STUDY OF ENTRAINMENT OF DROPS OF COOLANT BY WET VAPOR IN A HEAT-PIPE VAPORIZER

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Expressions are obtained to describe the performance of a heat pipe in a regime of wet vapor flow. The analytical results are experimentally confirmed.

A heat pipe, being a complex evaporative–condensative heat-transfer system, is characterized by high thermodynamic efficiency within a relatively narrow range of heat flux and temperature. This is because of several limitations on heat transfer connected with the hydrodynamics of the condensate and vapor flows, the evaporation kinetics, etc. [1]. Theoretical models which allow for these limitations make it possible to predict the performance of heat pipes with a certain error [2]. Nevertheless, some experimental studies [3, 4] do not agree satisfactorily with the theoretical models.

Present heat-pipe models [5] assume that the saturated vapor is dry. However, under real conditions the vapor will contain drops of liquid. There may be different reasons for the presence of these dopes: mechanical spraying as a result of vaporization [6]; interaction of the moving flow of vapor with the surface of the condensate [7]; a change in the thermodynamic parameters of the flow [8] leading to volume condensation of the vapor; organization of return of the condensate in the form of a disperse phase [6], such as in coaxial centrifugal heat pipes and evaporative thermosiphons; artificial spraying of drops into the vapor channel to organize circulation of the coolant in the vapor or liquid phase, such as in steam-life heat pipes. Thus, a more correct formulation of the problem would allow for the two-phase state of mass flow in the vapor channel.

It should be noted that two-phase vapor–drop systems have been studied in rather great detail already in the general case — without particular regard for heat pipes: the thermodynamics of such systems was examined in [10], hydrodynamics was investigated in [11], vaporization was investigated in [12], and volume condensation was studied in [13] in reference to vapor–liquid media. Attempts to use these models without substantial changes to describe hydrodynamics and heat exchange in heat pipes have been unsuccessful. A qualitative analysis of the behavior of a two-phase vapor–drop mass flow in the vapor channel of a heat pipe using the results in [10-13] showed that the picture of heat and mass transfer is complicated considerably. As a result, there is a decrease in heat-transfer capacity and an increase in losses to friction and temperature drops [14].

We analyzed heat and mass transfer in a heat pipe with the assumption that the vapor is wet and that the main mechanism of drop formation is vaporization in the heating zone. The latter assumption is explained by the fact that heat-tube development has required a substantial increase in the heat flux associated with the energy delivery. This in turn has necessitated the development of pipes which function in the boiling regime [15, 16]. Such pipes are used, e.g., for optical devices [17]. Relations were obtained for the maximum heat-transfer capacity of heat pipes with wet vapor and characteristic regimes of drop entrainment were determined. In evaluating the loss to friction, we showed that moisture does not significantly affect the performance of low-temperature heat pipes. The experimental results agree satisfactorily with the analytical model.

LIMITATION ON HEAT TRANSFER DUE TO MOISTURE CONTENT OF VAPOR

In most cases, the maximum amount of heat that can be transferred by a heat pipe with dry vapor is limited by the maximum mass flow of the liquid returned through the capillary structure or over the wall (for an evaporative thermosiphon):

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$$Q_{max} = G_{max} r^*. \quad (1)$$

This approach assumes that the mass flows of the vapor and returning liquid are equal. This assumption is most simply represented by the following expression:

$$G_v = G_{lw} = N_l' N_w N_{hp}, \quad (2)$$

where $N_l = \sigma\rho/\mu$ is a hydrodynamic parameter of the working liquid; $N_{hp} = S/l_{ef}$, a geometric parameter of the heat pipe; $N_f = 2K/r_{cap}$, a structural parameter of the capillary material of the wick.

