ABSTRACTS OF ARTICLES DEPOSITED AT VINITI*

EFFECTS OF A MASK ON THE TEMPERATURES OF SUBSTRATES AND FILMS

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UDC 536.241

A film is deposited on a substrate under vacuum via a mask [1], whose temperature is raised on account of the thermal radiation from the crucible, while it is cooled by emission. The mask thus affects the temperature distribution in the substrate and film.

The mask is made from ceramic, while the holder is made of bronze. The temperature of the mask may be derived from formula (Γ) of [1]. The holder is not heated in the above way, since it is covered by the mask and the substrate. Heat transfer occurs between the mask, substrate, and mount. The temperature of the substrate is given by

$$\Theta = \frac{c_s m_s \Theta_i + c_h m_h t_s + \lambda S_2 \tau \frac{\Theta_m - t_s}{\delta}}{c_s m_s + c_h m_h},$$

where Θ is in °C. Also, the temperatures of the film and substrate are affected by the side temperature of the mask, particularly the part directly adjoining the film. This contribution can be determined from the heat-balance equation. The final temperature of substrate and film is given by

$$\Theta_{fin} = \frac{S_l \lambda \Theta_m \tau + c_s m_s \Theta \delta_1 + c_f m_f \Theta \delta_1}{S_l \lambda \tau + c_s m_s \delta_1 + c_f m_f \delta_1}.$$

If the substrate is not metallic but an insulator (in the present case glass), then

$$\Theta_{fin} = \frac{S_l \lambda \Theta_m \tau \delta_s + \lambda_s S_s \Theta \tau \delta_1 + c_f m_f \Theta \delta_s \delta_1}{S_l \lambda \tau \delta_s + \lambda_s S_s \tau \delta_1 + c_f m_f \delta_s \delta_1}.$$

NOTATION

$c_s m_s$, $c_h m_h$, $c_f m_f$, thermal capacities of substrate, holder, and film; t_s, substrate temperature; λ, λ_s, thermal conductivities of mask and substrate; τ, radiation emission time; Θ_i, substrate temperature (neglecting effect of mask); Θ_m, temperature of mask at end of deposition; δ, δ_s, thicknesses of mask and substrate; δ_1, thickness of mask relative to film; S_s, S_2, S_l, areas of substrate, mask, and lateral surface of mask.

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STEADY-STATE RADIATION FIELD: REPRESENTATION BY PROJECTION

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Many physical experiments involve measuring the spatial characteristics of radiation fields; shadow images are frequently employed for the purpose, but they are not always convenient and sometimes require additional interpretation and processing, particularly if quantitative data are required.

We have developed a method and equipment for representing data on the spatial characteristics of radiation fields as three-dimensional (projection) images on the plane of a display unit. The method is based on algorithms that relate the spatial parameters of an object to the image in projection. For instance, the following algorithm can be used for the radiation flux from a point source:

$$\begin{aligned} X = \frac{F}{f} \{ & x' \cos\varphi_2 \cos\varphi_3 - y' \cos\varphi_3 \sin\varphi_3 + z' \sin\varphi_2 + [x' (\sin\varphi_1 \sin\varphi_3 \\ & - \cos\varphi_1 \sin\varphi_2 \cos\varphi_3) + y' (\cos\varphi_1 \sin\varphi_2 \sin\varphi_3 + \sin\varphi_1 \cos\varphi_3) \\ & + z' \cos\varphi_1 \cos\varphi_2] \operatorname{tg} \gamma_1 \}, \\ Y = \frac{F}{f} \{ & x' (\sin\varphi_1 \sin\varphi_2 \cos\varphi_3 + \cos\varphi_1 \sin\varphi_3) - y' (\sin\varphi_1 \sin\varphi_2 \sin\varphi_3 \\ & - \cos\varphi_1 \cos\varphi_3) - z' \sin\varphi_1 \cos\varphi_2 + [x' (\sin\varphi_1 \sin\varphi_3 - \cos\varphi_1 \sin\varphi_2 \cos\varphi_2) \\ & + y' (\cos\varphi_1 \sin\varphi_2 \sin\varphi_3 + \cos\varphi_1 \sin\varphi_1) + z' \cos\varphi_1 \cos\varphi_2] \operatorname{tg} \gamma_2 \}, \end{aligned} \tag{1}$$

where F is focal distance; φ_1, φ_2, φ_3 are parameters that define the setting of the image; γ_1 and γ_2 are parameters that define the perspective; Y and X are the vertical and horizontal components of the projection image; and f is the distance from the object to the recording plane.

