BOILING OF SODIUM IN A CIRCULAR PIPE

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A single-bubble model for the boiling of a coolant in channels is analyzed. Calculations are reported on the dynamics of the ejection of sodium from a circular pipe during the boiling of sodium as a function of the initial superheating, the heat flux, and the point at which the vapor bubble forms.

A problem concerning investigators in many countries is the safety of fast reactors. The working temperatures of the coolant (sodium) at present do not exceed 600°C and are far from the saturation temperatures. However, as the most likely emergencies we should focus on the processes associated with the boiling of sodium in the active zone.

The necessary research includes the development of mathematical models for the boiling, an experimental study of the boiling of sodium in situations resembling the real apparatus in terms of structure and parameters, and the study of this process with model liquids (water, Freon, etc.).

The results of many experiments with water, alcohol, and Freon [1-3] show that the boiling can take the form of a single bubble or several bubbles in the interior of the channel, with liquid ejected toward both ends. In most previous experiments the boiling was caused by artificially arranging a rapid pressure drop (usually by breaking a membrane).

In a reactor, boiling of this type can begin if a channel becomes clogged and there is a rapid drop in the flow rate. At a zero flow rate and at the heat flux densities characteristic of fast reactors the temperature can rise at a rate reaching 1000°C/sec. Since the heat evolution is not uniform along the length of the channel, the temperature reaches a maximum at the center of the active zone. Under otherwise equal conditions, boiling is most probable here, and the inertia of the liquid column in the channel has a significant effect on the bubble growth rate.

The single-bubble model for boiling has been adopted most frequently for calculations. Various assumptions are used. For example, it has been assumed [4] that there is no film at the wall; in [5, 6], only the equation of motion was considered, with the pressure in the bubble assumed constant and equal to the saturation pressure at the wall temperature. The most thorough treatment of the boiling problem on the basis of this model was reported in [7]. Here the heat transfer from the repulsed liquid columns and heat transfer from the film were taken into account. It was assumed that uniformly distributed heat sources are acting in the liquid, at a rate equivalent to the heat flux from the wall. The heat flux from the wall was determined through a solution of the heat-conduction equation in the film. The initial temperature of the superheated liquid was set equal to the wall temperature. Attempts have been made to treat this problem analytically [10, 11].

The choice of the superheating level is extremely arbitrary. Much work has been published on superheating. The superheating range observed in the case of liquid metals is extremely broad, ranging from a few degrees to 800°C. The superheating has been found qualitatively as a function of the surface finish, the presence of impurities and dissolved gases in the coolant, the edge wetting angle, the flow velocity, the heating method, etc.

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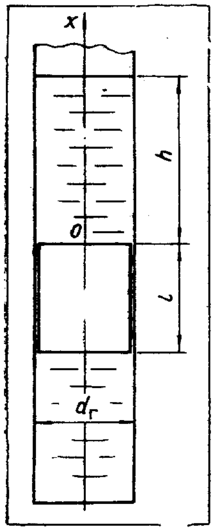

Fig. 1. The model channel.

However, these results are not an adequate basis for predicting the superheating in an actual apparatus. According to the data of [8], under the conditions prevailing in a fast reactor the superheating can range up to 150°C.

We report here the results of a theoretical study carried out to evaluate the influence of the initial superheating, the heat flux density, and the position at which the boiling begins on the course of the process. We adopt the simple rod-ejection model shown in Fig. 1.

A long, vertical, circular pipe is plugged at the bottom and filled with coolant to a certain level. The upper end of the pipe communicates with a free volume with a pressure of $P = P_0 = 2$ bars. Heat sources with a specific strength of $q_V = q_S(\Pi / F)$ are distributed uniformly throughout the coolant. There is no heat flux from the free surface of the coolant, nor is there heat transfer at the walls of the channel. At the beginning of vapor formation (the ejection), the coolant temperature is the same at all points $T_0 > T_S$. By adjusting T_0 we can set the initial superheating, $\Delta T = T_0 - T_S$, at various levels.

The vapor phase, which instantaneously fills the entire cross section of the channel, separating the liquid column into two distinct rods, forms at some distance h from the free surface.