Let us assume that, in the vapor flow of a disperse liquid phase with a flow-rate moisture content y_p, part of the liquid returned along the wick is transported without phase transformation along the vapor channel from the vaporizer to the condenser, i.e., is transported in the form of droplets, and carries almost no thermal energy. In this case, the maximum amount of heat transferred by a heat pipe with wet vapor may be represented in the form

$$Q_{max} = G_{lw}(1 - y_f) r^*. \quad (3)$$

The above expression discloses the mechanism by which the heat that can be delivered by the tube is limited. In actual structures, operating in a wet-vapor regime, G_{lw} and y_f change along the pipe. Meanwhile, the determining factor is the change in these quantities along the vaporizer. We will examine these features of transfer in a vaporizer below, assuming that the parameters of the two-phase medium are constant over the adiabatic zone.

DROP ENTRAINMENT IN VAPORIZATION ZONE

Let us examine the appearance of drops as a result of vaporization (boiling). Figure 1 shows a geometric model of the heat-pipe vaporizer.

Let rupture of the skin of vapor bubbles in the heat-pipe channel be accompanied by the spraying of drops of a total mass m_0 from a unit surface area of the heating zone. The size of these drops is described by a differential function of the drop distribution according to weight $\rho(r)$. Some of the drops will strike the wall, while other drops — of a diameter less than the free-falling diameter d_e — will be entrained by the vapor flow. The expression for the mass of these drops may be written as

$$m = m_0 \int_0^{r_e} \rho(r)\, dr = m_0 (1 - J). \quad (4)$$

The function $\rho(r)$ was experimentally studied by means of a vapor-bubble model [18]. The study showed that the spectrum of drop sizes is described by the expression

$$J = \exp\left[-\left(\frac{1}{\eta}\,!\,\frac{d_d}{\bar{d}_d}\right)^{\eta}\right]. \quad (5)$$

Since the inequality $d_d \leqslant d_e \ll \bar{d}_d$ is satisfied for the drops carried off by the vapor flow, the function J may be expanded into a series and we may limit ourselves to the first term. Then Eq. (4) takes the form

$$m = m_0 \left(\frac{d_e}{\bar{d}_d}\right)^{\eta}. \quad (6)$$

Since the function J is close to a log-normal distribution, we obtain the parameter distribution value

$$\eta = \frac{2}{\tilde{\sigma}} \approx 2, \quad (7)$$

where $\tilde{\sigma}$ is the variance of the log-normal distribution. The free-falling diameter of the drops may be found from the condition of equilibrium of the drops in the vapor flow

$$\Sigma \bar{F} = 0. \quad (8)$$

For the vaporization zone of a heat pipe located at an angle to the horizontal (Fig. 1) with type II boundary conditions in the vaporizer, drops in a laminar vapor flow will be acted upon by mainly three forces:

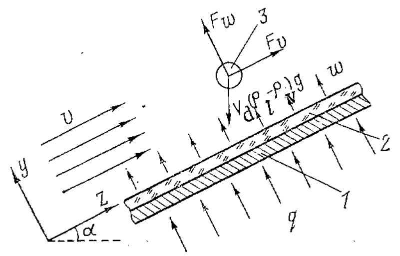

Fig. 1

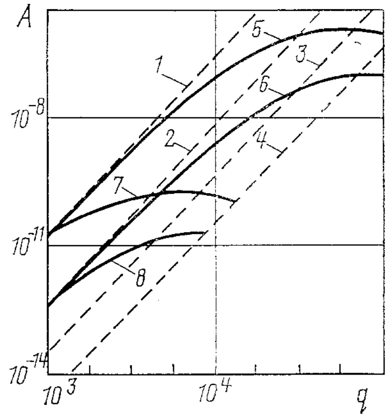

Fig. 2

Fig. 1. Element of vaporizer of heat pipe with wet vapor: 1) wall of heat pipe; 2) capillary structure; 3) drop of liquid.

