Modeling of the cooling process of long pipelines for transportation of liquid hydrogen

O V Kalyadin1,3, A V Sergeev1, A A Grebennikov2 and A Yu Lopatin1

1 Department of Solid State Physics, Voronezh State Technical University, Moscow Avenue, 14, Voronezh, 394026, Russia
2 Basic research and educational center «Physics and technology of thermoelectric phenomena», Voronezh State Technical University, Moscow Avenue, 14, Voronezh, 394026, Russia
3Corresponding author: kaljadin@gmail.com

Abstract. A simplified model of cooling long insulated pipelines with liquid hydrogen flowing through them has been developed. The model allows us to determine the flow parameters at different points in time and to estimate the pipeline cooling time. When developing the model, the following assumptions were made: the linear velocities of the liquid and vapor are equal, the phases are in a state of thermodynamic equilibrium, the dependencies obtained for a single-phase flow are applicable to determine the friction coefficient of a two-phase flow. Based on the model, an automated algorithm for calculating the cooling process of a long pipeline with cryogenic components was developed. It allows us to obtain data for plotting the temperature fields of the pipeline walls and the flow of the transported cryogenic liquid at different times, as well as to determine the cooling time of the pipeline and the moment of the onset of a stationary single-phase flow. The calculation results are in good agreement with the experimental data.

1. Introduction
Testing of liquid rocket engines (LRE) is a comprehensive test for equipment, components and systems of an engine to establish compliance of their characteristics with the specified tactical and technical requirements [1]. Tests of the LRE are a complex and multifaceted system of measures that allows us to obtain reliable information about the operation of all LRE components and to make an informed decision about its installation in a launch vehicle. Unlike other components or systems of the launch vehicle, LRE tests are characterized by some features related both to the specifics of the processes occurring in them and to operating conditions.

The most important feature that significantly limits the number of possible tests is their high cost, which is associated largely with the high cost of fuel components and the uniqueness of the equipment used [2]. In this regard, high informativity of tests, rational planning and possible minimization of both the total number of tests and fuel losses in each of them is necessary. To solve this problem, it is necessary to work out the optimal modes of difficult-to-control technological processes, which include the process of preparing a bench system for testing. However, to carry out such processes under different initial conditions in order to collect statistical data and to obtain the optimal mode is a very expensive and difficult task. Therefore, the modeling of physical processes actually occurring in the system is the only right decision [1,2].

2. Features of the cooling process of pipelines during transportation of cryogenic liquids through them
It is necessary to cool the main pipeline to achieve the required flow rate of cryogenic products. Cooling can be carried out both by liquid and gas. The choice of the phase state of the cryogenic product is determined by the purpose of the system, the requirements for the temporary and dynamic characteristics of the transition regime, the permissible levels of temperature stresses in the structural
elements in liquid systems, cooling is usually performed directly by liquid products. The cooling stage of the walls of the main pipeline is characterized by intensive steam generation. This is due to the fact that the initial wall temperature is close to the ambient temperature; therefore, its value significantly exceeds both the saturation temperature (T_s) and the overheat limit temperature (T_o) - the upper limit of the existence of cryogenic products in liquid form. Thus, cryogenic liquids cannot exist near the pipeline walls, while walls temperature is high, because they instantly turn into steam.

When cryogenic liquids enter a warm pipeline, the pipeline walls are stably blocked by the vapor film, i.e. film boiling is realized. With a decrease in the wall temperature below the T_o temperature, a film boiling crisis occurs, and then it is replaced first by transitional and then bubble boiling. Bubble boiling degenerates into convective heat transfer as the temperature of the pipe wall approaches the saturation temperature. Boiling of cryogenic liquids at the stage of pipelines cooling is realized under various modes of movement of the vapor-liquid mixture. The film boiling is characterized by the presence of a vapor film in the near-wall region; therefore, the modes of motion of vapor-liquid mixtures are called reversed. The most typical of cooling conditions are the rod, shell and dispersed reversed modes. As wall temperature decreases, conditions for wetting the walls with liquid appear, so the reversed regimes are replaced by the usual ones, which are characterized by the presence of a liquid phase in the near-wall region. The cooling and filling of extended main pipelines with cryogenic liquids are accompanied by fluctuations in flow rate and pressure, which are caused by intensive steam generation.

