LITERATURE CITED

1. M. P. Malkov, I. B. Danilov, A. G. Zel'dovich, and A. B. Fradkov, Textbook on Physicotechnical Principles of Cryogenics [in Russian], Énergiya, Moscow (1973).
2. L. A. Novitskii and I. G. Kozhevnikov, Thermophysical Properties of Materials at Low Temperatures (Handbook) [in Russian], Mashinostroenie, Moscow (1975).
3. I. I. Perepechko, Low-Temperature Properties of Polymers [Russian translation], Khimiya, Moscow (1977).
4. A. P. Butler, G. B. James, B. I. Maddock, and W. T. Norris, "Improved pool-boiling heat transfer to helium from heated surfaces and its application to superconducting magnets," Int. J. Heat Mass Transfer, 13, No. 1, 105-115 (1970).
5. I. P. Vishnev, Ya. G. Vinokur, and V. V. Gorokhov, "Crisis nucleate boiling of helium at various surfaces," Inzh.-Fiz. Zh., 28, No. 2, 223-230 (1975).
6. D. N. Lyon, "Boiling heat transfer and peak nucleate boiling fluxes in saturated helium between λ-point and critical temperatures," Int. Adv. Cryogen. Eng., 10, 371-379 (1965).
7. V. A. Vasil'ev, Yu. P. Dmitrevskii, S. S. Kozub, and U. Ésher, "Temperature effects in superconducting pulse magnet," Preprint, Inst. Fiz. Vys. Energ., Serpukhov (1979).
8. I. P. Vishnev, A. I. Krauze, S. I. Sergeev, et al., "Studies of heat transfer and hydrodynamics in helium for cooling of superconductor devices," in: Reports to Scientific—Technical Conference on Equipment for Cryogenic Cooling of Superconducting and Cryoresistive Electrical Transmission Lines and Electrical Machinery with Superconductor Windings, Institute of Electrical Engineering, Slovak Academy of Sciences, Bratislava (Czechoslovakia), June 12-16 (1978), pp. 96-122.
9. I. P. Vishnev, Ya. G. Vinokur, O. A. Sedov, et al., "Study of certain problems pertaining to boiling of helium," in: Reports to Sixth All-Union Confer. on Heat Transfer and Hydraulic Drag during Flow of Two-Phase Stream through Components of Electrical Machines and Apparatus, Section I, Leningrad, September 24-26 (1978), pp. 75-77.
10. M. Jergel and R. Stevenson, "Static heat transfer to liquid helium in open pools and narrow channels," Int. J. Heat Mass Transfer, 14, No. 12, 2099-2107 (1972).
11. I. A. Glebov, I. P. Vishnev, V. G. Pron'ko, and I. A. Filatov, "Criterion of thermal stabilization in superconducting devices," Izv. Akad. Nauk SSSR, Energ. Transport, No. 4, 3-9 (1980).
12. I. P. Vishnev, Ya. G. Vinokur, and V. V. Gorokhov, "Boiling of helium under various pressures," Khim. Neft. Mashinostr., No. 9, 18-21 (1975).

PECULIARITIES OF CHILLING AN ARRAY OF PARALLEL CRYOGENIC PIPES

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The process of chilling parallel cryogenic pipes by a uniflow stream is analyzed with heat transfer between them taken into account.

Modern cryogenic power equipment often constitutes an array of parallel channels. Chilling of such equipment, i.e., dropping of its temperature from initial down to operating or some intermediate level is effected principally by means of a gaseous cryogenic coolant.

The object of this study is to determine the effect of heat transfer between pipes on the time taken to chill the equipment. The heat transfer can be effected in various modes: heat conduction through residual gases, radiation, and also heat passage through "thermal bridges." Thermal bridges in real structures are provided, for instance, by a dielectric layer between current-carrying components to be cooled, various electrically insulating spacers, etc.

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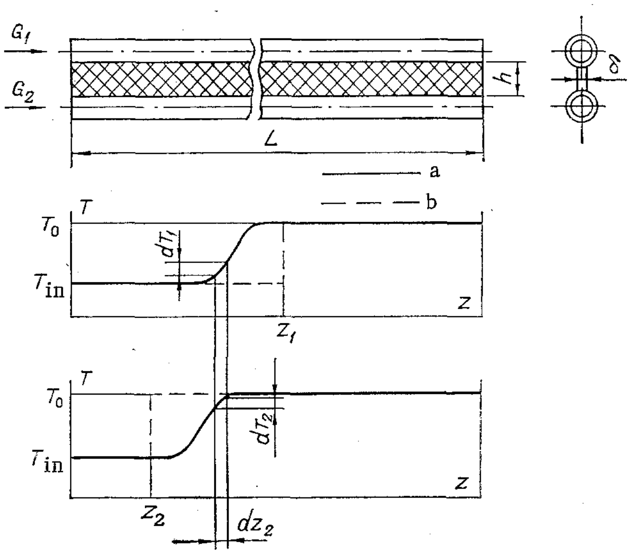


Fig. 1. Schematic diagram for calculation of chilling process: (a) S-form longitudinal temperature profile in channel, (b) jumpwise longitudinal temperature profile.