No matter what the orientation of the object in real space, this algorithm allows one to specify readily any convenient point of view by varying φ_1, φ_2, and φ_3. Analogous algorithms can be derived for various parameters under detailed experimental conditions.

This method has been realized in equipment for examining the spatial distribution of the x rays from a betatron; the equipment consists of a scanning system, a data-processing unit, and a display. The data are captured by scintillation counters arranged in lines in a plane of size 30 × 40 cm. The data-processing unit realizes an algorithm of the type of (1). The resulting images are extremely clear and differ from shadow images in allowing one to directly determine quantities such as the radiation intensity at any point in the scan.

The method can also be used in examining temperature distributions, in flaw detection, and so on.

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RADIANT HEAT TRANSFER IN POWDERED MATERIALS AT HIGH TEMPERATURES

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We have investigated the thermal conductivity of mineral powders with a dispersion from 0.1 to more than 15 mm used in metallurgy, construction, and the refractory industry (Table 1). The thermal conductivity was determined in the 100-1100°C range by measurements of steady-state radial heat flow. The experimental results show a rapid increase of the effective thermal conductivity with temperature.

TABLE 1. Characteristics of Powders Investigated

Material	Fractional composition, %					Bulk density, kg/m^3	$\lambda_{eff}=\lambda_0+\beta T^3$	
	>15 mm	15—8 mm	8—3 mm	3—0,5 mm	<0,5 mm		λ_0, W/m·°K	$\beta\cdot 10^{10}$ W/m·°K^4
Concrete	—	—	100	—	—	620	0,193	3,87
The same	—	100	—	—	—	580	0,203	5,83
The same	100	—	—	—	—	520	0,253	8,50
Schungizite gravel	5	80	15	—	—	510	0,211	5,53
Chamotte lightweight	50	40	10	—	—	330	0,227	6,02
Vermiculite;	—	15	30	40	15	290	0,114	3,29
Pearlite	—	—	—	—	100	60	0,080	0,93
Pearlite—graphite mixture	—	—	—	—	100	220	0,060	1,19
Ash—graphite mixture	—	—	—	—	100	760	0,142	1,04

Assuming the additivity of heat fluxes in heat transfer in disperse systems, the effective thermal conductivity is

$$\lambda_{eff} = \lambda_s + \lambda_g + \lambda_{con} + \lambda_{rad}. \tag{1}$$

Here the terms take account, respectively, of the amount of heat transferred through the solid phase, through the gas in the pores, by convection, and by radiation. A detailed consideration shows that the temperature dependence of the first three terms is hardly stronger than linear.

Radiative heat transfer makes the largest contribution to the temperature dependence. Since $\lambda_{rad} \sim T^3$, $\lambda_{eff} \approx \lambda_0 + \beta T^3$, which is confirmed by the linearization of the relation obtained in appropriate coordinates. The parameters λ_0 and β are listed in Table 1.

In the 200-1100°C range the fraction of the heat transfer contributed by radiation increases approximately from 20 to 90% for coarse-grained ($\bar{d} \sim 10$ mm) powders and from 10 to 70% for fine-grained ($\bar{d} \sim 0.1$ mm) powders. At lower temperatures the thermal conductivity of coarse-grained powders is practically independent of the size of the particles and the kind of material.

For concrete powders of various granular structure there is semiquantitative agreement with the Chudnovskii—Kaganov formula

$$\beta = 2\varepsilon^2\sigma h, \tag{2}$$

where ε is the emissivity of the surface of a grain, σ is the Stefan—Boltzmann constant, and h is the diameter of the pores.

