their mean values $\bar{t}_1$, $\bar{t}_2$, $\bar{t}_3 = t_{t.s}$; $t_{t.s}$, set temperature of the sensor; $\bar{P}$, P^{max}, mean and maximal power, respectively, of the final control element; T, period of free oscillations; τ_1, τ_2, cooling and heating time, respectively, of the chamber; A_1, A_2, amplitudes of the temperature oscillations of the object and of the chamber, respectively; γ, duty factor of the operation of the final control element.

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SIMULATING THE COOLING OF SPIRAL COMPONENTS IN CIRCULATION SYSTEMS FOR GAS COOLING. PART 1. SINGLE PANCAKE COIL

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A study has been made of the effects of design and thermal parameters on the temperature patterns and cooling times for pancake coils.

A large superconducting magnet with circulating coolant is frequently built up from two-layer disk sections [1-3]; each layer or pancake is a flat (archimedean) spiral formed out of insulated hollow wire and embedded in epoxide resin. The cooling channels in adjacent pancakes are usually connected in parallel. The pancake coils in equipments may differ in design, conductor length, number of turns, and insulating material and thickness. There is heat transfer through the insulation between turns and between coils, which sometimes has a substantial effect on the cooling. The mode of cooling must be chosen such that no dangerous thermal stresses arise, while the cooling time and coolant consumption are acceptable. It is possible to choose a state meeting these requirements by solving the nonstationary conjugate heat-transfer problem. The term conjugate here incorporates the fact that it is necessary to solve the energy-conservation equations together for all the components in the heat-transfer system (channel walls and flows) [4, 5].

The cooling of a single adiabatic channel has been examined in most detail (with ideal insulation between turns for a spiral). If the thermal parameters and coolant flow rate are constant, one can obtain an analytic solution if the coolant temperature at the inlet changes stepwise [4]. To allow for the change in heat-transfer coefficient along the channel and for the temperature dependence of the thermophysical parameters, one has to use numerical methods such as [6, 7] to solve the problem with general boundary conditions.

It is recommended [8, 9] that the dimensionless parameter $St^* = \alpha\Pi L/(Gc_p)_g$ should be used in distinguishing long channels ($St^* \geq 100$) from short ones ($St^* \leq 10$); long means that the zone of rapid heat transfer is substantially shorter than the channel, so one can use a temperature-step model to calculate the cooling [10], which can be used with a coolant inlet temperature step to estimate the cooling time from $\tau_b = (M_c)_w/(Gc_p)_g$, which follows from the heat-balance equation, and also to determine the pressure drop or coolant flow rate. A formula has been given [9] for the cooling time for a short channel.

An analytic solution can also be obtained [5, 11] for two parallel long channels with thermal interaction; solutions have been obtained for direct-flow and countercurrent forms of coolant motion for constant thermophysical parameters of the coolant wall, infinitely small thermal capacity of the bridge between channels, and inlet coolant temperature steps. In

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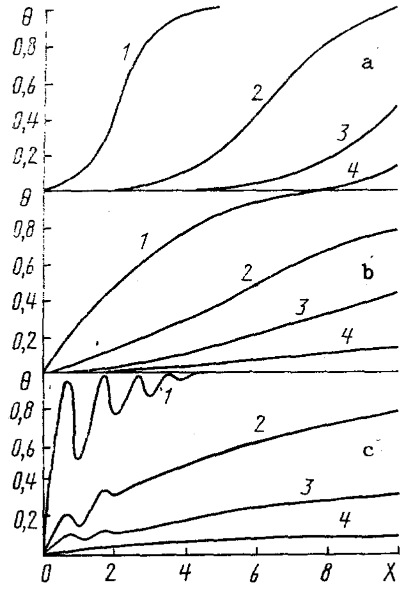

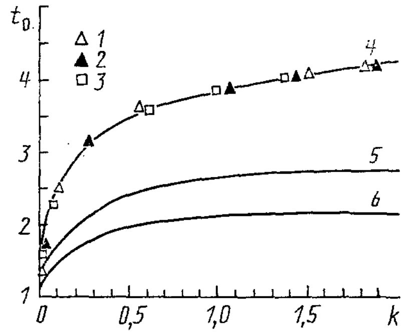

Fig. 1 Fig. 2

Fig. 1. Effects of $K = \lambda HL/((Gc_p)_g \delta N)$ on the temperature patterns on cooling a spiral channel ($St^* \geq 100$, N = 10) with a step change in inlet coolant temperature: a) K = 0.1; 1) t = 0.2; 2) 0.6; 3) 1.0; 4) 1.4; b) K = 1.5; 1) t = 0.2; 2) 0.6; 3) 1.0; 4) 1.6; c) K = 7; 1) t = 0.05; 2) 0.6; 3) 1.0; 4) 2.4.

