For example, for $\eta = 10^{-2}$ P, Re = 0.1, $\rho = 10$ g/cm³, $\omega = 100$ sec^{-1}, $R = 2 \cdot 10^2$ cm, and $\theta = 0.1$, the limiting particle size $a = 2.16$ μ.

In Fig. 1, trajectories are shown for particles of various sizes precipitating from level R_0 to level $r = eR_0$ at various rates of rotation.

These results for particle motion in a field of centrifugal forces may be used to classify powders by a method illustrated in Fig. 2.

The rotor 1 has two cavities: the internal cavity 2 and the sedimentation cavity 3, which are connected by a narrow slit 4. The rotor begins to rotate and cavity 3 is completely filled with a pure sedimentational liquid, for example, water. Then cavity 2 is filled with a suspension containing powder particles. The centrifugal force drives the particles through slit 4 into cavity 3. The initial radius-vector, determining the particle position in a coordinate system fixed in the rotor, is the same for all the particles.

Since Eq. (13) constitutes a one-to-one relationship between the angle of rotation and the particle size, the particles precipitating in a specific receiving chamber 5 will all belong to a specific fraction, the composition of which is determined by the constructional dimensions of the rotor and may be regulated by appropriate choice of the sedimentational liquid and the angular velocity of the rotor.

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DEVELOPMENT OF A STABLE TWO-PHASE POROUS COOLING SYSTEM

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UDC 532.546:536.24

The use of a two-layer porous wall instead of a one-layer wall is shown to create the necessary precondition for the practical realization of a two-phase porous cooling system.

The problem of the stationary two-phase porous cooling of a homogeneous plate is formulated in [1, 2]. The properties of this process are investigated and it is established that it is impossible to create a stable two-phase porous cooling system with a homogeneous wall using water as the coolant. The cause of the instability is the substantial reduction in the flow rate of the coolant with a deepening of the zone of vaporization from the outer surface into the porous plate. From this follows the condition for increasing the stability of the system: the variation in the flow rate of the coolant with a deepening of the vaporization zone must be reduced somehow.

An easily realized method of increasing the stability of a two-phase porous cooling system, consisting of using a two-layer porous wall, is proposed and investigated below. The essence of the method becomes clear when an analogy is drawn between the process described

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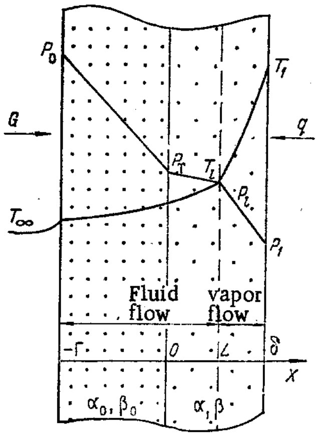

Fig. 1. Physical model of equilibrium two-phase porous cooling of a two-layer plate.

and the process of the vaporization of a fluid in heated channels. The movement of a fluid in evaporation channels is unstable due to a sharp increase in the resistance when the mass vapor content in the two-phase flow is increased [3]. A way of eliminating the aperiodic instability of this process is, however, known: an additional resistance in the form of a choke washer is installed at the channel inlet where a single-phase fluid flows.

For two-phase porous cooling the role of the additional resistance is played by the inner (structural) layer of porous material of thickness Γ (Fig. 1) having viscous α_0 and inertial β_0 coefficients of resistance considerably greater than the corresponding characteristics α and β of the material of the outer (heat-proof) layer. It follows from the very idea of the method that the zone of coolant vaporization must lie in the outer layer: $L \geq 0$.

The parameters α_0, β_0, and Γ of the inner layer must be determined in order to ensure that the cooling system operates stably and safely when the density of the external heat flow varies over a given range. The magnitude of the coefficient of thermal conductivity of the porous material of the outer layer should also be found.

The analysis is carried out for all the assumptions made in [1, 2].