A schematic diagram for calculation of the chilling process is shown in Fig. 1. Here heat transfer between parallel channels occurs through a layer of thermally insulating material over the entire length of the channels. The chilling problem for this configuration is formulated as follows: into the pipes at some initial temperature enters a gaseous cryogenic coolant, and its temperature at the inlet does not change throughout the chilling process; we are to determine how much time it takes till the pipes reach the temperature of this coolant.

In the general case this problem can be expressed as a system of nonlinear partial differential equations written for each channel and the thermal bridge between them [1]. An analytical solution to the problem has, after a few simplifications characteristic of uniflow heat exchangers, been obtained in the mapping region [2, 3]. A complete solution of the problem is possible only by certain numerical methods [1], but already for only two channels this requires intricate special programs. The application of numerical methods to a larger number of parallel channels is difficult, even with the aid of a third-generation computer.

When the channels can be regarded as "long" ones in terms of chilling (superconducting and cryoresistive magnet structures of the circulation type, superconducting and cryoresistive cables), however, then a simple physicomathematical model will make it possible to obtain an analytical solution to the problem for two channels and to extend that solution, within a wide scope, to the case of several parallel channels. It has been demonstrated [4-7] that a single pipe can be regarded as a long one and the "temperature jump" model will be applicable to it for calculation of the chilling process when the heat transfer zone is much shorter than entire channel. In the first of these studies [4] a channel has been shown to be long when its dimensionless length, equal to the Stanton number, satisfies the inequality

$$\alpha PL/Gc_p > 80. \tag{1}$$

In two studies [4, 7], building on the ideas of the authors of studies [8-10], it has been noted that the cooldown time for a long pipe can be tentatively split into two periods, the dimensionless temperature $V = (T - T_{in})/(T_0 - T_{in})$ at the end of the pipe understood to be changing from 1 to 0.5 in the first one and from 0.5 to 0 in the second one.

We will now demonstrate the possibility of applying the "temperature jump" model and thus also the two-period representation to a multichannel array. On the diagram in Fig. 1 is shown the jumpwise change in the temperature of the refrigerant somewhere along the pipes at a certain instant of time when no heat transfer between pipes occurs. For specificity, let us assume that the cooldown rate is faster for the first pipe than for the second one, which will make the coordinate of the chill front larger in the first pipe than in the second one ($z_1 > z_2$). Heat transfer between the pipes will cause an additional amount of heat to move from the second pipe to the first one and, consequently, the second pipe to chill at a faster rate while a transition from jumpwise to S-form temperature profile occurs along the chill front.

For obtaining analytical relations we consider the case of constant refrigerant flow rates in both pipes throughout the cooldown time. We also assume a uniform flow of refrigerant without heat leakage from the ambient medium, constant thermophysical properties of refrigerant and pipe walls, and a mass of heat conducting material which is either negligibly small or evenly divided between the pipes. Application of the "temperature jump" model makes it possible to replace two energy equations, one for a pipe wall and one for the gas, with a single equation of heat balance. On the basis of an assumed negligibly small mass of thermal insulation material between the channels, one can replace the differential equation of heat conduction for the latter with the equation of stationary thermal flux.

Under the assumed conditions, the system of equations describing the chilling of an array can be written as

$$\begin{gathered}
M_1 c_{m1} \frac{\partial T_1}{\partial \tau} + G_1 c_p \frac{\partial T_1}{\partial \bar{z}} - \frac{T_2 - T_1}{R_{\text{T}}} = 0; \\
M_2 c_{m2} \frac{\partial T_2}{\partial \tau} + G_2 c_p \frac{\partial T_2}{\partial \bar{z}} - \frac{T_2 - T_1}{R_{\text{T}}} = 0; \\
T_1(0, \bar{z}) = T_2(0, \bar{z}) = T_0;\ T_1(\tau, 0) = T_2(\tau, 0) = T
\end{gathered} \tag{2}$$

where $R_T = h/\lambda L\delta$.