Fig. 2. Dependence of numerical estimate of mass drop-entrainment parameter $A = m/c[(4z \sin\alpha)/d + \cos\alpha]$ on heat flux in vaporizer of heat pipe operated on acetone for $kn \to \infty$ (1 — T_0 = 253°K, 2 — 293°K, 3 — 333°K, 4 — 373°K); for kn = 1000 (5 — T_0 = 253°K, 6 — 293°K); for kn = 100 (7 — T_0 = 253°K, 8 — 293°K). q, W/m².

a) the resultant of the force of gravity and buoyancy:

$$F = \frac{\pi d_d^3}{6} (\rho_l - \rho_v) g, \tag{9}$$

b) the resistance of the radial vapor flow, moving with the corrected velocity $w_v = q/r^*\rho_v$:

$$F_w = \xi_w \frac{\pi d_d^2}{4} \frac{\rho_v}{2} (w_d - w_v)^2, \tag{10}$$

c) the resistance of the axial vapor flow, moving with the mean velocity $v_v = 4zq/r^* \cdot \rho_v d_c$:

$$F_v = \xi_v \frac{\pi d_d^2}{4} \frac{\rho_v}{2} (v_d - v_v)^2. \tag{11}$$

Assuming that $\xi = 24/Re$, $v_d = 0$, and $w_d = 0$, we obtain the following expression from Eqs. (8)-(11) to determine the free-falling diameter of the drops

$$d_e = \left[\frac{18\,\mu_v q}{(\rho_l - \rho_v)\rho_v g r^*} \left(\frac{4z}{d_{vc}} \sin\alpha + \cos\alpha \right) \right]^{1/2}. \tag{12}$$

The mean drop diameter $\overline{d}_d$ can be found on the basis of the customary laws of drop division [19]:

$$\frac{\overline{d}_d}{l} = C_1 \mathrm{We}^{-0.5}, \tag{13}$$

where l is a characteristic dimension for which we may take the capillary constant, proportional to the bubble radius:

$$l \sim \sqrt{\sigma/(\rho_l - \rho_v) g}. \tag{14}$$

Allowing for Eqs. (13) and (14), we may obtain an equation for the mean drop diameter

$$\overline{d}_d = C_2 \frac{\sigma}{\sqrt{\rho_v} \sqrt[4]{\sigma(\rho_l - \rho_v) g}} \frac{1}{w_v}. \tag{15}$$

An experimental investigation [18] showed that the value of m_0 depends on the mass velocity of the vapor flow $\rho_v w_v$ and is proportional to $\sqrt{(\rho_m - \rho_v)/\rho_v}$.

$$m_0 = C_3 \sqrt{(\rho_l - \rho_v)/\rho_v}\, \rho_v w_v. \tag{16}$$

Substituting Eqs. (7), (12), (15), and (16) into the equation for the mass of the entrained drops (6), we obtain

$$m = C\,18\,\mu_V \sqrt{\frac{\rho_V^3}{\sigma^3 g}} \left(\frac{4z}{d_{VC}} \sin\alpha + \cos\alpha\right) \left(\frac{q}{\rho_V r^*}\right)^4 . \tag{17}$$

Analysis of Eq. (17) shows that drop entrainment depends on the complex $18\mu_V(\rho_V^3/\sigma^3 g)^{1/2}(q/\rho_V r^*)^4$. Meanwhile, if the heat flux is determined by the external conditions of the vaporizer in actual heat pipes, then the density of the vapor is a derivative of the total heat flux Q and depends both on the boundary conditions in the condenser and the geometric parameters of the heat pipe. The density of the vapor in the heat pipe is unambiguously related to its temperature, and the dependence of the latter on the heat flux may be represented in the form $T_V = T_0 + q/kn$, where kn is a complex determined by the geometry of the heat pipe and the boundary conditions in the condenser.