Special Features of Movement of Coolant Vaporized in Outer Layer of a Two-Layer Porous Plate. The movement of the coolant in a porous metal—ceramic material is described by a modified Darcy equation:

$$-\frac{dP}{dX} = \alpha\mu v G + \beta v G^2. \tag{1}$$

It is assumed that the pressure drop $P_0 - P_1$ in the two-layer plate is maintained constant. The L coordinate of the region of vaporization is selected as the parameter which takes its value from the $L \in [0, \delta]$ interval. The total drop is made up of the pressure drops at the individual sections of coolant flow, which differ in the physical properties of the porous material or of the coolant: the inner layer ($-\Gamma < X < 0$) and the fluid ($0 < X < L$) and vapor ($L < X < \delta$) sections of the outer layer.

By integrating the equation of motion, bearing in mind the assumptions about the constancy of the physical properties of the porous material and both phases of the coolant within the bounds of the corresponding sections, and putting the result into a dimensionless form, we obtain

$$1 = g\left[\frac{\alpha_0}{\alpha}\gamma + l + \frac{\nu''}{\nu'}(1-l)\right] + g^2 \mathrm{Re}\left[\frac{\beta_0}{\beta}\gamma + l + \frac{v''}{v'}(1-l)\right], \tag{2}$$

where

$$\gamma = \frac{\Gamma}{\delta};\ l = \frac{L}{\delta};\ g = \frac{G}{G_1};\quad G_1 = \frac{P_0 - P_1}{\alpha\nu'\delta};\ \ \mathrm{Re} = \frac{G_1(\beta/\alpha)}{\mu'}.$$

Relation (2) is a quadratic equation relative to the magnitude of the dimensionless flow rate g. Its solution takes the form

$$g = \frac{m}{2n\,\mathrm{Re}}\left(-1 + \sqrt{1 + 4\mathrm{Re}\frac{n}{m^2}}\right), \tag{3}$$

where

$$m=\left[\frac{\alpha_0}{\alpha}\gamma+l+\frac{v''}{v'}(1-l)\right];\quad n=\left[\frac{\beta_0}{\beta}\gamma+l+\frac{v''}{v'}(1-l)\right]. \tag{4}$$

It should be noted that the individual terms of the right-hand side of Eq. (2) have the sense of a relative pressure drop, caused by the viscous and inertial components of the resistance, at each of the three sections of coolant flow.

The flow conditions in the two-layer porous wall are best determined in terms of the parameters of the movement of the fluid coolant being vaporized on the outer surface (l = 1). Under viscous conditions $\left[\mathrm{Re}<0.01\left(1+\frac{\alpha_0}{\alpha}\gamma\right)^2\Big/\left(1+\frac{\beta_0}{\beta}\gamma\right)\right]: g(1)\left(1+\frac{\alpha_0}{\alpha}\gamma\right)\to 1$; under inertial conditions $\left[\mathrm{Re}>100\left(1+\frac{\alpha_0}{\alpha}\gamma\right)^2\Big/\left(1+\frac{\beta_0}{\beta}\gamma\right)\right]: g(1)\left(1+\frac{\beta_0}{\beta}\gamma\right)^{\frac{1}{2}}\to \mathrm{Re}^{-\frac{1}{2}}$; and under intermediate conditions the magnitude of g(1) is calculated by formula (3).

It should be noted that all the relations obtained here describing the movement of a coolant being vaporized in the outer layer of a two-layer wall contain within themselves the analogous relations derived in [1] for a one-layer wall as a special case when $(\alpha_0/\alpha)\gamma = (\beta_0/\beta)\gamma = 0$.

The influence of the inner layer on the patterns governing the movement of the coolant being vaporized in the outer layer is investigated most easily under viscous conditions of flow (Re → 0) for which the influence of the parameters Re and $(\beta_0/\beta)\gamma$ disappears and the expression for calculating the coolant flow rate is simpler: g = 1/m.

Figure 2 presents the dependences of the relative flow rate of the coolant through a two-layer plate on the coordinate l of the phase-transition surface inside the outer layer. The physical properties of water and water vapor in the saturated state when P_1 = 1 bar are used for the calculation. The flow conditions are viscous. The desired effect of a reduction in the coolant flow rate with a deepening of the region of vaporization from the outer surface into the porous wall is achieved as the $(\alpha_0/\alpha)\gamma$ parameter is increased.