We treated the cases with and without a liquid film at the wall during the ejection. Without this film, the thermal energy expended on the formation of the vapor phase was supplied by conduction from the liquid rods. With a film, there was also a heat flux through the lateral surface of the wall in the bubble zone, expended entirely on evaporation from the film. The film was assumed infinitesimally thin; it was assumed to be at the saturation temperature; and it was assumed to not evaporate completely during the ejection. The interfaces are planar and at thermodynamic equilibrium with the vapor phase throughout the ejection process. The vapor in the bubble is at the saturation temperature. The upper liquid column is repelled by the excess pressure of the vapor phase. The mass of repelled liquid is assumed constant; the inertia of the vapor and its resistance are neglected. Under these assumptions the motion of the liquid column can be described by the one-dimensional equation

$$\gamma' h \frac{d^2 l}{d\tau^2} = P_v - P_0 - \gamma' g h - \xi \frac{\gamma' h}{2d}\left(\frac{dl}{d\tau}\right)^2. \tag{1}$$

The initial conditions are

$$l = 0, \quad \frac{dl}{d\tau} = 0. \tag{2}$$

at $\tau = 0$. To find the values of P_V, which depend on T_{in}, we must solve the heat-conduction equation in the liquid columns.

It follows from the assumption that the temperature in the coolant is initially uniform and that the heat transfer from the two rods is symmetric that it is sufficient to consider the one-dimensional heat-conduction equation for one of the rods:

$$\frac{\partial T}{\partial \tau} = k \frac{\partial^2 T}{\partial x^2} + \frac{q_v}{c_p \gamma'}, \tag{3}$$

$$T = T_0 \quad \text{at} \quad \tau = 0. \tag{4}$$

If there is no liquid film at the wall, the boundary conditions for the heat-conduction equation are

$$\begin{aligned} &\frac{\partial T}{\partial x} = 0 \quad \text{at} \quad x = h, \\ &2\lambda \frac{\partial T}{\partial x} = r\gamma'' \frac{dl}{d\tau} \quad \text{at} \quad x = 0. \end{aligned} \tag{5}$$

If there is a film at the wall, only boundary condition (6) is changed in Eqs. (1)-(6). This boundary condition becomes

$$2\lambda \frac{\partial T}{\partial x} + q_s \frac{\pi d l}{F} = r\gamma'' \frac{dl}{dt} \quad \text{at} \quad x = 0, \tag{6'}$$

where the second term on the left side takes into account the heat flux through the liquid film on the wall.

Equations (1)-(6) can be solved only by a finite-difference method. The quantities $d^2l/d\tau^2$ and $(dl/d\tau)^2$ which appear in Eq. (1) are written in terms of finite differences as follows:

$$\frac{d^2l}{d\tau^2}=\frac{l_{n-1}-2l_n+l_{n+1}}{(\delta\tau)^2},\qquad \left(\frac{dl}{d\tau}\right)^2=\frac{l_n-l_{n-1}}{\delta\tau}\,\frac{l_{n+1}-l_{n-1}}{2\delta\tau}. \tag{7}$$

The resistance coefficient ξ_n corresponding to time τ_n is written as

$$\xi_n=\frac{0{,}316}{\left(\frac{l_n-l_{n-1}}{\delta\tau}\cdot\frac{d}{\nu}\right)^{0.25}}. \tag{8}$$

Then the difference equation equivalent to (1) can be written as

$$a(l_{n+1}-2l_n+l_{n-1})+b(l_n-l_{n-1})^{0.75}(l_{n+1}-l_{n-1})=f_n(\delta\tau)^2,$$

where $a=\gamma'h$; $b=\dfrac{0{,}316\gamma' h}{4d\left(\frac{d}{\nu}\,1/\delta\tau\right)^{0{,}25}}$; $f_n=P_v-P_0-hg\gamma'$. Alternatively, solving for l_{n+1}, we have

$$l_{n+1}=\frac{2al_n-[a-b(l_n-l_{n-1})^{0.75}]\,l_{n-1}+f_n(\delta\tau)^2}{a+b(l_n-l_{n-1})}. \tag{9}$$

In accordance with conditions (2) we can write $l_0=0$, $l_1=0$. If we know the values of the function f_n, we can use Eq. (9) to find l_2, l_3, etc.

