DYNAMICS OF THE ICING-OVER OF LOW-TEMPERATURE PIPELINES IN STAGNANT WATER

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The icing-over of a pipe conveying cooled gas while immersed in water is investigated experimentally. The data obtained are in good agreement with the results of numerical calculations.

Investigations of the interaction between a low-temperature pipeline and the ambient have been receiving ever-increasing attention in recent years [1]. The results of such investigations can be used to solve reliability problems and ensure the stability of low-temperature pipelines, and they can also be useful in solving problems of congealment prediction [2], artificial freezing of grounds and water reservoirs [3], etc.

From a theoretical point of view, this problem involves heat exchange between a pipe cooled and the ambient as the state of aggregation of the latter changes. In the standard congealment model [4], the material under investigation is initially at a constant temperature T_0, equal to, or exceeding, its melting point T_f. Cooling of the surface with which the material in question is in contact then starts at a certain given time, in which the temperature T_w at this surface drops below T_f, which causes congealment of the medium in contact with this surface. The freezing front propagates into the medium, and the heat liberated in congealment is transferred by thermal conductivity through the solidified layer to the cooled surface. If the temperature of the material exceeds the melting point, thermal conductivity ensures heat transfer to the moving freezing front, which is at the melting point. In the course of time, the thickness of the frozen layer $\xi(t)$ increases, although at a diminishing rate, which is due to the increasing thermal resistance of this layer.

It can be considered that the process of solidification of a liquid that is not superheated ($T_0 = T_f$) corresponds to the described classical model. At the same time, experiments [5, 6] on the congealment of superheated liquids ($T_0 > T_f$) show that the effect of free convection must be taken into account in this case (these experiments were performed on n-eicosane, which congeals at 36°C [5], and n-heptadane [6], whose melting point is close to room temperature). In our experiments, the icing-over of a low-temperature pipe took place in stagnant water, whose temperature exceeded 0°C. The experiments were performed by means of equipment based on a TKhM-125 cooling turbine machine.

The experimental device includes (Fig. 1) a simulation tank, the cooling turbine machine, air ducts, heaters for maintaining the air temperature in the pipe in the range from −10 to −120°C, and a control panel for monitoring all the controlled parameters.

The air cooled in the cooling turbine machine is conveyed through air ducts and heaters, where it is heated to the assigned below-zero temperature, to the pipe under investigation. The pipe is thus cooled, and the air is returned to the machine. The pipe is placed in the simulation tank, which is filled with water and is well insulated. Since the experiments were performed with water at 2 and 6°C, the simulation tank was placed outside the building, and the experiments were performed at times when the ambient air temperature was close to the water temperature.

The thickness and profile of the ice crust were first measured by means of mechanical, and then electrical, instruments. Electrical measurements are based on the considerable difference between the electrical resistivities of water and ice. The electrical conductivity data-unit consists of a cylindrical copper rod with a diameter of 1 mm, fastened in a special frame. The connecting wire is brazed to the rod at one end, and the junction is sealed with epoxy resin. The frame containing the data-units is placed in the pipe under investigation.

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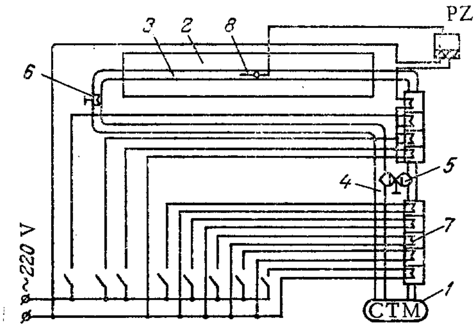

Fig. 1

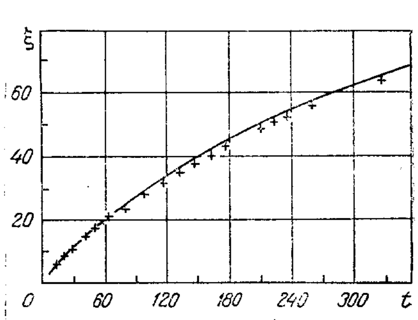

Fig. 2

Fig. 1. Schematic diagram of the experimental arrangement. 1) TKhM-125 cooling turbine machine; 2) 2 × 2 × 2 m simulation tank; 3) pipe under investigation; 4) air ducts; 5) bypass gate valve; 6) cutoff gate valve; 7) electric heaters; 8) temperature data unit.

