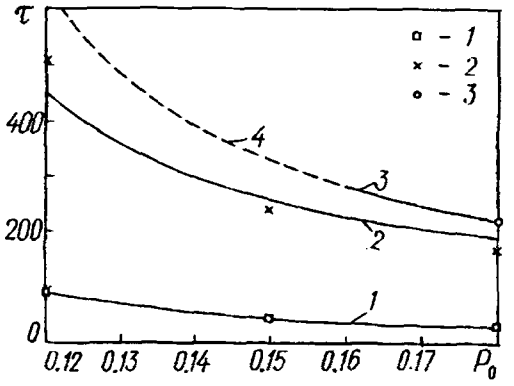


Fig. 3. Self-oscillations of liquid–vapor boundary in filling of channel by liquid helium (calculation). External heat load q = 2 W/m (∅ 6 × 0.32 mm, L = 30 m, P_0 = 0.18 MPa, T_{in} = 320, $T_0 = T_s$ K).

Fig. 4. Duration of transition process for different heat loads and pressures at the inlet to the pilot element (∅ 6 × 0.32 mm, L = 30 m, $T_0 = T_s$; curves-calculation, dots – experiment): 1) q = 0, 2) q = 0.6 W/m, 3) 2.0, 4) 2.0, the region of the transition to the self-oscillatory mode.

initial acceleration in reverse liquid flow. As a result of this acceleration the heat flux from the wall to the liquid increases, evaporation grows, and the pressure in the channel continues to increase, which with the discharge of a portion of liquid through the outlet orifice additionally leads to flow acceleration. With pressure growth, the effect has a cumulative character. The validity of the experimentally obtained data was checked by comparison of time phases of pressure with the direction of liquid flow in the channel (using a "windvane" indicator). AlLowance for this effect made it possible to correctly estimate the amplitude of pressure peaks arising in the pipeline.

Another effect revealed by the method of mathematical simulation is the appearance of pressure peaks at the final stage of cooling. The principal specific feature of the filling process as compared to the quasistationary process of cooling is the decrease in the flow rate at the inlet, since the drag in the liquid portion is prevailing. However, at the end of the process the resistance in the gas portion becomes substantial, and as it decreases, the flow rate increases. This leads to considerable acceleration of two-phase flow, a sharp increase in evaporation, and pressure fluctuations with a relatively high frequency whose amplitude is much smaller than at the initial stage.

The most interesting effect obtained in modeling of cooling by liquid helium is the possibility of cessation of cooling at some value of external heat load when the process reaches a self-oscillatory or new steady-state regime. The essence of the phenomenon is that at a certain external heat load after the cessation of reverse liquid flow the walls of long channels are heated to the extent that the forward flow, being heated, evaporates, the pressure increases, and reverse of the flow a cross-section approximately equal to the previous one occurs again. As a result the flow reaches a self-oscillatory mode and cooling stops (Fig. 3).

The effect of cooling cessation was confirmed experimentally on a pilot helium element [12]. In Fig. 4 a comparison is made between the calculated and experimental data of the duration of the transition process for different values of pressure at the pipeline inlet and of external heat loads.

Thus, the developed model is experimentally confirmed for experimental pipelines of different geometries, different cooling modes, and types of cryoagent.

As a result we can make the following conclusions:

1. The developed physicomathematical model adequately describes the dynamics ot cooling and filling of a nonadiabatic channel by a liquid cryoagent, including the amplitudes and frequencies of pressure and temperature fluctuations, reciprocating motion of the cryoagent in the channel, the duration of the transition process, and also selfoscillatory modes. The agreement between the results of calculation and experiment is satisfactory.

2. The level of model implementation allows one to numerically simulate and calculate the process of cooling by cryoagents of the heat-transfer-continuous flow portion of superconducting devices.

NOTATION

c, heat capacity, J/(kg·K); D, diameter, m; F, surface area, m^2; G, flow rate, kg/sec; I, enthalpy, J/kg; L, length, m; P, pressure, Pa; q, specific heat flux, W/m; r, specific heat of vapor generation, J/kg; R, universal gas constant; T, temperature, K; U, velocity, m/sec; x, coordinate, m; X, mass vapor quality; α, coefficient of heat transfer, W/(m·K); λ, drag coefficient; π, perimeter, m; ρ, density, kg/m^3; τ, time, sec; μ, molecular mass. Subscripts: 0, inlet, 3, right-boundary of the third portion; 4, outlet; liq, liquids; liq v, two-phase; p, isobaric; ev, evaporating; s, on the saturation line; v, vapor, w, wall.

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