Fig. 2. Effects of $k = \lambda H\ell((Gc_p)_g \delta N)$ on the dimensionless cooling time for a spiral channel: 1) N = 10; 2) 30; 3) 50; 4) $\Theta_{max} = 0.01$; 5) 0.05; 6) 0.1.

[7], a similar problem was solved numerically with more general assumptions. In [11], a study was made of the effects of heat transfer between long channels on the cooling in the presence of hydraulic nonuniformities. A formula was recommended for estimating the cooling time as a function of the thermal resistance in the insulation between channels.

A method has been given [12-14] for calculating the cooling in cryostatic systems having spiral channels; here there are some special features by comparison with those in adiabatic channels on account of the heat transfer between the turns via the insulation and the periodic spiral structure. Simple estimates show that the heat leak between turns in the superconducting magnet substantially exceeds the heat flux due to longitudinal conduction and may be comparable with the convective component. In [13, 14], calculations were compared with experiment for an experimental component simulating a pancake coil for a superconducting winding producing a toroidal field in a T-15 thermonuclear system.

Here we examine the effects from various thermophysical and working parameters with allowance for the nonuniformity in coolant distribution in parallel spiral pancakes. The first part of the paper deals with a single spiral pancake coil.

As there is good agreement between theory and experiment [13, 14], the cooling for a single pancake at a constant flow rate (G = constant) may be described by the system

$$\begin{cases} (mc)_w \dfrac{\partial T_w}{\partial \tau} = \alpha\Pi (T_g - T_w) + q_c + q_L; \\ (Gc_p)_g \dfrac{\partial T_g}{\partial x} = \alpha\Pi (T_w - T_g); \end{cases} \tag{1}$$

$$T_w(x,\ 0) = T_{ini};\ T_g(0,\ \tau) = \varphi(\tau),$$

where

$$q_c = \frac{T_{w,i-1} - 2T_{w,i} + T_{w,i+1}}{R_T}$$

is the heat influx to turn i from adjacent turns i − 1 and i + 1. Here the thin-tube approximation is used for the solid, while a one-dimensional flow model is used for the coolant. The x coordinate is directed along the channel. Heat transfer between the channel wall and the coolant is determined by the heat-transfer coefficient α, while the heat leaks between adjacent turns are determined by the thermal resistance per unit length of the conductor $R_T = \delta/(\lambda H)$.

For St* ≥ 100 (1) can be simplified [13, 14]:

$$(mc)_w \frac{\partial T_w}{\partial \tau} + (Gc_p)_g \frac{\partial T_w}{\partial x} = q_c + q_L; \qquad T_w(x,\ 0) = T_{ini};\ T_w(0,\ \tau) = \varphi(\tau). \tag{2}$$

We reduce (1) and (2) to dimensionless form to analyze the effects of the factors on the cooling, where we follow [15] in using the quantities appearing in the uniqueness conditions as scale factors. The spatial scale is provided by the mean length l of a spiral turn via l = L/N (l determines the heat-transfer surface between adjacent turns), while the time scale is provided by the balance cooling time for the channel $\tau_b = (Mc)_w/(Gc_p)_g$, and the temperature-difference scale is provided by $\Delta T - T_{ini} - T_{fin}$. We introduce the dimensionless excess temperatures of the wall $\Theta = (T_w - T_{fin})/(T_{ini} - T_{fin})$ and the coolant $\vartheta = (T_g - T_{fin})/(T_{ini} - T_{fin})$, as well as the dimensionless independent variables $t = \tau/\tau_b$ and $X = x/\ell$; we substitute these into (1) and (2) to get