Fig. 2. Thickness of the ice layer forming around a cooled pipeline as a function of time for a pipe diameter of 219 mm at T_0 = 6°C (the curve pertains to the numerical solution); ξ is given in millimeters, t is given in minutes.

The data-unit operates in the following manner. An electric current, measured by means of an ammeter, passes through water between the rod contact and the frame of the simulation tank. From the time the contact is frozen in ice, when the circuit resistance sharply increases, the milliammeter indicates zero. After recording this time, we pass to the next contact, etc. The rod contacts are spaced at 5-mm intervals, 40 of them in each of the three directions, upward, downward, and sideways in relation to the pipe.

The temperature field in the ice layer that has formed and in the adjacent water layer is measured by means of thermocouples, braided into an 18-piece strand and connected through a switch to a UPIP-60M secondary instrument. The first thermocouple is pressed directly against the pipe, while the other 17 thermocouples are spaced at 10 mm.

The experiments were performed on a pipe with a diameter of 219 mm, a length of 4 m, and a wall thickness of 6 mm. The distance between the free surface of the water and the upper generatrix of the pipe was equal to 60 mm. The temperature of the air entering the pipe T_g was kept constant at −60°C. The duration of an experiment was equal to 6 h for water at 6°C (Fig. 2) and approximately 30 h for water at 2°C (Fig. 3).

Analysis of the experimental data has shown the following: 1) The temperature distribution in the solid phase over the pipe section is close to a logarithmic distribution [7]; 2) the temperature at the ice–water interface may be below zero; 3) the buildup of ice with respect to the pipe diameter is nonsymmetric in character; 4) there is no strongly pronounced cooling zone in the water surrounding the pipe.

The obtained experimental data were compared with the numerical solution of the problem, which was stated as the problem of congealment (with an allowance for free convection) of a homogeneous, isotropic medium around an infinitely long round cylinder with cooled gas flowing through it.

It is assumed that there is a very thin ice layer at the instant of time $t = t_f$ on the outside surface of the cylinder, while the frozen layer surface adjacent to the cylinder has a temperature equal to the temperature of the cylinder's outside surface $T_w(t)$. The moving boundary in contact with the liquid phase is at the temperature of phase transition T_f. Transition of the water surrounding the cylinder from the liquid to the solid state occurs at this boundary, which liberates the latent fusion heat L, which is transferred by thermal conductivity through the ice layer to the cylinder wall. The heat from the surrounding water is transferred to the phase transition front (PTF) as a result of free convection. It is also assumed that the thermophysical properties of either phase are constant, although different, while the phase transition occurs at $T_f = 0$.

For the statement given above, the mathematical formulation of the problem is given by