For solving the problem, we apply to the fundamental system of equations the double Laplace transformation with respect to variables τ and $\bar{z}$. A transformation with respect to the space coordinate is possible, because the boundary conditions here have been stipulated at the entrance only and because each channel can be regarded as a semiinfinitely long one. The transfer function with respect to temperature is a rather intricate one, and an inverse transformation is most expediently performed by the method of integral estimate [2]. As the approximating function we use

$$W(p) = \exp(k_1 p) \frac{k_2}{k_3 p + 1}, \tag{3}$$

where k_1, k_2, and k_3 can be determined from the initial conditions with both the first derivative and the second derivative of the transfer function taken into account. The simplest solution is obtained with only the first derivative taken into account. The first period of the chilling process is then

$$\tau_f = \frac{M_2 c_{m2}}{G_2 c_p} \frac{1 + \dfrac{M_1 c_{m1}}{M_2 c_{m2}}}{1 + \dfrac{G_1}{G_2}} + R_{\text{T}} M_2 c_{m2} \frac{G_1}{G_2} \frac{\dfrac{M_1 c_{m1}}{M_2 c_{m2}} - \dfrac{G_1}{G_2}}{\left(1 + \dfrac{G_1}{G_2}\right)^2} \ln 0.5. \tag{4}$$

The second period can be determined according to the relation [4]

$$\tau_s = 52.5 M_2 c_{m2} / \alpha_2 L P_2. \tag{5}$$

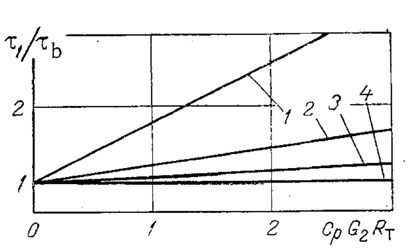
The thermophysical properties appearing in Eq. (5) are calculated corresponding to the refrigerant temperature at the pipe entrance.

The first term on the right-hand side of Eq. (4) represents the time based on the equation of heat balance between total mass of pipes and total mass flow rate of refrigerant:

$$\tau_b = \frac{M_1 c_{m1} + M_2 c_{m2}}{(G_1 + G_2) c_p}. \tag{6}$$

This is the shortest cooldown time. The second term on the right-hand side of Eq. (4) characterizes the effect of interchannel heat transfer on the chilling process. It follows from this equation that the cooldown time is equal to the heat balancing time in two cases: when the gas flow rates in the channels are proportional to the masses of the latter and when there is no thermal resistance between channels. The dependence of the relative cooldown time on these two factors is shown in Fig. 2.

In the dependence of the cooldown time on the thermal resistance between channels one can



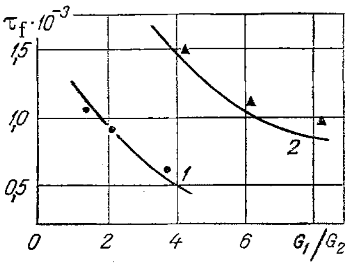


Fig. 2.   Fig. 3.

Fig. 2. Cooldown time for array, calculated for the condition $M_1 = M_2$: 1) $G_1/G_2 = 0.25$; 2) 0.5; 3) 0.7; 4) 1.0.

Fig. 3. Theoretical and experimental cooldown time for an array under the conditions $M_1 = M_2$, R_T = idem, G_2 = idem, and with various ratios G_1/G_2: 1) $G_2 = 2.85 \cdot 10^{-5}$ kg/sec; 2) $1.04 \cdot 10^{-5}$. $\tau_f \cdot 10^{-3}$ sec.

distinguish three ranges. In the first range the thermal resistance is low and the length of the first period of the chilling process differs by less than 10% from the heat balancing time ($\tau_f/\tau_b < 1.1$). This condition prevails when

$$R_{T1} \leqslant \frac{0.1}{G_1 c_p \ln 0.5} \; \frac{\left(1 + \frac{M_1 c_{m1}}{M_2 c_{m2}}\right)\left(1 + \frac{G_1}{G_2}\right)}{\frac{M_1 c_{m1}}{M_2 c_{m2}} - \frac{G_1}{G_2}} \tag{7}$$

In the second range the dependence on the thermal resistance is moderately strong and relation (4) must be used for determining the cooldown time. We will regard the channels independent of one another, thermally, when the length of the first period of the array chilling process differs by less than 10% from the time for the worst channel ($\tau_2/\tau_f < 1.1$). It can be demonstrated that this condition prevails when the thermal resistance between channels is

$$R_{T2} \geqslant \frac{1}{G_1 c_p \ln 0.5} \; \frac{0.91\left(1 + \frac{G_1}{G_2}\right) - 1 - \frac{M_1 c_{m1}}{M_2 c_{m2}}}{\frac{M_1 c_{m1}}{M_2 c_{m2}} - \frac{G_1}{G_2}} \left(1 - \frac{G_1}{G_2}\right). \tag{8}$$

On the basis of inequalities (7) and (8), one can determine the effect of interchannel heat transfer on the cooldown time for several parallel channels. Indeed, when condition (7) is satisfied for all channels, one can determine the cooldown time from the heat balance in the array within a 10% accuracy. When condition (8) is satisfied, then the cooldown time for the array is determined by the cooldown time for the "worst" channel and the error of calculation also does not exceed 10%. When the thermal resistance between channels is somewhere between R_{T1} and R_{T2}, then other methods must be used for determining the cooldown time for multichannel arrays.