$$\begin{cases} \dfrac{1}{N}\dfrac{\partial \Theta}{\partial t} = \dfrac{\mathrm{St}^*}{N}(\vartheta - \Theta) + Q_c + Q_L; \\ \dfrac{\partial \vartheta}{\partial X} = \dfrac{\mathrm{St}^*}{N}(\Theta - \vartheta); \end{cases} \qquad \Theta(X,\ 0) = 1;\ \vartheta(0,\ t) = \varphi_1(t); \tag{3}$$

for St* ≥ 100

$$\frac{1}{N}\frac{\partial \Theta}{\partial t} + \frac{\partial \Theta}{\partial X} = Q_c + Q_L;\ \ \Theta(X,\ 0) = 1;\ \Theta(0,\ t) = \varphi_1(t), \tag{4}$$

where

$$Q_c = \frac{q_c l}{(Gc_p)_g \Delta T};\ Q_L = \frac{q_L l}{(Gc_p)_g \Delta T};\ \ Q_c = K(\Theta_{i-1} - 2\Theta_i + \Theta_{i+1});$$

$$K = \frac{l}{(Gc_p)_g R_T} = \frac{\lambda H L}{(Gc_p)_g \delta N}.$$

We consider the generalized independent variables and the dimensionless parameters appearing in the description. As t is normalized to τ_b, the time required for an adiabatic channel to be cooled with the ideal use of cold, the meaning of the dimensionless cooling time is the relative excess of the actual cooling time over the value most favorable from the viewpoint of coolant consumption.

The variable $X = x/\ell$ is normalized to the length of an average turn, so the total dimensionless channel length coincides numerically with the number N of turns.

The definitive dimensionless parameters are St*, Q_L, K, N; the modified Stanton parameter St* characterizes the heat transfer between the channel wall and the coolant. Q_L is the dimensionless heat load per unit length due to external heat leaks. The dimensionless heat flux due to heat transfer between the turns Q_c is dependent on K, which characterizes the rate of heat transfer through the insulation between turns. One can treat K as the dimensionless thermal conductivity of the insulation between adjacent turns.

In [12-14] it has been found that heat transfer between turns has a substantial effect on the temperature patterns in cooling spiral channels; the rate of that transfer determined whether the temperature profile is similar to that for an adiabatic channel or differs considerably from it. If there is sufficiently rapid transfer between turns, one can get a wave-type temperature distribution along the channel, with a peak on the temperature profile in a turn. The amplitude of the temperature wave decreases as time elapses and also away from the inlet. Calculations have been made with variable K from (4) for long channels with various N (N = 10; 20; 30; 40) on the assumption of an inlet coolant temperature step and the absence of external heat leaks ($Q_L = 0$), which has shown that similar temperature patterns arise for the same values of K in spirals differing in number of turns, so one can identify three characteristic ranges in K as regards the effects on the temperature patterns: $K \leq 0.1$ little effect, $0.1 < K < 2$ moderate effect, and $K \geq 2$ marked effect.

When $K \leq 0.1$, the patterns and the cooling time are close to those for the long adiabatic channel, with a temperature wave and with the dimensionless time close to one (Fig. 1a).

In the case of moderate effects, the active heat-transfer zone is comparable in length with the channel (Fig. 1b) and the dimensionless time is about two.

In the region of marked effect, there is a temperature wave in the initial stage (Fig. 1c) and the dimensionless time is greater than two.

Figure 1 shows that the wall temperature at the outlet at the start ($t < 1$) decreases more rapidly than for an adiabatic channel as K increases, which reduces the amount of heat transferred to the coolant in unit time and is the cause of the increase in cooling time.

The value of N affects the cooling rate, which is dependent on K as well as N. The cooling is more rapid if there is high resistance to heat transfer between turns, as that transfer reduces the cooling rate. Calculations with K variable for $St^* \geq 100$ and inlet temperature steps have been performed with (4) for various spirals (N = 10; 30; 50) (Fig. 2), which has shown that the dimensionless time is the same for different spirals if $k = K/N$ is the same.

Here we may note that $k = K/N = \lambda H \ell / ((Gc_p)_g \delta N)$ is the dimensionless heat-transfer coefficient via N turns along the radius of the pancake. The heat flux due to exchange between turns is the same for different spirals if $k = K/N$ is the same for them, and such spirals cool at the same rate and have the same dimensionless cooling time.