Figure 2 shows the results of numerical calculation of the dependence of the parameter $A = m/c[(4z/d_{VC}) \sin\alpha + \cos\alpha]$ on q in the vaporizer of a heat pipe using acetone. Curves 1-4 were obtained for the case where the complex $kn \to \infty$. In practice, such a case (when the temperature of the vapor is nearly independent of the heat flux) is similar to the case where $q/kn \to$ const, which is typical of the operation of gas-controlled heat pipes. It may be concluded on the basis of analysis of curves 1-4 that drop entrainment depends significantly on vapor density and increases with a decrease in temperature. It was found for the case where the complex kn has a finite value (curves 5-8) that mass entrainment of drops decreases with a decrease in kn. Comparison of the amounts of drop entrainment at different heat fluxes and vapor temperatures with a finite value of kn shows that the entrainment phenomenon may have a significant effect on start-up of the heat pipe. Thus, e.g., if a high heat flux is supplied with the start-up of a heat pipe, the vaporizer may begin to dry out (burn) as a result of the large quantity of moisture ejected (sprayed) at low vapor densities. Numerical calculations show that, for a constant finite value of the complex kn, the heat-flux dependence of the mass entrainment may be of an extreme nature and may decrease at certain values of heat flux and vapor density.

BALANCE OF MASS FLOWS IN THE VAPORIZER

For the vaporizer of a heat pipe of length l_s and vapor-channel diameter d_{VC}, the balance equation for the mass flows of vapor and drops in the wick may be written in the form

$$\frac{dG_d}{dz} = \pi d_{VC} m_d,$$
$$\frac{dG_V}{dz} = \pi d_{VC} m_V, \tag{18}$$
$$\frac{dG_{lw}}{dz} = -\frac{dG_d}{dz} - \frac{dG_V}{dz},$$

where $m_V = q/r^*$ and m_d is found from Eq. (17).

Solving system (18) with the boundary conditions:

$$z = 0:\ G_V = 0,\ G_d = 0,\ G_{lw} = 0;$$
$$z = l_s:\ G_V + G_d = -G_{lw},\ G_{lw} = G_w, \tag{19}$$
$$0 < z < l_s:\ q = \text{const},$$

we obtain

$$G_V(z) = \pi d_{VC} \frac{q}{r^*} z,$$
$$G_d(z) = 18\mu_V C \left(\frac{q}{r^*\rho_V}\right)^4 \left(\frac{\rho_V^3}{g\sigma^3}\right)^{1/2} \left(\frac{2z^2}{d_{VC}} \sin\alpha + z\cos\alpha\right) \pi d_{VC}, \tag{20}$$
$$G_{lw}(z) = -G_w + \pi d_{VC} \frac{q}{r^*}(l_s - z) + 18\mu_V C \left(\frac{q}{r^*\rho_V}\right)^4 \times$$
$$\times \left(\frac{\rho_V^3}{g\sigma^3}\right)^{1/2} \left[\frac{2(l_s^2 - z^2)}{d_{VC}} \sin\alpha + (l_s - z)\cos\alpha\right] \pi d_{VC}.$$

The constant C may be determined if we know the moisture content of the vapor-drop flow at the inlet to the adiabatic zone $y_f(l_s)$:

$$C = \frac{y_f(l_s)}{1-y_f(l_s)} \left[18\,\mu_v \left(\frac{q}{r^* \rho_v}\right)^3 \left(\frac{\rho_v}{g\sigma^3}\right)^{1/2} \left(\frac{2\,l_s}{d_{vc}} \sin\alpha + \cos\alpha\right)\right]. \qquad (21)$$

With allowance for Eq. (3), system (20) makes it possible to evaluate the maximum amount of heat transferred by the heat pipe with consideration of drop entrainment according to Eq. (17).