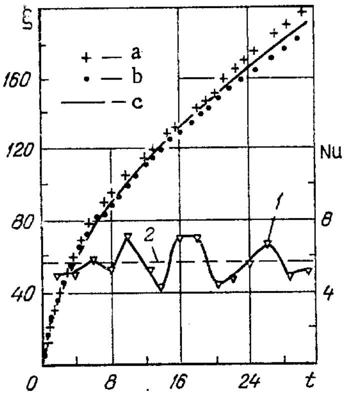

Fig. 3

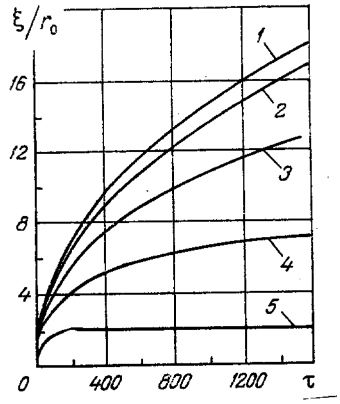

Fig. 4

Fig. 3. Thickness of the developing ice layer (a, above the pipe; b, under the pipe; c, numerical solution) and the Nu values (1, calculated on the basis of the experimental data obtained; 2, averaged with respect to time) as functions of time. The pipe diameter is equal to 219 mm; T = 2°C, ξ is measured in millimeters; t is given in hours.

Fig. 4. Time dependences of the PTF location, obtained as a result of a numerical investigation of the congealment of a superheated ($T_0 > T_f$) liquid (water) at the cylinder surface due to heat transfer by thermal conductivity in the solid phase and by natural convection in the liquid phase. 1) T_0 = 2°C; 2) 6; 3) 10; 4) 15; 5) 20°C.

$$\frac{\partial T_1(r,\ t)}{\partial t} = \frac{a_1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial T_1(r,\ t)}{\partial r}\right), \quad r_{ou} \leqslant r \leqslant r_f, \quad t \geqslant t_f; \tag{1}$$

$$T_2(r,\ t) = T_0 = \text{const}, \quad r \geqslant r_{ou}, \quad T_0 > T_f; \tag{2}$$

$$\lambda_1 \frac{\partial T_1(r,\ t)}{\partial r}\bigg|_{r=r_f} = \alpha_f(T_0 - T_f) + \gamma_2 L\frac{dr_f}{dt}, \quad t \geqslant t_f; \tag{3}$$

$$T_1(r_f,\ t) = T_f = \text{const}; \tag{4}$$

$$\lambda_1 \frac{\partial T_1(r,\ t)}{\partial r}\bigg|_{r=r_{ou}} = k[T_w(t) - T_g], \quad t \geqslant t_f. \tag{5}$$

System (1)-(5) was solved numerically by using the method of finite differences with explicit separation of the interface between phases, which Tien and Churchill used earlier [8] for solving the problem of the icing-over of a cylinder at T_w = const with conductive heat transfer in the liquid phase.

We now pass to dimensionless variables:

$$U = \frac{T_1 - T_f}{T_f - T_g}; \quad V_0 = \frac{\lambda_2}{\lambda_1}\frac{T_0 - T_f}{T_f - T_g}; \quad R = \ln\left(\frac{r}{r_{ou}}\right); \tag{6}$$
$$\mathrm{Bi} = \frac{kr_{ou}}{\lambda_1}; \quad \mathrm{Nu} = \frac{2\alpha_f r_{ou}}{\lambda_2}; \quad \tau = \frac{a_1 t}{r_{ou}^2}.$$

In terms of these variables, the finite-difference statement of problem (1)-(5) is given by [9]

$$\hat{U}_i = U_i + (U_{i-1} + U_{i+1} - 2U_i)\frac{\Delta\tau}{(\Delta R)^2}\exp(-2R); \tag{1a}$$

$$V(R,\ \tau) = V_0;\tag{2a}$$

$$\Delta R_f = \frac{C_1(T_f - T_g)\Delta\tau}{L\exp(2R_f)}\ \frac{\gamma_1}{\gamma_2}\left[\left(\frac{\partial U}{\partial R}\right)_{R=R_f} - \frac{\mathrm{Nu}V_0\exp(R_f)}{2}\right];\tag{3a}$$

$$U(R_f,\ \tau) = 0;\tag{4a}$$

$$\hat{U}_w = \hat{U}_1 = U_2 - \Delta R\,\mathrm{Bi}\,(U_w + 1).\tag{5a}$$

Expression (1a) is used for calculating the new temperature values in the solid phase on all grid lines beginning with the second (measurements start at the cylinder's outside surface, R = 0), with the exception of the line directly adjoining the PTF. In view of the temperature gradient discontinuity at the interface between phases, a stable temperature value cannot be obtained on this line. We used Lagrange's five-point interpolation for this purpose [10].