Determining Structural and Thermophysical Characteristics of a Two-Layer Porous Plate. In [2] the conditions for stable and safe operation of a system for the two-phase porous cooling of a homogeneous plate are shown to impose very strict, practically unattainable, restrictions on the magnitude of the effective coefficient of thermal conductivity λ of the vapor section

$$\lambda^{**}<\lambda<\lambda^{*}, \tag{5}$$

where λ* is determined from the stability condition

$$\frac{\lambda^*}{G\delta c''}=\frac{\left[(l-1)\frac{1}{g}\frac{dg}{dl}+1\right]}{\left[\frac{1}{g}\frac{dg}{dl}+\frac{1}{i_l''-cT_\infty}\frac{di_l''}{dl}\right]}. \tag{6}$$

The magnitude of λ** is found from the condition for the safe operation of the cooling system when the temperature of the porous material on the outer surface does not exceed the maximum permissible magnitude T**:

$$\frac{\lambda^{**}}{G\delta c''}=\frac{1-l}{\ln S}, \tag{7}$$

where

$$S=\frac{i(P_1,T^{**})-cT_\infty}{i_l''-cT_\infty}. \tag{8}$$

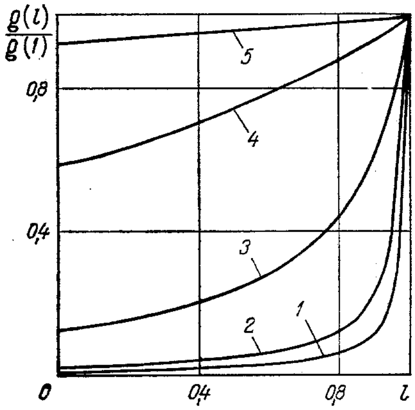

Fig. 2. Dependence of relative flow rate of coolant on coordinate of phase-transition surface: 1) $(\alpha_0/\alpha)\gamma$ = 0; 2) 1; 3) 10; 4) 100; 5) 1000.

After analyzing the conditions under which expression (6) is obtained in [2], it can be concluded that they are also applicable for the two-phase cooling of a two-layer porous plate. We shall restrict ourselves here also to the viscous flow conditions only. Water ($i'' \approx$ const) is used as the coolant. In this case the stability condition (6) for the two-layer plate is

$$\frac{\lambda^*}{G\delta c''}=\frac{1+\dfrac{\alpha_0}{\alpha}\gamma}{\dfrac{v''}{v'}-1}. \tag{9}$$

It follows from this that for a two-layer plate, other conditions being equal, the permissible magnitude of the coefficient of thermal conductivity of the porous material of the outer layer can be increased compared with the corresponding characteristic of a one-layer plate $[(\alpha_0/\alpha)\gamma = 0]$ proportional to the parameter $(\alpha_0/\alpha)\gamma$.

It is essential to note that the value of using a two-layer porous wall is not restricted to this result. Another, no less important, property consists in the fact that the range of variation in the external heat flow can be extended considerably in the stable and safe operating mode of the cooling system without adjusting the pressure drop at the wall by the appropriate selection of the structural characteristics of the two-layer wall.

The most effective means of investigating a two-phase porous cooling system with aperiodic instability is the thermal characteristic which establishes a relationship between the coordinate of the equilibrium phase-transition surface and the density of the external heat flow [1]. The equation for the thermal characteristic is written in its normalized form as

$$\frac{q(l)}{q(1)}=\frac{g(l)}{g(1)}\exp\left[B(1)\frac{g(l)}{g(1)}(1-l)\right]. \tag{10}$$

Here $B(1) = G(1)\delta c''/\lambda$. The system is stable if at the operating point $dq/dl < 0$.

Figure 3 shows the thermal characteristics of the systems for which the $g(l)/g(1)$ dependences are depicted in Fig. 2. For the calculation it is assumed that $\mathrm{Re} \to 0$; P_1 = 1 bar; T_∞ = 20°C; $q(1) = 10^6$ W/m²; δ = 5 mm; λ = 10 W/(m·deg); and the coolant is water.

