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#### COOLING OF OVERHEATED SURFACES BY A FALLING LIQUID FILM

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UDC 536.242:532.62

A method is presented for calculating the cooling of overheated surfaces by a falling liquid film. Comparison of the results with available empirical data shows that they agree well.

The problem of rewetting overheated surfaces under transient conditions with a moving wetting front has attracted the attention of researchers in recent years in connection with the problem of emergency cooling of the active zone of nuclear reactors [1, 2] and other equipment [3].

To keep the temperature of fuel elements in the core of a nuclear reactor below the tolerable level during emergency loss of coolant, reactor designers usually provide an emergency cooling system which injects water or floods the area with water from below.

When coolant is delivered to the surface of fuel elements, a water film is formed and flows down the element at a certain velocity.

The physical reality of the rewetting process is complicated, since the initial temperature of the fuel element wall is fairly high and it is not wetted by the liquid. Here, a so-called cooling front is formed. This front advances over the surface as it cools to the temperature at which the liquid begins to wet the surface. In the literature [4], this temperature is known as the Leidenfrost point. It was shown in [5, 6] that this point is a constant value at constant pressure. Thus, the problem of cooling a heated surface reduced to studying the laws of the process of advance of an isotherm corresponding to the Leidenfrost

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Translated from *Inzhenero-Fizicheskii Zhurnal*, Vol. 44, No. 3, pp. 383-390, March, 1983. Original article submitted September 8, 1981.

point on the surface being cooled. Mathematically, heat transfer in the wall is described in general form by the heat conduction equation

$$\operatorname{div}(\lambda \operatorname{grad} \theta) + q_v = \rho c_p \frac{\partial \theta}{\partial \tau} \quad (1)$$

It was empirically established in [5] that the rate of advance of the cooling front is nearly constant along the surface. This allows us to change over to a coordinate system connected with this front and to thus examine the problem in a quasistationary approximation. For this purpose, we direct the  $x$  axis along the cooling surface and the  $y$  axis normal to same and make the following substitution of variables:  $x = x - u\tau$ , where  $u$  is the forward velocity of the wetting front. Then Eq. (1) has the form

$$\operatorname{div}(\lambda \operatorname{grad} \theta) + \rho c_p u \frac{\partial \theta}{\partial x} + q_v = 0. \quad (2)$$

Henceforth, for simplicity we will omit the  $\sim$  sign.

Now let us formulate the boundary conditions. We will examine the problem in a cylindrical coordinate system and choose the following as the domain:  $-\infty < x < \infty$ ;  $R_0 \leq y \leq R_1$ . The wetting front corresponds to the coordinate  $x = 0$ . The processes occurring in a region sufficiently far from the front will no longer have an effect on the latter, so the longitudinal temperature gradients in these regions will be equal to zero. In the region  $-\infty < x \leq 0$ , the amount of heat removed from the surface will be determined by the coefficient of heat transfer to the film  $\alpha(x)$ . In the unwetted region  $0 < x < \infty$ , heat transfer to the air will occur mainly via radiation. We will further assume that no heat is removed from the inside surface of the channel. The boundary conditions will have the form:

$$\frac{\partial \theta(-\infty, y)}{\partial x} = 0; \quad (3)$$

$$\frac{\partial \theta(\infty, y)}{\partial x} = 0; \quad (4)$$

$$\lambda \frac{\partial \theta(x, R_1)}{\partial y} = \begin{cases} \alpha [\theta(x, R_1) - T_q(x)], & -\infty < x \leq 0; \\ q_{\text{rad}}, & 0 < x < \infty; \end{cases} \quad (5)$$

$$\frac{\partial \theta(x, R_0)}{\partial y} = 0. \quad (7)$$

As can be seen from the boundary conditions, they contain the mean mass temperature of the liquid, the value of which is related to the wall temperature and the heat-transfer coefficient, i.e., if the liquid is subheated to the saturation temperature, an additional equation will be necessary to determine the temperature of the liquid. From the energy balance we have

$$\frac{\partial T_q}{\partial x} = \frac{\alpha}{\rho_q c_{p,q} (v - u)} [\theta(x, R_1) - T_q(x)]. \quad (8)$$

The boundary conditions for Eq. (8)

$$T_q(-\infty) = T_{\text{qin}}, \quad (9)$$

where  $T_{\text{qin}}$  is the temperature of the cooling liquid at the channel inlet.