Equation (3a) is used for calculating the increment of the PTF radius during the time $\Delta\tau$. The value of the partial derivative $(\partial U/\partial R)_{R=R_f}$ was determined from the expression obtained by differentiating the corresponding Lagrange interpolation formula.

Equation (5a) is used for calculating the temperature at the cylinder's outside surface at successive instants of time.

For the numerical solution of Stefan's problems, based on the grid method with explicit separation of the moving interface between phases, in turn it is necessary to know the PTF location at the instant of time used for the start of measurements ($t = t_f$) and the corresponding distribution. For this purpose we composed a special subprogram, which was used at the beginning of measurements for organizing the initial data. It was used first to determine the time during which the water temperature at the assigned (very small) distance ξ_f from the cylinder surface is equal to, or lower than, the freezing point and also the corresponding temperature at the cylinder's surface. Then, in accordance with the solution obtained by Stefan for the two-dimensional problem of congealment of a homogeneous medium for constant boundary conditions [11], we calculated the freezing time for the layer ξ_f and the temperature distribution in it (for a very short time interval, the solution of the axisymmetric problem is close to the solution of the corresponding two-dimensional problem). In performing the calculations, the values of the thermophysical coefficients λ_1, λ_2, a_1, C_1, γ_1, and γ_2 were borrowed from [11], while the value of the latent fusion heat was obtained from [12].

The heat transfer coefficient k in Eq. (5) was determined on the basis of our experimental data in the following manner. The density of the quasistationary heat flux from the pipe's outside surface to the cooled air passing through it is defined as

$$q_w = k(T_w - T_g).\tag{7}$$

The density of the quasistationary thermal flux passing from the phase transition surface through the ice layer to the pipe's outside surface is equal to

$$q_{\mathrm{ther}} = \frac{T_f - T_w}{\dfrac{r_{\mathrm{ou}}}{\lambda_1}\ln\left(\dfrac{r_f}{r_{\mathrm{ou}}}\right)}.\tag{8}$$

However [7],

$$k = \frac{1}{\dfrac{1}{\alpha_g} + \dfrac{r_{\mathrm{ou}}}{\lambda_{\mathrm{wa}}}\ln\left(\dfrac{r_{\mathrm{ou}}}{r_{\mathrm{in}}}\right)}.\tag{9}$$

From the condition of thermal balance, we obtain

$$\alpha_g = \frac{1}{r_{\mathrm{ou}}\left[\dfrac{T_w - T_g}{\lambda_1(T_f - T_w)}\ln\left(\dfrac{r_f}{r_{\mathrm{ou}}}\right) - \dfrac{1}{\lambda_{\mathrm{wa}}}\ln\left(\dfrac{r_{\mathrm{ou}}}{r_{\mathrm{in}}}\right)\right]}.\tag{10}$$

Thus, by substituting in (10) the values of T_w and $r_f = r_{ou} + \xi$ measured in the course of experiments, we obtain the corresponding coefficients of heat transfer from the pipe's inside surface to cooled air, α_g.

Analysis of the results obtained shows that, for $\lambda_{wa} = 46.5$ W/m•°C, $\lambda_1 = 2.25$ W/m•°C, and $T_g = -60$°C, the time-averaged values of $\bar{\alpha}_g$ lie in the 40-44 W/m²•°C range.

We were unable to find in the literature the value of the coefficient of heat transfer from water to the ice surface α_f for the icing-over of pipelines in stagnant water. We attempted to estimate roughly this value on the basis of our experimental data.