The increase in the resistance of the inner layer obviously raises the stability of the system. In this context the possibility of determining the minimum parameter $[(\alpha_0/\alpha)\gamma]^\circ$ for a fixed λ, above which the system becomes absolutely stable, i.e., stable for any value of the vaporization region coordinate, is of special interest. The value of $[(\alpha_0/\alpha)\gamma]^*$ corresponding to the state at the boundary of stability accompanying a phase transition in the region with a coordinate l, can be found by converting (9) into the form

$$\left[1+\left(\frac{\alpha_0}{\alpha}\gamma\right)^*\right]^2 B(1)-\left[1+\left(\frac{\alpha_0}{\alpha}\gamma\right)^*\right]\left(\frac{v''}{v'}-1\right)-\left(\frac{v''}{v'}-1\right)^2(1-l)=0 \tag{11}$$

and solving the following equation:

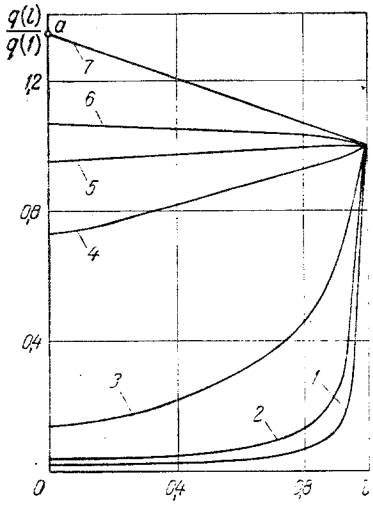

Fig. 3

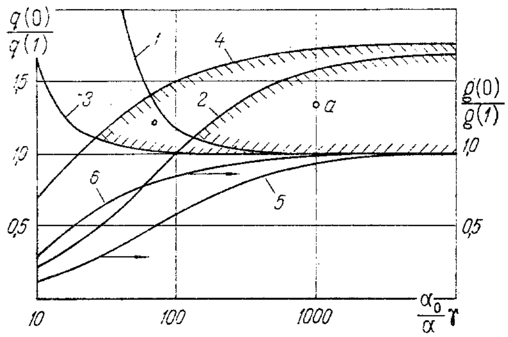

Fig. 4

Fig. 3. Thermal characteristics of two-phase porous cooling system: 1) $(\alpha_0/\alpha)\gamma$ = 0; 2) 1; 3) 10; 4) 100; 5) 177; 6) 250; 7) 1000.

Fig. 4. Dependence of range of variation in density of external heat flow in optimum mode: P_1 = 1 bar: 1) $q^*(0)/q(1)$; 2) $q^{**}(0)/q(1)$; 5) $g(0)/g(1)$; P_1 = 10 bar; 3) $q^*(0)/q(1)$; 4) $q^{**}(0)/q(1)$; 6) $g(0)/g(1)$.

$$\left(\frac{\alpha_0}{\alpha}\gamma\right)^* = \frac{\left(\frac{v''}{v'}-1\right)}{2B(1)}[1+\sqrt{1+4(1-l)B(1)}]-1. \quad (12)$$

From this it follows that when the vaporization region is deepened (l is reduced), $[(\alpha_0/\alpha)\gamma]^*$ increases and acquires a maximum value for a phase transition near the boundary line dividing the layers (l = 0). Thus, the system is converted gradually to stable operation as the $(\alpha_0/\alpha)\gamma$ parameter is raised: first the vaporization of the coolant on the outer surface becomes stable (for the data in Fig. 3 this corresponds to dependence 5 — $(\alpha_0/\alpha)\gamma = [(\alpha_0/\alpha)\gamma]^*|_{l=1}$ = 177) and then with a deepening of the vaporization region.

The cooling system is absolutely stable if

$$\frac{\alpha_0}{\alpha}\gamma \geqslant \left(\frac{\alpha_0}{\alpha}\gamma\right)^0 = \left(\frac{\alpha_0}{\alpha}\gamma\right)^*\Big|_{l=0} = \frac{\left(\frac{v''}{v'}-1\right)}{2B(1)}[1+\sqrt{1+4B(1)}]-1. \quad (13)$$

For the data in Fig. 3, $[(\alpha_0/\alpha)\gamma]^0$ = 231.