Thus, Eqs. (2) and (8), together with boundary conditions (3)-(7) and (9), completely describe the process of cooling of a heated surface by a falling liquid film.

The main difficulty in formulating the problem and, thus, in choosing a method of solving it is specifying the heat-transfer coefficient. In most researches, the coefficient is assumed to be constant close to the wetting front. However, its value is in question, and, for example, the authors of [7] chose a value such that the velocity of the cooling front obtained in solving unidimensional equation (2) would coincide with the empirical value. The incorrectness of such an approach is obvious. In [8], the region close to the front was broken down into three zones in which the heat-transfer coefficients were constant. However, the authors failed to physically substantiate their selection of the boundaries of the zones, as well as the fact that the heat-transfer coefficients in each zone were assumed constant. A further step in solving the problem being discussed was the attempt made in [6] to assume

that the heat-transfer coefficient near the front is proportional to the square of the temperature difference ( $\alpha \sim (\theta - T_q)^2$ ). However, this approximation does not agree with the physical picture of the process taking place, which can be represented as follows.

Far from the wetting front, the temperature of the surface is close to the temperature of the cooling liquid, and the heat-transfer coefficient is determined by convective heat removal to the liquid film. Approaching the front, the temperature of the surface increases. When it reaches a certain value several degrees higher than the saturation temperature, nucleate boiling begins, with its characteristic dependence of the heat-transfer coefficient on the temperature of the channel surface. With a further increase in wall temperature, nucleate boiling is gradually replaced by transitional boiling characterized by a reduction in the heat-transfer coefficient with an increase in the temperature head. Finally, at the Leidenfrost point, the liquid ceases to wet the surface, and the heat-transfer coefficient assumes the value characteristic of the dry surface. Unfortunately, there is no reliable data for determining the heat-transfer coefficients under the conditions described above.

All of the well-known works also fail to consider the dependence of the thermal conductivity of the wall on temperature. For example, the thermal conductivity of steel 1Kh18N9T is 1.5 times higher at 500°C than at 100°C, and failure to account for this obviously diminishes the accuracy of the calculations.

For the sake of simplicity, let us examine a unidimensional formulation of the problem. Using [5], Eq. (2) has the following form at  $qv = 0$

$$\frac{d}{dx} \left( \lambda \frac{d\theta}{dx} \right) + \rho c_p u d \frac{d\theta}{dx} - \alpha_1 (\theta - T_q) = 0, \quad (10)$$

the boundary conditions:

$$\frac{\partial \theta(-\infty)}{\partial x} = 0; \quad (11)$$

$$\frac{\partial \theta(\infty)}{\partial x} = 0; \quad (12)$$

$$\theta(0) = \theta_0, \quad (13)$$

as well as

$$\alpha_1 = \alpha \quad \text{at} \quad -\infty < x \leq 0; \quad (14)$$

$$\alpha_1 = 0 \quad \text{at} \quad 0 < x < \infty. \quad (15)$$

The problem can be stated in a unidimensional formulation if  $Bi \ll 1$ , which is valid for all thin shells.