It was found by experimental studies [3] that the thickness of the vapor film arising during the flow of underheated liquid is so small that it practically does not constrain the movement of the liquid jet. Moreover, the presence of a vapor film leads to a decrease in shear stresses, since the viscosity of the vapor is an order of magnitude lower than the viscosity of the liquid. The liquid warms up as it moves through the pipeline, respectively, the pressure in the pipeline increases and the filling rate decreases. If the length of the pipeline is large, then the front of the liquid warms up to the saturation temperature at some distance. In this case, the pressure in the pipeline and in the tank becomes the same, so the flow of liquid into the pipeline stops. At the same time, the core structure of the flow is destroyed due to the heating of the liquid and the increase in vapor content.

When a liquid is heated to a saturation state, almost the entire heat flux from the walls goes to evaporation. The steam generation rate and, consequently, the pressure in the pipeline continue to increase, despite the cessation of the new portions of liquid. The pressure in the pipeline can exceed the inlet pressure from 1.5 to 2 or more times. As the liquid evaporates and the vapor escapes, as well as the liquid, is partially displaced back into the vessel, the steam generation rate in the pipeline decreases, and after some time conditions for a new portion of the liquid to enter are created in it. A new flow of liquid leads to a smaller increase in pressure, since a fairly high pressure has remained in the pipeline and a part of the liquid has remained. During the entire period of cooling and filling of extended pipelines, fluctuations in the flow rate of the incoming liquid and pressure in the pipeline have a much lower amplitude compared to the first cycle.

An analysis of the experimental data shows that the steady flow of liquid occurs almost simultaneously with the cooling of the wall. The liquid appears at the outlet of the pipeline before the pipe is completely cooled, so a vapor-liquid mixture leaves the pipeline for some time. The entire cooling period \( \tau \) can be divided into two stages. The first stage \( \tau_1 \) is from the beginning of the liquid supply until the vapor-liquid mixture appears at the outlet of the pipeline. The second stage \( \tau_2 \) is from the moment the vapor-liquid mixture appears at the outlet until the wall is completely cooled and the stationary flow rate is established. The ratio between the duration of individual stages and the total cooling time (as well as between the length of the evaporation zone and the total length of the pipeline) can be very different. Three types of pipelines are distinguished - long, medium and short, depending on the relative duration of the individual cooling stages. In long pipelines, the time of the liquid appearance at the outlet does not differ much from the time of the complete cooling of the wall. The length of such pipelines far exceeds the evaporation zone \( \tau = \tau_1, l, \ll l \). The duration of the individual stages is commensurate with each other for medium pipelines. The evaporation zone in
such pipelines is slightly less than their length at the beginning of cooling. It increases with decreasing wall temperature and after some time exceeds the total length \( \tau = \tau_1 + \tau_2, l \ll l \). In short pipelines, the time of the appearance of the liquid at the outlet is negligible compared to the total cooling time \( \tau = \tau_1, l > l \). The length of such pipelines is insufficient for the complete evaporation of the liquid.

The relative length of the pipelines has the main influence on the ratio between the duration of individual stages and the total cooling time. As shown by the analysis of experimental data obtained on nitrogen and oxygen, long pipelines include pipelines that are longer than \((1,5\div2) \cdot 10^3\) calibers. short ones include pipelines shorter than \(0,5\cdot10^3\) calibers. In the practice of creating large cryogenic systems, the length of the main pipelines exceeds several thousand calibers; therefore, the cooling of long pipelines is of the greatest practical interest.

3. Model description
At the initial stage of cooling, the temperature of the pipeline walls significantly exceeds the saturation temperature of cryogenic liquids. Therefore, their partial or complete evaporation occurs at the outlet of the pipeline. When transporting cryogenic liquids along warm pipelines their partial or complete evaporation occurs. Therefore, the temperature distribution in the walls and the flow parameters are determined by solving a system of equations describing the change in the flow parameters in the single-phase and two-phase flow sections. The exact solution to this problem is associated with great difficulties, since the laws and forms of joint motion of a liquid and gas are much more complex and diverse than the laws and forms of motion of homogeneous flows. The discrete volumes of each phase contain a large number of molecules, so the movement of the medium inside any volume can be determined by the usual differential equations of continuity, motion, and energy. However, in two-phase flows, in addition to the external boundaries formed by the channel walls, there are internal interfaces, which are variable in time and space. Force and thermal interactions appear at the interface; therefore, the equations of hydrodynamics for each of the phases must be supplemented with equations describing the mechanical and thermal interaction of phases at the interfaces. From a practical point of view, in addition to determining liquid losses, an interesting task is to estimate the time for complete cooling of cryogenic pipelines and establish a steady flow rate of cryogenic liquid.