In the absolutely stable system the largest value of the external heat flow density corresponds to the vaporization of the coolant near the layer interface — optimum mode. Bearing this in mind, the relationship between the parameters of the inner layer and the range of permissible fluctuations in the heat flow is established. If for optimum mode operation (l = 0) the temperature of the vapor emerging is equal to the limiting T**, then this case corresponds to the maximum permissible magnitude $q^{**}(0)$ of the external heat flow. By writing expression (10), taking into account (7), for this modification we obtain

$$\frac{q^{**}(0)}{q(1)} = \frac{g(0)}{g(1)}S = \frac{1+\frac{\alpha_0}{\alpha}\gamma}{\frac{v''}{v'}+\frac{\alpha_0}{\alpha}\gamma}S. \quad (14)$$

At the same time, the process of cooling accompanying vaporization near the layer interface should be stable. For the state at the boundary of stability, taking into account (9), we have

$$\frac{q^*(0)}{q(1)} = \frac{g(0)}{g(1)} \exp\left[\frac{\frac{v''}{v'} - 1}{1 + \frac{\alpha_0}{\alpha}\gamma}\right]. \tag{15}$$

Relations (14)-(15) govern the range of variation in the external heat flow when the system is operating in the optimum mode. In Fig. 4 this range is denoted by cross-hatching. The calculations are made using the physical properties of water and water vapor for two pressures in the system: 1 and 10 bar. The limiting temperature of the material is assumed to be equal to $T^{**} = 1000°C$. The corresponding values of $g(0)/g(1)$ are also presented here.

By adjusting (14) and (15) the minimum value of $(\alpha_0/\alpha)\gamma$ parameter above which optimum-mode operation of the cooling system is in general possible can be found: $1 + (\alpha_0/\alpha)\gamma > [(v''/v') - 1]/\ln S$. In Fig. 4 this is the abscissa of the intersection points of lines 1, 2 and 3, 4. At atmospheric pressure in the system $(\alpha_0/\alpha)\gamma > 130$. An increase in the relative resistance of the inner layer from this minimum value gives rise to a broadening of the range of variation in the external heat flow when the system is operating in the optimum mode. At the same time, this range is restricted by the magnitude of $q^{**}(0)/q(1) \leq S$, which can be achieved within the limits as $(\alpha_0/\alpha)\gamma \to \infty$, when the coolant flow rate is hardly reduced as the region of vaporization is displaced from the outer surface to the interface of the layers: $[g(0)/g(1)] \to 1$. An increase in the pressure in the system also broadens the range of variation in the external thermal load.

For a certain fixed value of the $(\alpha_0/\alpha)\gamma$ parameter this or that magnitude of the density of the external heat flow from this range for operation in the optimum mode is ensured by selecting the porous material of the outer layer, the effective coefficient of thermal conductivity of which should lie in the interval

$$\frac{q(1)\,\delta c''\left(1 + \frac{\alpha_0}{\alpha}\gamma\right)}{(i'' - cT_\infty)\left(\frac{v''}{v'} + \frac{\alpha_0}{\alpha}\gamma\right)\ln S} = \lambda^{**} < \lambda < \lambda^* = \frac{q(1)\,\delta c''\left(1 + \frac{\alpha_0}{\alpha}\gamma\right)^2}{(i'' - cT_\infty)\left(\frac{v''}{v'} + \frac{\alpha_0}{\alpha}\gamma\right)\left(\frac{v''}{v'} - 1\right)}. \tag{16}$$

For example, when $(\alpha_0/\alpha)\gamma = 1000$ we obtain $\lambda^{**} = 6.85$ and $\lambda^* = 52.5$ W/(m•deg), if it is assumed for the calculation that $Re \to 0$, $P_1 = 1$ bar, $T_\infty = 20°C$, $q(1) = 10^6$ W/m^2, $\delta = 5$ mm, and the coolant is water. Then $q(0)/q(1) = 1.34$ — the point *a* in Figs. 3 and 4 — corresponds to the value $\lambda = 10$ W/(m•deg).