FLOW OF THE DROP-VAPOR MEDIUM

In the model examined above for heat-pipe operation, we examined only the vaporization zone in the vapor channel. Assuming the existence of a dry, saturated, incompressible flow of vapor in a cylindrical heat pipe and constant heat fluxes along the vaporizer and condenser [3, 5], we may obtain an analytical expression for the pressure loss along the pipe. Without considering the inertial and gravitational components, the pressure drop in the vapor flow may be written as follows for the laminar regime

$$\Delta P_v = -\frac{8\,\mu_v G_v}{\pi \rho_v r_{vc}^2}\left[l_a + \frac{l_s + l_{cond}}{2}\right]. \qquad (22)$$

For the case of a vapor-drop flow, it is necessary to allow for the additional pressure drop required for movement of the drops. Only the inertial terms need be considered if the volume moisture content β of the vapor-drop flow is low. Here, recovery may be assumed absent and the pressure drop may be written

$$\Delta P_d = G_d \bar{v}_d. \qquad (23)$$

We may use the equation of motion of the drops to determine their maximum mean velocity. Assuming that the friction coefficient is determined by the relation $\xi = 24/Re_d$, we find

$$\bar{v}_d = \bar{v}_v - \frac{(\rho_l - \rho_v)\, g \bar{d}_d^2}{18\,\mu_v} \sin\alpha. \qquad (24)$$

We may use Eqs. (22)-(24) to obtain an expression for the pressure change along the vapor channel of a heat pipe with wet vapor in the form of the ratio of the pressure drop in the vapor-drop flow to the pressure losses in the motion of dry vapor

$$\frac{\Delta P_{vc}}{\Delta P_v} = 1 + \frac{\Delta P_d}{\Delta P_v} = 1 + \beta\, \frac{\rho_l\, d_{vc}^2 \bar{v}_d}{32\,\mu_v l_{ef}}. \qquad (25)$$

Figure 3 shows calculated pressure-drop-ratio relations for a horizontal heat pipe of $2 \cdot 10^{-2}$ m diameter and 0.5 m effective length operated with acetone. Values of the relations for different moisture contents are shown. Analysis of the straight lines shows that when the vapor flow contains large volumes of moisture ($\beta > 0.001$), the pressure losses in the wet vapor may be more than two orders greater than the pressure drop in the dry vapor.

EXPERIMENTAL STUDY OF HEAT-PIPE PERFORMANCE IN THE WET-VAPOR REGIME

Experimental verification of the proposed model presents certain difficulties connected with the numerous different limitations on heat transfer which exist in actual heat pipes. The most reliable method for analyzing the state of a vapor-drop flow is probing with a light beam. The state of the flow may be characterized by the mean drop diameter (averaged in some fashion), drop distribution by size, calculated concentration of drops, or drop shape [20]. For the experimental study, we assumed the absence of deformation (spherical shape) and division or coalescence of the drops. Knowing the drop concentration and size, we can determine their mass. Then, relating the mass to the total moss of the medium enclosed within a unit volume, we can find the true or frozen moisture content y.

We will represent the actual vapor-drop flow, with a drop-size distribution in the form of f(r)dr, as a nominally monodisperse system. As the equivalent parameter we will use the arithmetic mean radius of the drops r_{ar}, which is most frequently used in analyzing two-phase flows:

$$r_{ar} = \bar{r} = \int_0^\infty r f(r)\, dr = \sum_{i=0}^{\infty} \frac{r_i n_i}{n_0}. \qquad (26)$$

The expression for the true, or "frozen" moisture content of the flow, assuming spherical drops, may be written in the form

$$y=\frac{m_d}{m_d+m_v}=\left[1+\frac{\rho_v}{\rho_l}\left(\frac{1}{V_d N_{ca}}-1\right)\right]^{-1}. \quad (27)$$

Assuming that the true and flow-rate mass concentrations (moisture contents) are related by the expression $y_f = \nu y/[(1-y)+\nu y]$ — where the slip coefficient $\nu = u_d u^{-1}$ is equal to the ratio of the velocity of the particles to the velocity of the continuous phase — and taking into account the condition of sphericity of the drops, we obtain the following expression for the flow-rate moisture content of the vapor

$$y_f=\left[1+\frac{\rho_v}{\rho_l \nu}\left(\frac{3}{4\pi}\bar{r}^{-3}N_{ca}^{-1}-1\right)\right]^{-1}. \quad (28)$$