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Ukrainian Scientific-Research Institute of Refractories.
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THEORY OF GALVANOTHERMOMAGNETIC REFRIGERATORS OF LONGITUDINAL TYPE

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The existing theory of such devices (thermobatteries and refrigeration units) is most fully presented in [1], where all the major steady-state working characteristics are presented. A major disadvantage of this theory is that it neglects (without justification) the effect of magnetic-field reversal on the thermo-emf.

The field-reversal effect means that $\alpha_i(H) \neq \alpha_i(-H)$, and this alters the characteristics of such a refrigeration unit. The expression for the temperature difference shows that some previously unknown types of cooling unit can thus exist.

1. A reversal cooler. This unit utilizes simply the change in sign of the thermo-emf on reversing the magnetic field, and this change can be attained if one half of the specimen is placed in a magnetic field of one polarity and the other half in the field of the opposite polarity, i.e., $a_1 = -a_2 = a$, $b_1 = b_2 = b$, $\varkappa_1 = \varkappa_2 = \varkappa$, $\alpha_1(-H) = \alpha_2(-H)$, $\alpha_2(-H) = \alpha_1(H)$.

Experiment shows that such a device can exist; the material for the branches was a bismuth single crystal having the following orientation: the axis of the specimen deviated by 10° from the binary axis and by 80° from the bisector axis in the binary-bisector plane. The experiments were performed under vacuum with H = 10 T, T = 77°K; under these conditions, $\rho = 10^{-5}\ \Omega\cdot m$, $\varkappa = 10$ W/m·°K, $|\alpha_1(H)-\alpha_2(H)| = 400\ \mu V/°K$. A parallelepiped of this orientation was prepared from the bismuth single crystal, and this was then cut into halves, with one half turned through 180° relative to the other, the ends being then joined together with Wood's metal.

The result was a temperature difference of about 1°K, which agrees well with that predicted by the theory. This cooler has certain advantages: first, both of the branches are made of the same material, which is important; secondly, it works at low temperatures, because the reversal effect is most prominent precisely in that temperature range [2]; and, thirdly, the quality factor is determined by the strength of the magnetic field.

2. The branches in the cooler have identical reversal parameters: $a_1 = a_2 = a$; then the temperature difference may be increased, since this parameter is not zero.

3. If the current is sufficiently large and in the range

$$\frac{\varkappa_1 l^{-1}}{\alpha_1(-H)-\alpha_1(H)} \ll J < \frac{\alpha_1(H)-\alpha_2(H)}{\rho_1 l} T_l$$

the temperature difference is linearly dependent on the current.

NOTATION

$a_i = J([\alpha_i(-H)-\alpha_i(H)]/\varkappa_i)$, reversal parameter; $b_i = \rho_i J^2/\varkappa_i$, parameter characterizing the Joule heat; i = 1 or 2, numbers of the cooler branches; J, current density; H, magnetic-field strength; l, branch length; T_l, thermostat temperature; $\alpha_i(H)$, ρ_i, $\varkappa_i$, thermo-emf, specific resistance, and thermal conductivity of branch i.

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THE OPTIMUM FIN PARAMETERS FOR A SURFACE COOLED BY A BOILING LIQUID

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A cylindrical shell has vertical rectangular fins, and the optimum parameters are derived from the condition for providing maximum heat flux through the load-bearing surface of the ribbed wall under conditions of bubble boiling. The optimum values are determined for the rib thickness and rib separation on the basis of heat-transfer and critical-flux relationships for the boiling of Freon-11 in the channels between the ribs.

The following expression is obtained for the critical heat flux density through the carrying surface on the basis of heat balance:

$$q_{cr} = \frac{2lE + d_0}{\delta_r + d_0} \cdot q_{0cr}. \quad (1)$$

The maximum desirable value for the height l of a rib is derived by examining the performance factor for a rectangular rib cooled by a boiling liquid; then $l = 1.2\sqrt{\lambda\delta_r/2\alpha_0}$, and (1) becomes

$$q_{cr} = \frac{1.2\sqrt{\lambda\delta_r/2\alpha_0} + d_0}{\delta_r + d_0} q_{0cr}. \quad (2)$$