As the cooling is asymptotic, the complete cooling time strictly speaking is infinite. Therefore, when one mentions the cooling time, one must also state the maximum wall temperature corresponding to a given instant. Figure 2 shows that the cooling time increases considerably with the depth of cooling. The following relations enable one to estimate the cooling times for long channels with inlet temperature steps and various cooling depths:

$$t_0(\Theta = 0{,}01) = 1{,}24 + 9{,}98k^{0{,}5} - 6{,}18k^{0{,}66} - 1{,}12k^{0{,}33}; \tag{5}$$

$$t_0(\Theta = 0{,}05) = 1{,}14 + 8{,}71k^{0{,}5} - 5{,}12k^{0{,}66} - 2{,}08k^{0{,}33}; \tag{6}$$

$$t_0(\Theta = 0{,}1) = 1{,}04 + 9{,}46k^{0{,}5} - 5{,}47k^{0{,}66} - 2{,}89k^{0{,}33}. \tag{7}$$

The cooling time for an adiabatic channel at high St* is close to the minimal limiting value $\tau_b = (Mc)_w/(Gc_p)_g$; a practical criterion for using this formula is $St^* \geq 100$, since for $St^* < 100$ the cooling time for an adiabatic channel increases because of deterioration in heat transfer from the wall to the coolant and correspondingly reduction in the heating of the latter along the channel. Heat transfer between turns reduces the wall temperature at the outlet but also reduces the coil cooling rate. Figure 3 shows the dimensionless cooling time as a function of St* for various values of k. The calculations have been performed from (3) for an inlet coolant temperature step. The time decreases as St* increases, with the curves tending to two values whose level is dependent on k. The St* at which each curve comes close to the asymptotic value is not more than about 100 for all k; the effects of St* on the cooling time in the range $10 < St^* < 100$ become less pronounced as k increases. The formula for the cooling time for $St^* > 10$, $0 < k < 2$ can then be put as

$$t(St^*, k) = t_0(k) + \Delta t(St^*, k), \tag{8}$$

where $t(St^*, k)$ is that time, $t_0(k)$ is the time for a long channel calculated from (5)-(7), and $\Delta t(St^*, k)$ is the correction for St*.

One can approximate $\Delta t(St^*, k)$ satisfactorily as

$$\Delta t(St^*, k) = A(k)F(St^*), \tag{9}$$

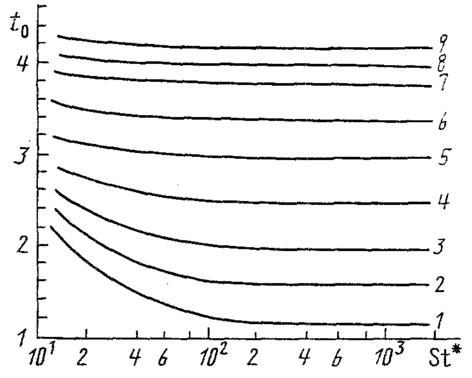

Fig. 3

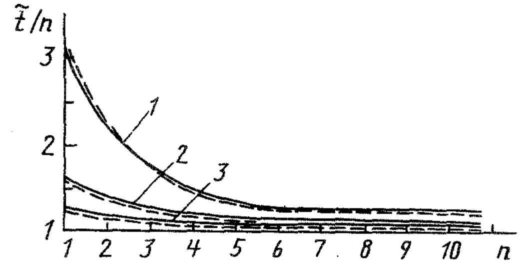

Fig. 4

Fig. 3. Cooling time for a spiral channel down to a maximal temperature Θ_{max} = 0.01 as a function of St* and k: 1) k = 0; 2) 0.02; 3) 0.05; 4) 0.1; 5) 0.2; 6) 0.4; 7) 1.0; 8) 1.3; 9) 2.0.

Fig. 4. Effects of n on cooling time (solid line from (4), dashed line from (10)): 1) K = 10; 2) 1; 3) 0.1.

where

$$A(k) = 0{,}124 - 0{,}27\exp(-k) + 0{,}83\exp(-10k) + 0{,}30\exp(-25k) + 0{,}31\exp(-k^2);$$

$$F(\mathrm{St}^*) = 0{,}51 + 2{,}07\left(\frac{\mathrm{St}^*}{10}\right)^{-2,3} + 0{,}56\left(\frac{\mathrm{St}^*}{10}\right)^{-4,6} - 2{,}13\left(\frac{\mathrm{St}^*}{10}\right)^{-1,15}.$$