If we assume that the flow-rate moisture content changes negligibly along the vapor channel of the heat pipe, then, allowing for (28), we may write on expression for the maximum heat-transfer capacity of the heat pipe

$$Q=(1-y_f)G_{lw}r^*=\left\{1-\left[1+\frac{\rho_v}{\rho_l \nu}\left(\frac{3}{4\pi}\bar{r}^{-3}N_{ca}^{-1}-1\right)\right]^{-1}\right\}G_{lw}r^*. \quad (29)$$

Equations (27)-(29) are convenient for experimental verification using optical methods of probing the vapor flow, making it possible to obtain information on $\bar{r}N_{ca}$ at a certain ratio ρ_v/ρ_l.

To study the performance of a heat pipe with a two-phase mass flow in the vapor channel, we experimentally investigated a heat pipe made of steel Kh18N9T. The pipe was 1.0 m long and $4.5\cdot 10^{-2}$ m in diameter. The capillary structure used was a single layer of a braided serge netting of 72% porosity. The behavior of the vapor was studied by the optical method [21]. For this purpose, the ends of the heat pipe were made optically transparent. To prevent vapor (C_3H_6O) condensation on these ends, the heat flow was formed symmetrically using a central vaporizer and two condensers close to the windows, and uncondensed (gaseous) nitrogen (0.01 g) was introduced into the vapor channel. The vapor flow was probed with a light beam $d_{lb} = 3\cdot 10^{-3}$ m formed by a universal monochromator. The state of the vapor was judged from the change in the intensity of the beam as it passed through the heat pipe in the axial direction.

The results of the experiment are shown in Fig. 4. In evaluating the possibility of using the above optical method as a diagnostic tool, it was assumed that in our case single scattering would occur. Thus, for the attenuation of the beam in the vapor channel, Bouguer's law applies:

$$J=J_0\exp(-\gamma l_{ef}). \quad (30)$$

According to [20], the attenuation factor, equal to the total attenuation cross section of the medium referred to 1 cm^3, is determined from the equality

$$\gamma=\int_0^\infty \pi r^2\theta(r)f(r)\,dr. \quad (31)$$

For monodisperse particles, the attentuation factor of the medium

$$\gamma=\pi r_{ar}^2\theta(r)N_{ca}. \quad (32)$$

Substituting (30) and (31) into the earlier-obtained equation (28) for the moisture content of the vapor, we obtain an expression for analyzing the experimental data on beam attenuation

$$y_f=\left[1-\frac{\rho_v}{\rho_l \nu}\left(\frac{3}{4}\frac{\theta(r)\,l_{ef}}{r_{ar}\ln(I/I_0)}+1\right)\right]^{-1}. \quad (33)$$

In analyzing the data obtained on I/I_0, we assumed that $\nu = 1$, $\theta \approx 0.5$, and the effective diameter of the drops $d_d \approx 0.05$ mm (from visual observations). On the basis of the derived expression (33) and the assumptions made, we obtained the dependence of the flow-rate moisture content on the heat flux. This dependence was also calculated with Eqs. (17), (20), and (21) and is represented in this case by the curve in Fig. 4.