Thus, it is possible by using a two-layer wall in a two-phase porous cooling system to ensure a stable and safe mode of operation when a porous material with a real magnitude of thermal-conductivity coefficient is used for the outer layer; it is also possible to extend substantially the range of variation in the external heat flow without adjusting the pressure drop at the plate.

A similar conclusion is also valid for a random and not only viscous mode of coolant flow in a porous two-layer wall.

NOTATION

G, g, dimensional and dimensionless specific coolant flow rates; L, l, dimensional and dimensionless coordinates of vaporization surface; Γ, γ, absolute and relative thicknesses of inner layer; δ, thickness of outer layer; q, density of external heat flow; P, pressure; T, temperature; μ, ν, dynamic and kinematic viscosities; v, specific volume; i, enthalpy; c, heat capacity; λ, coefficient of thermal conductivity of porous material of outer layer; α_0, β_0, α, β, viscous and inertial coefficients of resistance of porous materials of inner and outer layers; Re, Reynolds number of coolant flow in porous material. Indices: ', ", physical properties of fluid and vapor in saturated state for ambient pressure P_1; *, parameters at boundary of stability; **, parameters at boundary of safety; l, parameters in region of phase transition.

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ANALOGY BETWEEN A THERMISTOR AND A PLASMA GENERATOR ARC

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An analysis of the experimental data confirms the existence of an analogy between a semiconductor thermistor and a dc electric arc burning under linear plasma generator conditions.

In order to develop engineering methods of designing electrical circuits containing nonlinear elements it is important to draw an analogy between metallic and semiconducting thermistors and a "gaseous" temperature-dependent resistance — a dc electric arc. A preliminary qualitative analysis of the experimental data shows that, despite the difference in physical properties and internal processes, there are a number of common features, based on the sensitivity of the resistance to temperature changes.

There follows a qualitative and quantitative evaluation of the correspondence between the two elements. For comparison, all the integral thermal and electrical characteristics of the thermistor and the arc are treated in accordance with the procedures developed in the theory of solid temperature-dependent resistances. Parallels are drawn in the following order: static volt—ampere characteristics — region of stable operation — circuit power supply regime — sensitivity of resistance to changes in gas flow rate — temperature — power output.

For purposes of comparison we will use an electric arc vortex-stabilized in a cylindrical channel. The arc chamber [1], open at one end (Fig. 1), is formed by two tubular coaxial electrodes ($d_a = 8 \cdot 10^{-3}$ m, $l_c = 0.1$ m, $d_c = 5 \cdot 10^{-3}$ m, and $l_a = 0.1$ m), electrically insulated from each other by a ventilated air gap ($\delta = 1.5 \cdot 10^{-3}$ m). The gas is supplied tangentially to the working space through two diametrically opposed orifices ($d = 1 \cdot 10^{-3}$ m).

The thermistor and the electric arc are both characterized by a relationship between thermal and electrical phenomena expressed by the volt—ampere characteristic. Both for a type KMT-1 thermistor ($R_{20} = 102.8$ kΩ, $B = 4225$°K, and $T_m = 20$-80°C) in an air medium ($w = 0$) [2] and an electric arc [1] the dependence of the current on the voltage drop is nonlinear and the differential resistance is negative $(dU/dI) < 0$.

Graphical differentiation of the volt—ampere characteristics established that

$$\left(\frac{dU}{dI}\right)_{\text{arc}} = -(0.25-0.67) \leqslant \left(\frac{dU}{dI}\right)_{t} = -(1-3000).$$

Clearly, the slope of the volt—ampere characteristic for the thermistor may vary more broadly than for the electric arc. The specific properties and characteristics of the resistors compared become especially apparent when the region of stable operating regimes is considered. Under steady-state conditions, the heat transfer between the thermistor and an ambient medium of normal density obeys the law of convective energy transport

$$I_t^2 R_t = K(T - T_m). \tag{1}$$

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