In order to confirm these conclusions, a test cell was chilled in a series of experiments. The pipes of this cell were made of 12-m-long copper tubing 4 mm in diameter and 0.5 mm thick. Heat transfer between pipes was effected by conduction along a Teflon film wrapped around one of the pipes. The thermal resistance of this Teflon film in vacuum was determined in a special series of experiments and found to be 0.032 K/W.

The experimental data are compared in Fig. 3 with theoretical results according to relation (4). The maximum deviation of theoretical values from experimental ones is 20%, which can be attributed to the approximate determination of coefficients k_1, k_2, and k_3 in expression (3) as well as to the conditions of the experiments with the dimensionless length of pipes within the intermediate range ($10 < \alpha PL/Gc_p < 80$) [4].

It is worthwhile to compare our approximate analytical solution not only with experimental data but also with results of a numerical solution on the basis of the complete mathematical model. In [1] the process of chilling a coaxial superconductor cable, schematically shown here in Fig. 4, was analyzed on this basis. The cooldown time for such a cable under

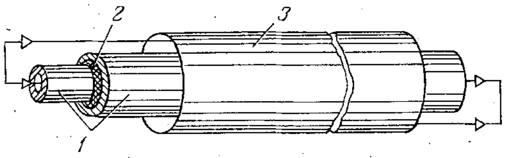


Fig. 4. Basic layout of LASL/PECO cable: single-phase coaxially shielded cable 1, electrical insulation 2, cable sheath 3.

various modes of operation was evaluated by numerical analysis. It has been demonstrated, for instance, that an LASL/PECO cable (variant C) chills from 300 to 10°K in 75 h with a refrigerant flow rate of 25 g/sec. Assuming the refrigerant flow rate to be divided between coaxial channels proportionally to their cross-sectional areas, the minimum dimensionless length of a channel will be approximately 2000 and, in accordance with condition (1), the channels can be regarded as long ones. The thermal resistance of the insulation between channels is theoretically $0.3 \cdot 10^{-3}$ °K/W, which is two orders of magnitude lower than R_{T1} for this case. Therefore, both channels should be chilling simultaneously and their cooldown time can be found according to relation (6). Indeed, the second term in expression (4) is equal to only 0.21 h and thus negligible in comparison with $\tau_b = 72.2$ h. The second period in the chilling process is 2.95 h, according to relation (5), and the total cooldown time is 75.15 h, which agrees almost exactly with calculations.

Thus, both an experiment and a comparison with the exact solution have confirmed the correctness of using the proposed physicomathematical model and the resulting analytical relations for calculation of the cooldown time for parallel channels.

Insertion of special thermal bridges between parallel channels with uniflow of the refrigerant can be recommended as means of reducing their cooldown time.

NOTATION

Here α is the heat transfer coefficient; c_p, c_m, mean-integral specific heat of the refrigerant and the channel wall material respectively; δ, thickness of a "thermal bridge"; h, height of a thermal bridge layer; G, mass flow rate of refrigerant; k_1, k_2, k_3, coefficients; λ, thermal conductivity; L, length of a channel; M, mass of metal in a channel wall; P, channel perimeter; R_T, thermal resistance of a thermal bridge; τ, cooldown time; T, stream temperature; V, dimensionless temperature; z, longitudinal coordinate; $\bar{z} = z/L$, dimensionless longitudinal coordinate; p, variable in the Laplace transform. Subscripts: 1, channel with faster cooldown rate; 2, channel with slower cooldown rate; 0, initial state; in, entrance; b, balance; f, first period; and s, second period.