The optimum value of δ_r for a given d_0 is defined by $\partial q_{cr}/\partial \delta_r = 0$; this gives

$$\delta_{r\,opt} = d_0\left[1 - \frac{2}{c^2}\left(\sqrt{d_0^2 + c^2 d_0} - d_0\right)\right], \quad c = 1{,}2\sqrt{\lambda/2\alpha_0}. \tag{3}$$

The optimum value of d_0 is found by examining (2) and (3) together; the heat-transfer coefficient α_0 and the critical heat flux q_{0cr} at the base of a rib have been determined from studies on these quantities using models for the channels between the ribs with uniform heat production in the walls. The measurements on these two quantities have been used with (2) and (3) to derive graphs for the critical heat-flux density for a load-bearing copper wall for various pressures and for various heights of the channels between ribs. It is found that the optimum value for the gap d_{0opt} is in the range from 0.9 to 1.4 mm for pressures of F-11 between 1.0 and 4.6 bar, while the rib thickness $\delta_{r\ opt}$ lies between 0.7 and 1.0 mm. The maximum heat-flux density at the load-bearing surface then varies in the range from $6.0 \cdot 10^6$ to $8.0 \cdot 10^6$ W/m^2.

Specimens of a klystron collector were made and tested in order to check the relationships; the critical heat-flux density at the surface of the collector was taken as the value corresponding to the knee on the $q = f(\Delta t)$ curve, which corresponds to the onset of film boiling at the base of the ribs. The measurements agreed satisfactorily with the calculation. The maximum heat-transfer capacity of a cylindrical shell with optimum ribs exceeds by more than a factor of 4 the value of q_{0cr} for a smooth isothermal surface.

NOTATION

q_{cr}, q_{0cr}, heat-flux densities for the load-bearing surface of a ribbed wall and for the surface of a rib at the base, respectively; α_0, heat-transfer coefficient at the foot of a rib; l, rib height; d_0, distance between ribs; δ_r, rib thickness; λ, thermal conductivity; E, rib performance factor.

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EFFECTS OF A THERMALLY INSULATED SECTION ON FLOW IN A CHANNEL WITH COLD WALLS

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Measurements are reported for the flow structure in a cylindrical pipe with cold walls ($T_W/T^* = 0.1$), in which there is a ceramic section of length $l = D$; the gas (air) was heated in a plasma source to $T^* = 3100$–3600°K, and the channel with D = 0.04 m was a direct continuation of the outer electrode of the plasma source, as it consisted of two water-cooled metal tubes of length L = 8D each. The stagnation temperature T* and the flow speed v were determined with a calorimetric probe having d = 0.003 m [1,2] for various points in the tube.

The distribution of T* and v at the outlet from the plasma source differ from those characteristic of developed turbulent flow in a pipe (n = 1/7) in that the boundary layer was cooler, the core of the flow was more highly elongated, and the maximum value of T* did not occur at the axis of the flow. In the absence of the ceramic section, the distortion of the T* and v distributions terminated at the point x/D = 8, where the distributions for T* and v corresponded to the result for n = 1/7.

The effect on the distribution of T* near the wall past the ceramic section was larger when the section was placed at the start of the pipe, i.e., in the flow-stabilization region, and the maximum T* occurred at the axis. The distortion of the T* distribution produced by this section in the stabilization path persisted downstream for at least 3-4 times the diameter. Therefore, a short thermally insulated section can reduce the thermal-stabilization length substantially. The T* distributions past a ceramic section were unaffected within the error of measurement by the section if this was placed in the middle, i.e., where the flow had stabilized, so the distribution corresponded to n = 1/7.

The distributions of v after the ceramic section in both cases deviated from the n = 1/7 form, although the deviation in v near the wall past the section was reduced somewhat when the section was placed at the start of a pipe. The reason is the increased roughness of the cold surface, since particles are deposited past the ceramic section that arise from erosion of the ceramic, no matter whether this is placed in the stabilization section or in the stabilized part, whereas the erosion products from the electrodes in the plasma source are deposited on the surface before the point $x/D = 8$.