A general specification is that the difference between the inlet coolant temperature and the maximum temperature of the spiral should not exceed a permissible value ΔT_{per}, which in some cases requires preliminary cooling with the coolant temperature gradually falling. Out of the modes of inlet temperature variation, the best is that where the temperature difference is somewhat less than the limiting permissible value, i.e., $T_{in} = T_{ini} - \Delta T_{per} + \varepsilon$, where ε is a temperature margin. As the outlet temperature falls, the inlet temperature is gradually reduced in such a way that the maximum temperature difference remains close to $\Delta T = (\Delta T_{per} - \varepsilon)$. The amount of heat extracted is then close to the maximum possible. Figure 4 shows the cooling time as a function of the number of temperature intervals n for various values of K. The value of n is the integer part of $(T_0 - T_{fin})/(\Delta T_{per} - \varepsilon)$. The calculations are based on (4). The dashed line shows calculations on the cooling time for variable inlet temperature from

$$\tilde{t} = n + t_0(k), \tag{10}$$

where t is the dimensionless cooling time for a variable inlet temperature, n is the number of temperature intervals, and $t_0(k)$ is the cooling time for a long spiral for an inlet temperature step as calculated from (5)-(7). Figure 4 shows that (10) agrees satisfactorily with (4); (10) can be used in estimating the minimum cooling time with a variable inlet temperature and a preset permissible temperature difference between inlet and outlet.

These results can be used in analyzing the cooling of a superconducting toroidal winding for the T-15 tokamak. Calculations from (5)-(10) showed that the cooling time with a flow rate $G_g = 0.6 \cdot 10^{-3}$ kg/sec through a pancake is dependent to a considerable extent on n (i.e., on ΔT_{per}) and is 38 h for ΔT_{per} = 70 K, 56 h for ΔT_{per} = 45 K and 78 h for ΔT_{per} = 30 K. A change in the thermal resistance between turns by a factor two has little effect on the cooling time.

NOTATION

c, specific heat capacity, J/(kg·K); G, mass flow rate, kg/sec; H, pancake width, m; i, No. of turn; $K = \lambda HL/((Gc_p)_g \delta N)$, $k = \lambda H\ell/((Gc_p)_g \delta N)$, dimensionless parameters; L, ℓ, cooling channel length and mean length of turn, respectively, m; M, mass, kg; m, mass per unit length of the channel, kg/m; N, number of turns; Q, $\dot{q}$, dimensionless and dimensional (W/m) heat load per unit length; $R_T = \delta/\lambda H$, thermal resistance per unit length of the coil insulation, K·m/W; T, temperature, K; ΔT, temperature difference, K; t, dimensionless time;

X, x, dimensionless and dimensional (m) space coordinates; α, heat-transfer coefficient $W/(m^2 \cdot K)$; δ, thickness of the coil insulation, m; λ, thermal conductivity of the insulation, $W/(m \cdot K)$; $\Theta = (T_w - T_{fin})/(T_{ini} - T_{fin})$, $\vartheta = (T_g - T_{fin})/(T_{ini} - T_{fw})$, dimensionless excess temperature of the wall and coolant, respectively; Π, heat transfer perimeter, m; $\phi(\tau)$, known time function; τ, time, sec; τ_b, balance time of cooling, sec; $St^* = \alpha \Pi L/(Gc_p)_g$ modified Stanton parameter. Subscripts: w, wall; g, coolant; p, at constant pressure; int, fin, initial and finite states, respectively; per, permissible; c, heat flux through the coil insulation; L, heat influx from the surrounding medium; max, maximum; $\sim$, cooling time of the spiral channel with a variable temperature of the coolant at the inlet.

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EXPERIMENTAL DEVICE FOR MEASUREMENT OF ISOBARIC SPECIFIC HEAT OF ELECTROLYTES AT HIGH STATE PARAMETERS

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UDC 536.2.083

Principles of operation and construction of an experimental device for measurement of isobaric specific heat of liquids (electrolytes) at high pressures are described.

The major shortcomings of existing calorimeters were pointed out in [1], which proposed a new pulse-regular method for simultaneous measurement of isobaric specific heat cp, thermal conductivity λ, and thermal diffusivity a of electrolytes. This method permitted measurements over wide ranges of temperature and pressure. In the present study this method is realized experimentally for measuring isobaric specific heat of liquids.

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