To calculate the cooling time of the pipeline, it is necessary to solve a system of differential equations of continuity, motion, flow energy, and thermal conductivity for walls with different boundary conditions on the inner and the outer surfaces of the pipe. In the case when the cooling of the pipeline proceeds without changing the aggregate state of the substance, the following expressions are used as the initial equations:
- flow continuity equation
  \[
  \frac{\partial \rho}{\partial \tau} + \frac{\partial (\rho W)}{\partial z} = 0; \quad (1)
  \]
- equation of motion
  \[
  \frac{\partial R}{\partial z} - \frac{4\sigma}{d_{int}} = \rho \frac{\partial W}{\partial \tau} + \rho W \frac{\partial W}{\partial z}; \quad (2)
  \]
- energy equation
  \[
  \rho \frac{\partial i}{\partial \tau} + \rho W \frac{\partial i}{\partial z} = \frac{4q}{d_{int}}; \quad (3)
  \]
- heat conduction equations for pipe walls
\[
\frac{\partial T_w}{\partial \tau} = a \left( \frac{\partial^2 T_w}{\partial R^2} + \frac{1}{R} \frac{\partial T_w}{\partial R} \frac{\partial^2 T_w}{\partial z^2} \right); \\
\lambda_w \frac{\partial T_w (R_{\text{int}} \tau)}{\partial R} = q_{\text{int}} = a [T_w (R_{\text{int}} \tau) - T]; \\
-\lambda_w \frac{\partial T_w (R_{\text{int}} \tau)}{\partial R} = q_{\text{ext}}.
\]

\( q_{\text{int}}, q_{\text{ext}} \), heat flux density on the inner and the outer surfaces of the pipe; \( R_{\text{int}}, R_{\text{ext}} \), inner and outer pipe radii; \( T_w, T \), the temperature of the wall and the one of the flow.

When describing the process of cooling thin-walled insulated pipelines by a flow of a cryogenic liquid with partial or complete evaporation, the form of writing the heat conduction equation for the pipe walls (4) - (6) is preserved, but the equations of continuity (7), motion (8) and flow energy (9) change due to the formation of a vapor-liquid mixture.

\[
\frac{\partial}{\partial \tau} \left( (1 - \phi) \rho_i + \phi \rho_s \right) + \frac{\partial}{\partial z} \left( (1 - \phi) \rho_i W_i + \phi \rho_s W_s \right) = 0; \\
g \left( (1 - \phi) \rho_i + \phi \rho_s \right) - \frac{\partial p}{\partial z} \frac{4 \sigma}{D} = (1 - \phi) \rho_i \frac{\partial W_i}{\partial \tau} + \phi \rho_s \frac{\partial W_s}{\partial \tau} + \\
+(1 - \phi) \rho_i W_i \frac{\partial W_s}{\partial \tau} + \phi \rho_s W_s \frac{\partial W_s}{\partial \tau} + (W_s - W_i) \left( \frac{\partial}{\partial \tau} \left( \phi \rho_s \right) + \frac{\partial}{\partial z} \left( \phi \rho_s \right) W_s \right) \\
\frac{\partial}{\partial \tau} \left( (1 - \phi) \rho_i i_i + \phi \rho_s i_s \right) + \frac{\partial}{\partial z} \left( (1 - \phi) \rho_i i_i W_i + \phi \rho_s i_s W_s \right) = \frac{4 q}{d_{\text{int}}}
\]

\( \phi \), volumetric steam content of the flow (in the evaporation section \( 0 < \phi < 1 \), in the gas flow section \( \phi = 1 \)); \( q \), heat flux density on the inner surface of the pipe; \( \sigma \), shear stress on the wall.

However, the system of equations remains open, therefore additional experimental dependencies are required to solve it. These dependencies should describe the relationship between the volumetric vapor content, the velocities of each of the phases, shear stresses on the wall, the temperature of the vapor phase and walls, the intensity of heat transfer from the walls to the product. Such dependencies were obtained only for some special cases. They are not universal due to the complex nature of two-phase flows. The absence of additional general equations makes solving the system of initial equations impossible. In this regard, methods for determining the characteristics of the process of cooling pipelines by cryogenic liquids are based on obtaining semi-empirical and empirical relationships from experimental data.