Comparison between the latent heat liberated in congealment with the physical heat liberated in supercooling a solid to a temperature below the melting point shows that the latter amounts to 5-10% of the first. Thus, in rough evaluations, the latter quantity can be neglected. Then, the amount of heat liberated in the freezing of a certain volume of water around the pipe can be represented as the difference between the amount of heat conveyed through the ice layer formed to the pipe surface on the one hand, and the heat supplied by free convection to the phase transition surface on the other. The first quantity is defined by Eq. (8), while the second quantity, reduced to the pipe's outside surface, is determined by

$$q_{conv} = \alpha_f (T_0 - T_f) \frac{r_f}{r_{ou}} . \tag{11}$$

On the other hand, if the PTF position is known at certain given instants of time, q_{con} is defined as

$$q_{con} = \frac{(\hat{\bar{r}}_f^2 - \bar{r}_f^2)\gamma_2 L}{2r_{ou}(\hat{t} - t)} . \tag{12}$$

Then,

$$\alpha_f = \left[\frac{\lambda_1 (T_f - T_w)}{r_{ou} \ln\left(\frac{\hat{\bar{r}}_f}{r_{ou}}\right)} - \frac{(\hat{\bar{r}}_f^2 - \bar{r}_f^2)\gamma_2 L}{2r_{ou}(\hat{t} - t)} \right] \frac{r_{ou}}{(T_0 - T_f)\bar{r}_f} . \tag{13}$$

The value of α_f can readily be determined from the above expression if the pipe surface temperature and the PTF location at the corresponding instants of time are known.

Figure 3 shows the values of the Nu number calculated according to expression (13) for different instants of time and a surrounding water temperature of 2°C. The time-averaged value $\overline{Nu}$ for this set of conditions was equal to 5.6. Investigating the melting of ice balls in standing water, Dumoré [13] found that the Nu value corresponding to this temperature was equal to 6.8. Considering that the estimate of the PTF radius, averaged with respect to the pipe diameter $\bar{r}_f$ is highly approximate due to the nonsymmetric buildup of ice on the pipe, we can assume that our Nu value agrees with the experimental data given in [13].

In Figs. 2 and 3, our experimental data are compared with the results of the numerical solution. In our opinion, there is good agreement. The scatter of experimental points on both sides of the theoretical curve in Fig. 3 is in all probability due to thermal stratification. The faster buildup of ice on top of the tube is probably due to the fact that for water at a temperature ≤4°C, the density of water diminishes with further cooling. As a result, water at a lower temperature is found above the pipe, which explains the somewhat larger thickness of the built-up layer in comparison with the theoretical value (as the water temperature drops from 2°C to 0°C, the value of the Nu number decreases from 6.8 to 2 [13]).

Figure 4 shows the results of the numerical solution of our problem for a pipe with a diameter of 219 mm at $T_g = -60$°C. The suitable Nu values were obtained directly from the experimental curve Nu = f(T_0) given in [13], while the value of α_g was assumed to be 42 W/m²•°C. It is evident from Fig. 4 that an increase in the water temperature leads to a reduction in the solidified layer thickness, all other conditions being equal. All the curves have the same character, namely the thickness of the developing ice layer first increases rapidly, after which the growth slows down and, finally, as is indicated by the bottom curve, stops altogether, i.e., in the presence of free convection, the congealment of the medium does not continue indefinitely, as in the case of a liquid which is not superheated, but, on the contrary, stops after a finite time interval has elapsed. This is also indicated by an analysis of Eqs. (8) and (11), which indicate that the amount of heat transferred by thermal conductivity decreases, while the amount of heat supplied by free convection increases, with an increase in r_f. Equilibrium naturally sets in at some time, and the congealment process stops altogether. It is clear that the zero-growth state is reached more rapidly with an increase in T_0 (all other conditions being equal). By choosing suitable values of the temperature drop, we can probably obtain a frozen layer of any size.