For the calculations, we assumed that the cooling process took place on a thin-walled tube of stainless steel 1Kh18N9T with a wall thickness of 1 mm. The thermal conductivity of the steel was calculated from the equation [9]  $\lambda = 0.0152\theta + 14.939$ . This formula interpolates the experimental data in [9] to within  $\pm 2\%$  in the range from 50 to 500°C. There are currently no reliable relations for calculating  $\alpha$ . To specify the heat-transfer coefficient on the basis of the above qualitative relation, we take the following for the region of convective heat transfer [10]

$$\alpha = 0.0066\lambda Re^{14/15} Pr^{0.344/\delta}. \quad (16)$$

Considerable difficulty arises in determining the wall temperature at which boiling begins, particularly if the coolant liquid is subheated to  $T_s$ . There is presently no accurate data on determination of the temperature of the beginning of boiling for our conditions. Thus, in the calculations we assumed that boiling begins when the wall temperature reaches 105°C. This value is typical of boiling in a large volume under atmospheric pressure [10]. At  $\theta > 105^\circ\text{C}$ , boiling begins with the heat-transfer coefficient equal to

$$\alpha_b = c(\theta - T_s)^2, \quad (17)$$

where  $c$  is determined from the condition  $\alpha = \alpha_b$  at  $\theta = 105^\circ\text{C}$ .

Transitional boiling occurs at a certain value of the temperature head. There is no recommendation in the literature on determining this value under our conditions. We therefore chose 30°C for the temperature head in the calculations, this temperature being characteristic of boiling in a large volume at atmospheric pressure [10]. The exact behavior of

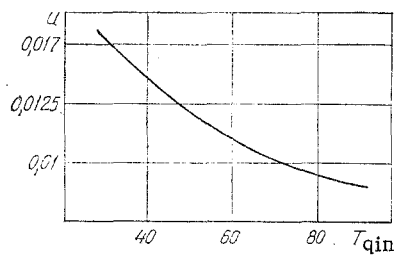


Fig. 1

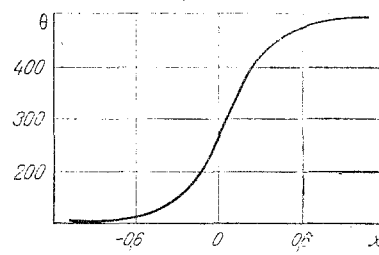


Fig. 2

Fig. 1. Dependence of velocity of cooling front on liquid temperature at inlet.

Fig. 2. Distribution of temperature of cooling surface near wetting front.

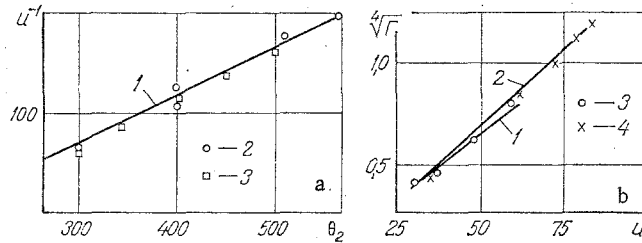


Fig. 3. Dependence of velocity of wetting front on initial temperature of heated dry surface (a) and flow rate of coolant liquid (b): a)  $\Gamma = 3.45$ ; 1) curve approximating the experiment in [5]; 2) experimental points in [5]; 3) calculated values; b)  $\theta_2 = 500^\circ\text{C}$ ; 1) curve approximating the experiment in [5]; 2) approximating the results of the calculations; 3) experimental points in [5]; 4) calculated values.

the heat-transfer coefficient during transitional boiling is unknown. The literature data is rather conflicting. Thus, for our calculations, we used several temperature-head dependences of the heat-transfer coefficient:

$$\alpha_h = b \exp[-[(\theta - T_s) - 30]^2/1600], \quad (18)$$

$$\alpha_h = b \frac{\theta - \theta_0}{130 - \theta_0}, \quad (19)$$

$$\alpha_h = \frac{b}{2} \left[ \cos\left(\frac{\theta - 130}{\theta_0 - 130} \pi\right) + 1 \right], \quad (20)$$

where  $b = c \cdot 900$ . If the wall temperature is higher than the Leidenfrost point, then we take zero for the heat-transfer coefficient.

Thus, the processes involved in the cooling of a heated surface by a falling liquid film were calculated on the basis of Eqs. (10) and (8), with appropriate boundary conditions and values for  $\lambda$  and  $\alpha$ . These equations were solved numerically on an ES-1030 computer.