In this investigation, a homogeneous model was used to solve the considered problem approximately [4,5]. Its main assumptions are based on the assumption that the linear velocities of vapor and liquid are equal, the phases are in thermodynamic equilibrium, and the dependencies obtained to determine the coefficient of friction of single-phase flow are applicable to two-phase flow. In the homogeneous model, two-phase flow is considered as a single-phase flow, in which the specific volume in each section is related to the mass vapor content and the specific volume of each of the phases, i.e.

\[
\bar{V} = \bar{V}_i (1 - \phi) + \bar{V}_s \phi
\]
In this case, the system of equations (1-6) can be used as a calculated one. We will assume that the mass flow rate is known, the thermal conductivity of the wall in the direction of motion is zero and in the direction of the normal, the wall on the outside is ideally isolated from the heat influx from the environment. These assumptions are justified from a technical point of view, they can significantly simplify the system of initial equations and its solution. In view of the above and taking into account that \( \frac{di}{dT} = c_p dT \) we transform the system (1-6) to the form

\[
\frac{\partial T}{\partial \tau} + W \frac{\partial T}{\partial z} = \frac{4\alpha(T_w - T)}{\rho_k c_{pg} d_{int}}
\]

(11)

\[
\frac{\partial T_w}{\partial \tau} + \frac{\alpha(T_w - T)}{\rho_w c_{pw} \delta} = 0
\]

(12)

The temperature distribution in the gas and the wall at the initial moment of time and the gas temperature at the inlet are set as the boundary conditions \( T(0,z) = f(z) \); \( T_w(0,z) = \varphi(z) \); \( T(\tau,0) = \psi(\tau) \). The heat transfer coefficient included in the system is calculated by the formula

\[
\alpha = \frac{\lambda_{st}}{d_{int}} Nu
\]

(13)

where the Nusselt number, taking into account the homogeneous flow model [5]

\[
Nu = 0.0065 Re^{0.8} \left( \frac{c_{pg}(T_0 - T_{wall})}{r} \right)^{-1/6}
\]

(14)

Functional dependencies are obtained as a result of solving the system of equations. They make it possible to obtain the distribution of wall and flow temperatures for any moment in time in each section of the pipeline. Thus, it is possible to estimate the cooling time of a random section of the pipeline or the pipeline as a whole. In addition, the obtained dependences, plotted for different points in time, can be used to calculate the current value of the average wall temperature or plot the curves of the change in the average temperature \( T_{ws}(\tau) \) and the thermal resource of the walls \( Q_w(\tau) \) in any section under various initial conditions

\[
T_{ws} = \frac{1}{L_{wall}} \int_0^{L_{wall}} T_w(z)dz
\]

(15)

\[
Q_w(\tau) = (T_{ws}(\tau) - T) c_{pw} (T_{ws}) \rho_w \delta
\]

(16)

4. Modeling results

An automated algorithm for calculating the cooling process of a long pipeline with cryogenic components is developed on the basis of the proposed model. A pipeline made of steel was taken as the object of modeling. The pipeline is insulated with screen-vacuum insulation. Total pipeline length is 272.5 m, bore is 96 mm, wall thickness is 2 mm. The total mass of the locking equipment located on the pipeline is 246 kg. Liquid supercooled hydrogen is fed into the pipeline under an excess pressure of 0.2 MPa, the hydrogen temperature at the inlet is 19 K. The ambient temperature is 293 K.

As a result of calculations, the distribution of hydrogen temperature and wall temperature along the length of the pipeline, as well as at different points in time, were obtained. These results are shown in figures 1, 2 for some sections and points in time.
Figure 1. Distribution of the temperature of the flow (a) and the temperature of the wall (b) along the length of the pipeline at different points in time.

Figure 2. Time dependence of the temperature of the flow (a) and the temperature of the wall (b) in different sections of the pipeline.

The current value of the average wall temperature according to the formula (15) can be calculated using the temperature fields plotted for different points in time along the length of the cooling pipes (figure 2) $T_w = T_w(x)$. Figure 3 shows the time dependence of the average wall temperature along the pipe length $T_{ws}(\tau)$. Based on the obtained data, the time dependences of changes in the thermal resource of the wall for the entire pipeline and its individual sections at a distance of 5, 50, 150 m from the inlet section were calculated using formula 16 (figure 4):
Figure 3. Time dependence of the average wall temperature.

Figure 4. Time dependence of the thermal resource of the wall for different sections of the pipeline.

It can be seen from the figures that the calculated cooling time of the pipeline is ~ 400 s, which is in good agreement with the data obtained as a result of experiments on the operating stands of the KBKhA.

5. Conclusion
A mathematical model of cooling of long pipelines with cryogenic components has been developed. Based on this model, an automated calculation algorithm was created. It allows us to obtain data for plotting the temperature fields of the pipeline walls and the flow of the cryogenic liquid at different points in time, as well as to determine the time of pipeline cooling and the moment of the onset of a stationary single-phase flow. Using this model under various initial and boundary conditions, it is possible to determine the optimal mode of pipeline cooling and achieve minimum losses of cryogenic components with minimum time costs.

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