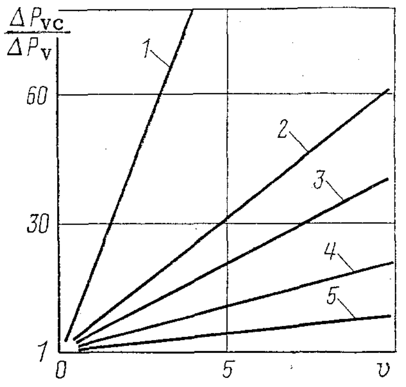

Fig. 3

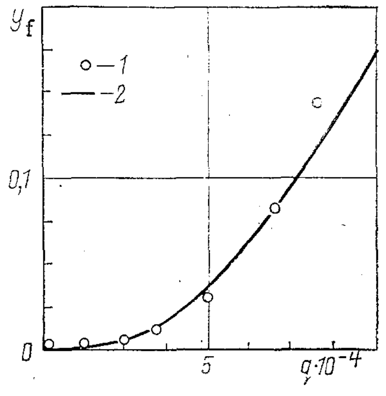

Fig. 4

Fig. 3. Dependence of ratio of pressure drops in the vapor channel of a heat pipe in a vapor-drop flow and a flow of dry vapor for different volume moisture contents in the vapor flow: 1) $\beta = 0.01$; 2) 0.03; 3) 0.002; 4) 0.001; 5) 0.00033. ν, m/sec.

Fig. 4. Dependence of mass moisture content of vapor in a heat pipe on heat flux in the vaporizer: 1) results of experiment; 2) calculated curve. $q \cdot 10^{4}$, W/m^{2}.

Analysis of the results obtained shows good agreement between the experimental data and the calculated dependence for the chosen value of the constant C = 2240. Quantitative studies confirmed the conclusion that vapor density affects drop mass and the moisture content of the vapor-drop flow, and the extreme character of this relation was noted. Values of mass moisture content greater than 70% were obtained in the experiments. It was noted that wet vapor is also present during operation of the heat pipe in the evaporative regime. In this case, the moisture content depends on both the density of the vapor and the constant C, characterizing the vaporization conditions.

NOTATION

Q, heat flow; q, heat flux in the vaporizer; T, temperature; P, pressure; S, surface area; l, length; r, radius; G, mass velocity; y, mass moisture content; n, number of droplets; V, volume; N_{ca}, calculated concentration; v, axial linear velocity; d, diameter; β, volume moisture content; α, inclination of heat pipe; m, mass velocity from a unit surface area; F, force; w, radial velocity; I, intensity of beam; K, permeability; N, parameter; ν, slip coefficient; g, gravitational constant; η, distribution parameter; $\tilde{\sigma}$, variance; x, y, z, coordinates; C, constant; θ, attenuation efficiency factor; ξ, friction coefficient; γ, attenuation factor of medium referred to a unit surface area; $n = S_d/S_s$, coefficient of heat-flow transformation; k, heat-transfer coefficient in the heat-pipe condenser. Indices: l, liquid; w, wick; hp, heat pipe; ef, effective value; cap, capillary structure; d, drop; v, vapor; f, flow-rate value; s, vaporizer; cond, condenser; e, free-falling; ar, arithmetic mean value; a, adiabatic zone; lb, light beam.

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HEAT TRANSFER AND THE COMBUSTION TEMPERATURE OF COKE PARTICLES IN A FLUIDIZED BED

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The temperature of carbon particles undergoing combustion in a fluidized bed is measured. Heat-transfer laws are ascertained.

In recent years power engineering has made use of coal combustion in a fluidized bed of a noncombustible, coarsely dispersed material [1]. Carbon particles burning in a fluidized bed are heat sources having a higher temperature than the surrounding particles of nonburning material. Thus, information on their temperature is of great practical interest. Such information is needed, first of all, to evaluate the particle combustion time and, second, to predict the moment of fusion of the ash portion of the fuel.

Several experimental studies have recorded the temperature of a relatively coarse carbon particle burning in a fluidized bed of a finely dispersed incombustible material [2-4]. A carbon sphere was immersed in the bed and the temperature of its center measured with a thermocouple in [2, 3]. The particle was attached to a flexible thermocouple and could thus move a little. The authors of [4] used a different method to measure particle temperature. They prepared a carbon sphere with several loops of wire with a known melting point. They then determined the temperature of the layer in which the loops melted and used this determination

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