Now let us specify the calculation method for solving system (8), (9). Since these equations are nonlinear, the choice of calculation method is important. It was shown in [11] that the adjustment method is best for solving problems of this type. In such a formulation, Eq. (10) takes the form

$$\frac{\partial \theta}{\partial \tau} + \frac{\partial}{\partial x} \left( \lambda \frac{\partial \theta}{\partial x} \right) + \rho c_p u \frac{\partial \theta}{\partial x} - \frac{\alpha_1}{d} (\theta - T_q) = 0. \quad (21)$$

We will write Eq. (21) in finite differences in implicit form:

$$\frac{\theta_i^{h+1} - \theta_i^h}{\tau} = \frac{2\lambda_{i+1}^h \theta_{i+1}^{h+1}}{h_i(h_{i-1} + h_i)} + \frac{2\lambda_i^h \theta_{i-1}^{h+1}}{h_{i-1}(h_{i-1} + h_i)} - \frac{\lambda_{i+1}^h + \lambda_i^h}{h_{i-1}h_{i+1}} \theta_i^{h+1} + \frac{\rho c_p u}{h_{i-1} + h_i} (\theta_{i+1}^{h+1} - \theta_{i-1}^{h+1}) - \frac{\alpha_i^h}{d} (\theta_i^{h+1} - T_i^h), \quad (22)$$

where  $\theta_1^k$ ,  $\lambda_1^k$ ,  $\alpha_1^k$ ,  $T_1^k$  are the temperature and thermal conductivity of the wall and the heat-transfer coefficient and temperature of the liquid, respectively, at the  $i$ -th point on the  $k$ -th time layer.

To determine  $T$ , we solve Eq. (8), using one of the modifications of the fourth-order Runge-Kutta method, and take the value of  $\theta$  from the preceding time layer to calculate the value for each current step. The computing process is done in the following sequence. We select some value of  $u$  (for example,  $u = 0.001$ ). We first take the interval  $-\infty < x \leq 0$ . Instead of  $-\infty$ , we take  $x = 0.01$  m. We chose the step so that it has a minimum value ( $h_N = 0.00001$  m) in the neighborhood of  $x = 0$ . The number of steps  $N = 400$ . To begin calculating Eq. (22), it is necessary to assign the initial temperature field on the surface. As shown in [12, 13], the initial temperature field can be chosen arbitrarily as long as the boundary conditions are satisfied. We thus find the field  $\theta_1^i$  as follows:

$$\theta_1^i = \theta_s + \frac{\theta_v - \theta_s}{2} \left[ 1 + \cos \left( \pi + \frac{x_1 - x_i}{x_1} \frac{\pi}{2} \right) \right]. \quad (23)$$

The values of  $\theta_1^i$  are used to determine  $T_1^i$  from Eq. (8). We then use unidimensional trial runs of  $\theta_1^i$  and Eq. (22) to change to a new time layer. The calculation is continued until the temperature values on adjacent time layers differ from one another by less than  $0.01^\circ\text{C}$ . Then the temperature gradient of the wall at the front is calculated from the formula

$$\left. \frac{d\theta}{dx} \right|_{x=0^-} = \frac{0.5\theta_{N-2} + 1.5\theta_N - 2\theta_{N-1}}{h_{N-1}}. \quad (24)$$

Then Eq. (10), with the corresponding value of  $\alpha$ , is solved on the interval  $\infty < x \leq 0$ . The solution is found in the manner similar to the above. Finally, we find the temperature gradient at  $x = 0^+$  on this interval:

$$\left. \frac{d\theta}{dx} \right|_{x=0^+} = \frac{2\theta_2 - 1.5\theta_1 - 0.5\theta_3}{h_2}. \quad (25)$$

After this, we compare the values of  $\left. \frac{d\theta}{dx} \right|_{x=0^-}$  and  $\left. \frac{d\theta}{dx} \right|_{x=0^+}$  and use the secant method to see if the difference  $\left. \frac{d\theta}{dx} \right|_{x=0^-} - \left. \frac{d\theta}{dx} \right|_{x=0^+}$  differs in absolute value by less than 0.1% from half the sum of these quantities. As a result, we obtain the forward velocity of the cooling front  $u$  and the temperature profile of the surface in the vicinity of the front.