To find the values of f_n, which depend on $(T_{in})_n$, we must solve Eq. (3) with boundary conditions (5), (6) for each time τ_n. We partition the solution interval $0<x<h$ into m equal subintervals. We place the calculation points at the centers of these intervals; i.e.,

$$x_i=\left(i-\frac{1}{2}\right)\delta x,\ i=1,2,\ldots,m.$$

To write boundary conditions (5), (6) we need the fictitious angles

$$x_0=-\frac{1}{2}\delta x\quad\text{and}\quad x_{m+1}=h+\frac{1}{2}\delta x.$$

We write the difference equation for T, equivalent to Eq. (3), in accordance with an explicit scheme:

$$T_i^{n+1}=T_i^n+\frac{k\delta\tau}{(\delta x)^2}\,[T_{i-1}^n-2T_i^n+T_{i+1}^n]+\frac{q_v}{c_p\gamma'}\,\delta\tau,\qquad i=1,\ 2,\ldots,m, \tag{10}$$

where T_i^n are the temperatures at points x_i at time τ_n. We find the functions T_0^{n+1} and T_{m+1}^{n+1} with the help of boundary conditions (5), (6):

$$T_{m+1}^{n+1}=T_m^{n+1},\ T_0^{n+1}=T_1^{n+1}-\delta x\,\frac{r\gamma''}{2\lambda}\cdot\frac{l_{n+1}-l_n}{\delta\tau}. \tag{11}$$

The temperature of the liquid–bubble interface is calculated from

$$T_{in}^{n+1}=\frac{1}{2}\,(T_0^{n+1}+T_1^{n+1}). \tag{12}$$

The values of P_v as a function of T_{in} are taken from tables for P_s, T_s. The calculation based on Eqs. (10)-(12) is stable under the condition

$$\frac{k\delta\tau}{(\delta x)^2}\leqslant\frac{1}{2}.$$

According to (10)-(12) the temperatures are calculated for time τ_{n+1} under the condition that the function l_{n+1} is already known.

Accordingly, for each time we first calculate l_{n+1} from Eq. (9) and then calculate all the T_i^{n+1} and T_{in}^{n+1} from Eqs. (10)-(12).

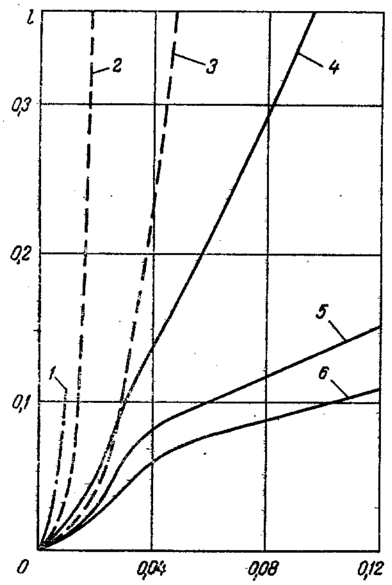

Fig. 2

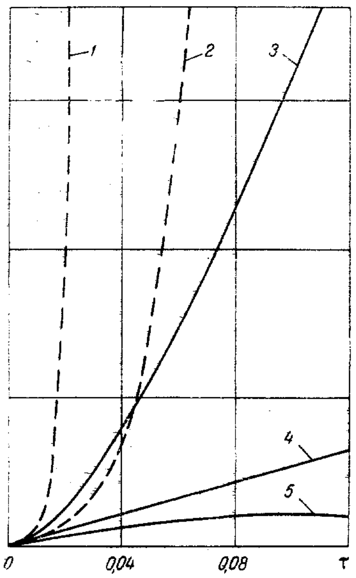

Fig. 3

Fig. 2. Length of the vapor bubble, l (m), as a function of the time τ (sec) for $\Delta T = 40°C$: 1) evaporation from a sodium film at the wall is taken into account, $q_s = 1.75 \cdot 10^6$ W/m^2, h = 0.1 m; 2) the same, $q_s = 1.75 \cdot 10^6$ W/m^2, h = 0.4 m; 3) the same, $q_s = 0.175 \cdot 10^6$ W/m^2, h = 0.1; 4) no evaporation from film, h = 0.4 m, $q_s = 1.75 \cdot 10^6$ W/m^2; 5) the same, $q_s = 0.175 \cdot 10^6$ W/m^2; 6) the same, $q_s = 0$.