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FREE CONVECTION ABOVE A HORIZONTAL PLATE

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UDC 536.253

The problem of free convection of a one-component gas above an isothermal horizontal plate is solved in the boundary-layer approximation by the method of integral relations. Restricting ourselves to third-degree polynomials, and taking account of the boundary conditions, we obtain for the flow velocity of the gas v_x and the enthalpy h

$$v_x = U_1(\xi)\,\eta\,(1-\eta)^2,\quad \vartheta = \vartheta_w(1-\eta_1)^3,\quad \vartheta = \frac{h}{h_0} - 1;$$
$$\eta = \frac{\eta'}{\delta(\xi)},\quad \eta_1 = \frac{\eta'}{\Delta(\xi)}.$$

The method of integral relations enables one to obtain analytical expressions for the unknowns $U_1(\xi)$ and the thickness of the dynamic boundary later $\delta(\xi)$ in the form

$$U_1(\xi) = A_0(K_w, \mathrm{Pr}, \mathrm{Gr})\,\xi^{\frac{1}{5}},\quad \delta(\xi) = A_1(K_w, \mathrm{Pr}, \mathrm{Gr})\,\xi^{\frac{2}{5}},$$

and the equation for the local Nusselt number

$$\mathrm{Nu}_x = \left(\frac{x}{L}\right)^{\frac{3}{5}} \mathrm{Gr}^{\frac{1}{5}} \left\{ 3\frac{\rho_w}{\rho_\infty}\left(\frac{\mathrm{Pr}}{210K_w}\cdot\frac{\langle \nu \rangle}{\nu_\infty}\right)^{\frac{2}{5}} \left[\frac{\frac{3}{20}(1-\langle\beta\rangle\Delta T)}{\left(\frac{\mathrm{Pr}}{210}+\frac{4}{525}\right)}\right]^{\frac{1}{5}} \right\}, \tag{1}$$

where

$$\mathrm{Gr} = \frac{g\langle\beta\rangle\Delta T L^3}{\langle\nu\rangle^2}\quad K_w = \frac{\rho_w\mu_w}{\rho_\infty\mu_\infty},\quad \mathrm{Pr} = \frac{C_p\mu}{\lambda},\quad \Delta T = T_w - T_\infty,$$

and the symbol <> denotes the average value of the quantity enclosed.

Equation (1) enables one to take account of the average change in density and the temperature dependence of the thermophysical parameters of the gas.

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A PROBLEM IN HALF-SPACE COUPLING

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An analysis has already been presented [1,2] of ordinary (parabolic) temperature distributions in two adjoining half-spaces; since heat transfer takes a wave form in rapid nonstationary processes, the true temperature distributions will be described by functions that represent solutions to a system on generalized (hyperbolic) thermal-conduction equations [3] if one assumes that the relaxation times for the thermal stresses are independent of direction:

$$b_i^2 \frac{\partial^2 u_i}{\partial t^2} + c_i^2 \frac{\partial u_i}{\partial t} - a_i^2 \Delta u_i = f_i(x, t) \tag{1}$$

subject to the initial conditions

$$u_i|_{t=0} = \varphi_i(x); \left.\frac{\partial u_i}{\partial t}\right|_{t=0} = \psi_i(x) \tag{2}$$

and the following conditions at the interface:

$$u_1|_{x_3=a} - u_2|_{x_3=a} = \alpha(x', t); \ k_1 \left.\frac{\partial u_1}{\partial x_3}\right|_{x_3=a} - k_2 \left.\frac{\partial u_2}{\partial x_3}\right|_{x_3=a} = \beta(x', t). \tag{3}$$

Here $u_i = u_i(x_1, x_2, x_3, t) \equiv u_i(x', x_3, t)$; Δ is a three-dimensional Laplace operator; b_i, c_i, a_i, k_i are real non-negative numbers that satisfy $a_i^2 = k_i$, $c_i^2 = \bar{c}_i\gamma_i$; $b_i^2 = k_i/W_{r_i}^2$ and has the following physical meaning: k_i is the thermal conductivity, $\bar{c}_i$ is specific heat, γ_i is density, and $W_{r_i} = \sqrt{k_i/\bar{c}_i\gamma_i\tau_{r_i}}$ is the rate of propagation of heat, where τ_{r_i} is the relaxation time for the thermal stress (i = 1, 2). If the spaces are bounded in x' by constant $l' = (l_1, l_2)$, the boundary conditions further become