Figures 1 and 2 show the results of calculations performed by the above method. Figure 1 shows the dependence of the front velocity on the subcritical temperature of the coolant  $\theta_2 = 500^\circ\text{C}$  and  $\Gamma = 1.96$ . The heat-transfer coefficient for transitional boiling was assigned with Eq. (18). Analyzing the results, we can conclude that subheating the coolant liquid increases the forward velocity of the front, although only slightly so in terms of absolute value. Figure 2 shows the distribution of the temperature of the surface being cooled on both sides of the cooling front for the same regime parameters. As our calculations show, the temperature curves nearly coincide close to the front for different laws of change in the heat-transfer coefficient, although the velocity of the front turns out to be somewhat sensitive to the behavior of  $\alpha$ . This phenomenon can evidently be explained by the fact that the velocity of the front depends not only on the path of the temperature curve, but also on the temperature gradient in the immediate vicinity of the front. At  $\Gamma = 1.96$  and  $\theta_2 = 500^\circ\text{C}$ , the velocity of the front is roughly 20% lower at  $\alpha$  calculated from Eq. (19) than at  $\alpha$  calculated from Eq. (18). Thus, for more thorough and accurate future calculations, it will be necessary to specify the relation for calculating  $\alpha$ .

Since most of the experimental researches of which we are aware do not indicate the geometric parameters of the working sections, the flow rate of the coolant liquid, or the thermo-physical properties of the surface being cooled, comparison with other data is difficult. We will compare our data with the empirical findings in [5].

The parameter ranges for the calculations were as follows: the initial temperature of the dry section ranged from  $300$  to  $500^\circ\text{C}$ , with spray densities from  $0.49$  to  $18.6$ . Figure 3 compares the calculated data with the experimental data.

It can be seen from the figure that the calculated data agrees satisfactorily with the experimental results, despite the imperfection of the unidimensional model used. This indicates that the surface-temperature dependence of the heat-transfer coefficient which we chose is very close to the actual dependence. It is apparent from Fig. 3a that the agreement becomes particularly good at low coolant flow rates. This is to be expected, since, as already noted, the unidimensional analysis is a more accurate approximation of the actual situation, the thinner the cooling surface and the lower the Bi numbers. Higher liquid flow rates or a thicker wall would require a two-dimensional analysis and, as already noted above, additional study of the heat-transfer coefficient. It is apparent from Fig. 3b that the dependence for the theoretical velocity of the cooling front also agrees satisfactorily with the experimental data. However, it should be noted that it is inadequate for more detailed analysis of experimental data, and is applicable even now only to thin surfaces and low liquid-coolant flow rates. With thicker walls and higher flow rates, we should expect the calculated data obtained by the unidimensional method to deviate from the experimental data.

In comparing the calculated data obtained to the experimental data, special attention should be paid to the following important difference between the above-described method and methods published earlier. All of the known calculation methods have one or several parameters (heat-transfer coefficient, Leidenfrost point, etc.) which is specially chosen so that the calculated results agree with empirical results. Naturally, the parameters thus chosen are random in nature and have no physical substantiation. Moreover, it should be expected that these calculations would not be applicable to experimental data obtained with other regime parameters. In the above-described method, all of the parameters were chosen on the basis of actual physical prerequisites and were not tied to any empirical results. The satisfactory agreement of the calculated results with experimental findings indicates that, at least for the regime parameters in the experiment, all of the physical premises were correctly chosen. To make the theoretical model more reliable, it will, as already noted, be necessary to obtain reliable empirical data for calculation of the coefficient of heat transfer to the liquid film in the above-indicated regimes, as well as to develop a two-dimensional model and define the areas of application of one- and two-dimensional models.