The coefficient of heat transfer from water to the ice surface following the establishment of equilibrium between these fluxes is defined by

$$\alpha_{fe} = \frac{\lambda_1 (T_f - T_w)}{\bar{r}_f (T_0 - T_f) \ln \left(\frac{\bar{r}_f}{r_{ou}} \right)} . \tag{13a}$$

For T_0 = 20°C this value was found to be 13.5 W/m²·°C. Then, the value of the Nusselt number, defined as $Nu = 2\alpha_{fe}\bar{r}_f/\lambda_2$, is equal to 16.2. The Nu value obtained from the experimental curve given in [13] in this case amounts to 16, i.e., it is very close to the theoretical value.

The attempt at determining Nu from the relationships given in [13] yielded for a pipe with a diameter of 219 mm results exceeding by a large factor the experimental data obtained by Dumoré et al. for ice balls. Calculations based on this relationship have shown that results agreeing with these data are obtained only for pipes with a diameter ≤10 mm, i.e., the relationship proposed in [13] cannot be used for determining the Nu value in the case of the icing-over of large-diameter pipes. At the same time, the experimental curve $Nu(T_0)$ obtained in investigating the melting of ice balls in standing water can be used for calculating the icing-over of pipelines.

NOTATION

r, present radius; r_{ou} and r_{in}, outside and inside radii of the pipe; T_1, C_1, a_1, λ_1, and γ_1, temperature, specific heat, thermal diffusivity coefficient, thermal conductivity coefficient, and density of the solid phase, respectively; T_2, λ_2, and γ_2, temperature, thermal conductivity coefficient, and density of the liquid phase, respectively; λ_{wa}, thermal conductivity coefficient of the pipe wall; r_f, radius of location of the phase transition front; k, coefficient of heat transfer from the pipe's outside surface to the air passing through the pipe; α_f, coefficient of heat transfer from water to the phase transition surface by free convection; α_g, coefficient of heat transfer from the pipe's inside surface to air; L, specific heat of phase transition.

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EXPERIMENTAL STUDY OF EVAPORATIVE COOLING IN RODS MADE OF CAPILLARY-POROUS NICHROME

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Results are presented from an experimental study of evaporative liquid cooling in rods of capillary-porous Nichrome with a different structural characteristics in the case of a volumetric heat release ranging from $2 \cdot 10^7$ to $4 \cdot 10^8$ W/m^3.

The high efficiency and high rate of heat transfer in systems with evaporative cooling explain the considerable scientific and practical interest in such systems. Experimental results are particularly important for constructing a proper physical model of the evaporation of a liquid in capillary-porous materials. Published works on the porous cooling of materials with volumetric heat release do not contain any data on such systems in the case where evaporation occurs through the lateral surface of a porous cylindrical specimen in which capillary makeup was accomplished through the ends. There has also been no study of the cooling of a previously impregnated specimen without makeup. Such a system differs significantly from conventional systems [1, 2], in which coolant is pumped through a porous layer and evaporation occurs at the opposing surface.

The goal of the present experimental study is to investigate liquid evaporation from porous cylindrical Nichrome rods in relation to the structural characteristics, the energy supplied, and the conditions of delivery of the liquid being evaporated. The formulation of the problem and the experiment is dictated by the problem of determining the limiting values of volumetric heat release in a porous material with a continuous makeup of the evaporating liquid as a result of capillary forces. Such information can be of practical value in several instances involving heat transfer in capillary-porous materials.

The method of powder metallurgy was used to make porous cylindrical specimens 8 mm in diameter and 80 mm in length (Table 1). The porosity of the finished specimens was determined by the gravimetric method. Maximum pore size was determined by the method of displacing liquid from an impregnated specimen.

Evaporative cooling was studied on the experimental unit shown in Fig. 1.

TABLE 1. Characteristics of Capillary-Porous Structures of Nichrome (80% Ni, 20% Cr)

Number of group	Fraction of original powder, μm	Porosity, %	Maximum pore radius, μm
1	0—40	52	6
2	40—75	60	12
3	75—125	64	20

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