$$u_i|_{x'=0} = 0; \ u_i|_{x'=l'} = 0 \ (i = 1, 2). \tag{4}$$

Integral Fourier transformation with respect to x' and Laplace transformation with respect to t provide an exact solution to (1)-(3); an approximate analytical solution to (1)-(4) may be constructed by the straight-line method applied to x' and Laplace transformation applied to t. In particular, a parabolic temperature distribution is obtained in one of the regions or in both simultaneously if $b_1 \to 0$ or $b_2 \to 0$, or $b_1 \to 0$ and $b_2 \to 0$, while pure wave distributions are obtained if $c_1 \to 0$ or $c_2 \to 0$, or $c_1 \to 0$ and $c_2 \to 0$. Numerical calculations have been performed on parabolic temperature distributions for particular bodies.

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HEAT TRANSFER IN THE INITIAL THERMAL SECTION OF A RECTANGULAR CHANNEL WITH BOUNDARY CONDITIONS OF THE SECOND KIND

V. L. Rvachev, A. P. Slesarenko, and V. I. Popivshchii

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A laminar and hydrodynamically stabilized flow of a liquid is considered with piecewise-constant heat-flux densities at the faces; the initial boundary-value problem takes the form

$$W \frac{\partial T}{\partial z} = a\left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2}\right), \tag{1}$$

$$\left.\frac{\partial T}{\partial \nu}\right|_{\Gamma_1} = -\frac{q_1}{\lambda}, \ \left.\frac{\partial T}{\partial \nu}\right|_{\Gamma_2} = -\frac{q_2}{\lambda}, \tag{2}$$

$$T|_{z=0} = T_0. \tag{3}$$

The problem of (1)-(3) may be solved by using integral Laplace transformation in conjunction with the structural method (R-function method) and Ritz's method. The structure of the solution in the image plane is defined by

$$\bar{T} = \bar{\varphi}_0 + B(\Phi, p). \tag{4}$$

Here $\overline{\varphi}_0$ is a certain function, while B is a certain operator, which is constructed in such a way that the function of (4) satisfies the boundary conditions of (2) exactly for any choice of Φ from some set. This Φ is chosen by means of Ritz's variational method. Standard theorems are applied to return to the original.

Results are presented on the Nusselt number averaged over the perimeter for the initial section; the results for the dimensionless temperature in the stabilization region are compared with the exact solution.

NOTATION

x, y, z, Cartesian coordinates; W, velocity distribution; ν, direction of the internal normal to contour Γ_1 or Γ_2; q_1, q_2, heat-flux densities at the faces; λ, thermal conductivity; a, thermal diffusivity; T_0, constant temperature of the liquid at the inlet; p, parameter of the integral Laplace transformation; $\overline{T}$, Laplace transform of temperature T.

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ENERGY DISTRIBUTION IN THE DEGREES OF FREEDOM IN A CONDENSED MEDIUM

V. A. Bubnov

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Boltzmann used a Maxwellian velocity distribution in his molecular-kinetic interpretation of the Clapeyron—Mendeleev empirical equation of state, the thermal velocities of the atoms molecules being uncorrelated in this distribution. He then assumed that the mean kinetic energy is proportional to the temperature and obtained the result that half the thermal energy resided in each degree of freedom as a consequence of the statistical independence of the components of the thermal velocity. The latter result is known in statistical mechanics as the theorem on the equipartition of energy.

Any correlation in the components of the thermal velocity of the atoms or molecules results in a distribution of the following form:

$$f = A\exp\left[-\frac{1}{2}\left(\frac{R_{11}}{R}\cdot\frac{\xi^2}{\sigma_1^2}+\frac{R_{22}}{R}\cdot\frac{\eta^2}{\sigma_2^2}+\frac{R_{33}}{R}\cdot\frac{\varsigma^2}{\sigma_3^2}+2\frac{R_{12}}{R}\cdot\frac{\xi\eta}{\sigma_1\sigma_2}+2\frac{R_{13}}{R}\cdot\frac{\xi\varsigma}{\sigma_1\sigma_3}+2\frac{R_{23}}{R}\cdot\frac{\eta\varsigma}{\sigma_2\sigma_3}\right)\right].$$

Here R is a correlation determinant, while σ_1, σ_2, and σ_3 are standard deviations, which are proportional to the kinetic energies in the corresponding degrees of freedom.