#### NOTATION

$\lambda$ , thermal conductivity of the material of the surface, W/m·K;  $\theta$ , temperature of the surface, °C;  $\theta_0$ , Leidenfrost point, °C;  $\theta_2$ , initial temperature of the dry heated surface, °C;  $q_v$ , internal heat flux, W/m<sup>2</sup>;  $\rho$ ,  $\rho_q$ , density of material and liquid, kg/m<sup>3</sup>;  $c_p$  and  $c_{p,q}$ , specific heat of the surface material and liquid, J/kg·K;  $u$ , forward velocity of wetting front, m/sec;  $\alpha$ , heat-transfer coefficient from surface to falling liquid film, W/m<sup>2</sup>·K;  $T$ , mean mass temperature of liquid, °C;  $T_s$ , saturation temperature of liquid, °C;  $\Gamma$ , weight density of liquid spray, N/m·sec; Bi, Biot number,  $Bi = \alpha d / \lambda$ ;  $d$ , wall thickness, m.

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## ATOMIZATION OF A LIQUID BY A ROTATING ATOMIZER BLOWN WITH AN AIRFLOW

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UDC 66.069.8

Monodispersed atomization of a fluid by a rotating atomizer blown by an airflow is experimentally studied. An equation, agreeing satisfactorily with the experimental data, is proposed.

Atomization of a fluid by a rotating disk or cone in monodispersed regimes has been examined in many papers. However, the case when air is blown coaxially past the disk has not been examined. This case is important for many practical applications. For example, when air is blown onto a disk, a turbulent air-drop spray, used for spraying commercially valuable plants with pesticides, is formed; in this case, it is important that the air flow not disrupt the monodispersed fragmentation of the liquid, i.e., the main advantage provided by the disk — the identity of the drops formed — must be conserved.

The experimental investigation was performed using the setup indicated schematically in Fig. 1. The rotating atomizer (cone) 1 with the drive from the electric motor 2 was placed in the outlet section of the exit nozzle 3, which was shaped like a converging tube and was oriented downwards. Air, blown by a fan, flowed through the nozzle 3 downwards, forming a turbulent free jet ( $Re = 10,600-67,000$ ; the diameter of the nozzle was 42 mm). The velocity of the air in the exit section of the converging tube and in the constant velocity core, shown in Fig. 1 by the dashed lines, was constant and equalled  $u$  (with the exception of a small section of the aerodynamic wake after the atomizer). The air velocity decreased beyond the boundaries of this core.

The liquid was delivered to the atomizer by an insert pump 4 from the cylinder 5 with a constant low flow rate  $Q = 0.005$  ml/sec and entered the rotating cone (diameter 25 mm) as a continuous jet.

We performed the experiments with liquid paraffin, diesel fuel, motor oil, and water. The physical characteristics of these liquids are indicated in Table 1.

The density of the test liquids varied insignificantly, the surface tension varied by a factor of 2.5, and the viscosity varied by a factor of 264.

A total of 61 experiments were performed. We shall examine the results of the first series of experiments, performed with liquid paraffin (25 experiments). We used a strobe tachometer to visualize the formation of the drops on the edge of the rotating cone.

The observations showed that when the velocity of the air flow  $u$  varied from 0 to 33 m/sec and the rotational frequency of the cone varied from 725 to 10,000 rpm (diameter of the cone 25 mm, fluid flow rate  $Q = 0.005$  ml/sec), in atomizing liquid paraffin, the nature of the fragmentation of the liquid did not change and corresponded to the well-studied first monodispersed regime ([1-3]; Fig. 2): protuberances appeared on the liquid torus, from which branches formed; the branches grew, stretched out, and dropped off the edge in the form of approximately identical drops. Some displacement of the liquid torus downwards, toward the aerodynamic shadow formed by the cone (by 0.5-1 mm), was observed only for very low rotational velocities of the cone (725-1160 rpm) and appreciable air velocities (25-30 m/sec), but otherwise the nature of the drop formation remained, in this case, unchanged.

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All-Union Scientific-Research Institute of Phytopathology. Translated from *Inzhenerno-Fizicheskii Zhurnal*, Vol. 44, No. 3, pp. 390-396, March, 1983. Original article submitted June 23, 1981.