LITERATURE CITED

1. M. C. Jones, "Cooldown of superconducting power transmission lines with single-phase helium," in: Superconducting Electrical Transmission Lines [Russian translation], Energ. Inst., Moscow (1979), pp. 139-166.
2. A. A. Shevyakov and R. V. Yakovleva, Engineering Methods of Calculating the Dynamics of Heat Exchangers [in Russian], Mashinostroenie, Moscow (1968).
3. I. Chermak, V. Peterka, and I. Zavorka, Dynamics of Regulated Systems in Heat Power and Chemical Engineering [Russian translation], Mir, Moscow (1972).
4. S. P. Gorbachev, A. A. Krikunov, and O. A. Goncharuk, "Transient chilling of superconductor cable by gaseous cryogenic coolant," in: Superconducting Electrical Transmission Lines [in Russian], Izd. Energ. Inst., Moscow (1969), pp. 47-56.
5. V. E. Keilin, I. A. Kovalev, and S. A. Lelekhov, "Cooling time for circulation systems," Inzh.-Fiz. Zh., 27, No. 6, 1081-1089 (1974).
6. A. M. Baron, V. M. Eroshenko, and L. A. Yaskin, "Studies of cooldown of cryogenic cables," Cryogenics, 17, No. 3, 161-166 (1977).
7. S. P. Gorbachev, A. A. Krikunov, and O. A. Goncharuk, "Cooldown time for cryogenic array with gaseous refrigerant," Inzh.-Fiz. Zh., 39, No. 4, 629-635 (1980).
8. E. P. Serov and B. P. Korol'kov, Dynamics of Steam Generators [in Russian], Énergiya, Moscow (1972).
9. E. P. Serov and B. P. Korol'kov, Dynamics of Processes in Heat and Mass Exchangers [in Russian], Énergiya, Moscow (1967).

10. B. P. Korol'kov, Special Functions for Mathematical Analysis of Transient Heat Transfer Dynamics [in Russian], Nauka, Moscow (1976).

HEAT TRANSFER AND VAPORIZATION IN THE BOILING OF A SURFACTANT SOLUTION IN STEAM GENERATORS

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Results are presented from an experimental study of heat transfer and vaporization in the boiling of a solution of octadecylene under atmospheric pressure at different heat fluxes and concentrations.

There has recently been a trend toward the use of surface-active agents (SAA) in steam turbines. This development is related to the need to actively influence the process of vapor expansion and control the structure and characteristics of high-velocity flows of supersaturated and moist vapor, which leads to an increase in turbine efficiency and a decrease in the erosion of the through parts of the turbine. One procedure is the introduction of the surfactant octadecylene (ODA) into the vapor flow ahead of the turbine [1-3]. After condensation of the vapor, the ODA may be returned together with the condensate to the turbine. Such use of ODA raises the need to examine the effect of its addition on heat transfer and vaporization on the heating surface. Heat transfer was found to be improved in earlier experiments [4, 5] in which different surfactants were added to a boiling medium. This result is explained by a reduction in the surface tension of the solution and a corresponding increase in the number of centers of vapor formation on the heating surface and the frequency of bubble generation, which leads to intensive agitation of the thermal boundary layer.

In our studies, we used an experimental unit which simulated a vaporizing apparatus of the boiling type [6]. The heating surface was a horizontal copper tube 34 mm in diameter and 120 mm in length, enclosing an electric heater. The vaporizing chamber, 9 liters in volume, was placed in an air thermostat. The dynamics of vapor formation were studied visually and by filming in two projections. Heat flux was determined from the electric power supplied to the working section, and the temperature of the heating surface was measured by the compensation method using six thermocouples placed in the heating surface about its perimeter. The error of the heat-transfer coefficient measurement was no greater than 8-12%. The concentration of ODA in the solution was determined by measuring the intensity of methyl orange coloration of the ODA reaction product extracted with chloroform. The ranges of the parameters: pressure 100 kPa, heat flux 40-120 kW/m^2, ODA concentration 0.3-70 mg/liter.

The main tests were conducted with open-cycle operation of the unit, i.e. the condensate from the condenser was discarded and the level in the vaporization chamber was maintained by making up the solution with distillate heated to 1-2°K below the saturation temperature. Since ODA volatilizes with steam [7], it was also removed from the unit with the condensate. During the experiment, we also measured the temperature of the wall of the heating tube and sampled the solution to analyze the ODA concentration.

The tests with the ODA solution were conducted by vaporizing it over 2-3 h with an initial concentration of 50-70 mg/liter. The tests established that boiling of the solution for 0.5-1.5 h reduces the concentration of ODA in the solution to 0.3-0.5 mg/liter, which corresponds to the sensitivity of the method used to determine the concentration (Fig. 1). The concentration of ODA in the solution decreased quite a bit more slowly in the tests with closed-cycle operation of the unit (return of the condensate from the condenser to the vaporizing chamber), since the ODA was returned to the chamber along with the condensate.

The ODA is vaporized more rapidly from the solution with an increase in heat flux from

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