If the correlation is isotropic, this quadratic form can be reduced to the canonical form

$$f = Ae^{-\frac{1}{2\sigma^2}}(s_1\xi^2+s_2\eta^2+s_3\varsigma^2). \tag{1}$$

Here the symbols are

$$s_1 = 1-2n,\ s_2 = 1+n,\ s_3 = 1+n,$$

$$n = \frac{\beta r_1}{1+r_1} = \frac{r}{1+r},$$

$$\sigma_1 = \sqrt{\frac{R_{11}}{R}}\,\sigma,\quad \sigma_2 = \sqrt{\frac{R_{22}}{R}}\,\sigma,\quad \sigma_3 = \sqrt{\frac{R_{33}}{R}}\,\sigma.$$

Formula (1) allows us to generalize the equation of state for an ideal gas as

$$pv = \psi(r_1, \beta)RT \tag{2}$$

and to calculate the energies in the degrees of freedom:

$$E_1 = \frac{m}{2}\overline{\xi^2} = \frac{kT}{2s_1},$$

$$E_2 = \frac{m}{2}\overline{\eta^2} = \frac{kT}{2s_2},$$

$$E_3 = \frac{m}{2} \varsigma^2 = \frac{kT}{2s_3}.$$

The function ψ introduced here has the form

$$\psi = \frac{(1 + r_1)[1 + (1 - \beta) r_1] \sigma^2}{[1 - (2\beta - 1) r_1][1 + (1 + \beta) r_1]},$$

and this is indicated as being objective because (2) describes the observations on the compressibility factor for hydrogen over wide ranges in temperature and pressure.

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TEMPERATURE DISTRIBUTION IN A CHANNEL BETWEEN STRAIGHT LONGITUDINAL FINS PRODUCED BY BOILING OF A LIQUID MOVING IN A CHANNEL BELOW ITS BOILING POINT

V. K. Shcherbakov and L. V. Kovalenko

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Moving liquids below their boiling points are widely used to absorb high heat loads by boiling, which makes it necessary to calculate the temperature distribution in the walls of a heat-exchange device with complex geometry, in particular, a surface with straight longitudinal fins.

An approximate analytical method is presented for calculating the excess temperature around the perimeter of a closed channel between fins, the latter being of high thermal conductivity and the temperature being such that the liquid begins to boil.

Two characteristic cases of the propagation of bubble boiling over the perimeter occur, the exact circumstances being dependent on the heat load, the thermophysical characteristics, the dimensions of the finned wall, and the parameters and properties of the liquid:

1) boiling starts at the axis of symmetry and extends to part of the wall between the fins;

2) boiling covers the entire wall between the fins and extends up part of the faces of the fins.

It is assumed that a fin and the wall between fins may be considered as thermophysically thin, and the temperature distribution for these cases is determined in a one-dimensional formulation subject to the following assumptions:

1) the heat input from the wall side is constant;

2) the maximum density of the heat flux absorbed by the channel is less than the critical value for the working conditions;

3) the heat transfer to the contacting wall is negligible at the end of a fin;

4) the temperature gradient along the y axis in the wall between the fins has no effect on the heat flow along the x axis;

5) the heat-transfer rate on the convective parts of the perimeter of the channel is constant and is independent of the vapor formed in the boiling parts; and

6) the heat-transfer factor is proportional to the square of the temperature difference at any point on the surface in the bubble-boiling region.

The equations of thermal conduction for a planar wall and straight rib are solved together to define the temperature distribution and the distribution of the heat-flux density over the perimeter, as well as the boundaries to the bubble boiling.

The temperature distribution and the distribution of the heat flux removed by the liquid in turn allow one to estimate the performance of the fin system under conditions of bubble boiling, and the parameters may be varied to give the best relationship between the geometrical dimensions, and also to obtain data for calculations on the hydrodynamic characteristics of two-phase flows of variable temperature.

An example is presented of computer solution for the second case for a particular channel with specified heat input and cooling conditions.

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