Fig. 3. The length l (m) as a function of τ (sec) for $\Delta T = 10°C$, h = 0.4 m: 1) evaporation from the sodium film at the wall is taken into account, $q_s = 1.75 \cdot 10^6$ W/m^2; 2) the same, $q_s = 0.175 \cdot 10^6$ W/m^2; 3) no evaporation of the sodium film, $q_s = 1.75 \cdot 10^6$ W/m^2; 4) the same, $q_s = 0.175 \cdot 10^6$ W/m^2; 5) the same, $q_s = 0$.

If the boundary condition for T at x = 0 is used in form (6'), the difference equation for T_0^{n+1} is

$$T_0^{n+1} = T_1^{n+1} - \delta x \frac{r\gamma''}{2\lambda} \cdot \frac{l_{n+1} - l_n}{\delta\tau} + \frac{q_s \pi d}{4\lambda F}(l_{n+1} + l_n)\delta x. \qquad (13)$$

Calculations were carried out for heat flux densities of $q_s = 1.75 \cdot 10^6$ W/m^2 and $q_s = 0.175 \cdot 10^6$ W/m^2. The superheating was varied from 10 to 40°C, and the pressure was assumed to be $P_0 = 2$ bars. To evaluate the influence of the position of the boiling point we carried out calculations for both h = 0.4 m and h = 0.1 m.

The calculated results are shown in Figs. 2–4. In the case in which there is no liquid film on the wall, at the heat fluxes typical of the central part of a fuel element the influence of superheating is important only during the initial stage of the process. The maximum bubble growth rates and pressure changes are observed only in the initial stage. Thereafter the pressure decreases, becoming constant.

In the case with evaporation from the film, the ejection velocity increases sharply. While at a heat flux on the order of $1.75 \cdot 10^6$ W/m^2 the evaporation of a channel 0.4 m long lasts 0.10–0.12 sec (depending on the initial superheating) this evaporation occurs in 0.02 sec if there is no evaporation from a film; i.e., it occurs five times faster when evaporation from a film is taken into account. The velocity at which the liquid is ejected reaches 300 m/sec. The pressure rises to 50 bars at the maximum heat flux and to 7 bars at a heat flux an order of magnitude lower.

To show the influence of the position at which the boiling begins on the course of the process, we show in Fig. 2 the results of one of the calculations for a column height of h = 0.1 m. As expected, the process occurred more rapidly, since the mass of the liquid column was smaller.

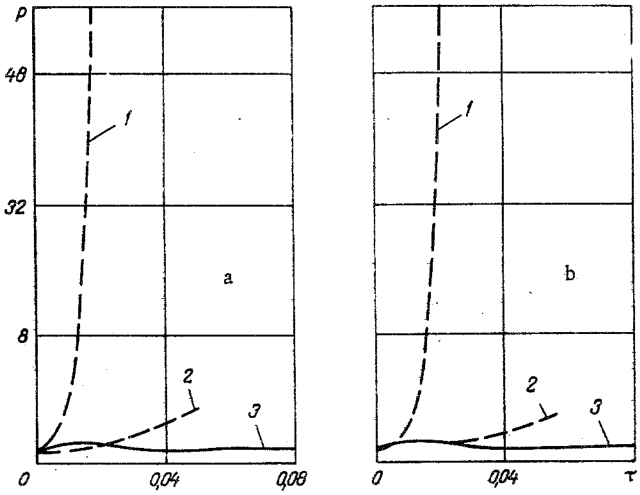

Fig. 4. Pressure in the vapor bubble, P (bars), as a function of time τ (sec): a) $\Delta T = 40°C$; b) $\Delta T = 10°C$; 1) with evaporation from the film, $q_s = 1.75 \cdot 10^6$ W/m^2; 2) the same, $q_s = 0.175 \cdot 10^6$ W/m^2; 3) without evaporation from the film, $q_s = 1.75 \cdot 10^6$ W/m^2.

The assumption that the sodium film is retained throughout the process may appear questionable. Estimates show that at a zero flow rate and at $q_s = 1.75 \cdot 10^6$ W/m^2 the sodium film thins at a rate of 0.65 mm/sec. According to the experimental data of [9], the thickness of a potassium film is 0.07-0.23 mm in the piston flow regime. Assuming the same thickness for the sodium film, we conclude that its evaporation lasts 0.1-0.4 sec. The coolant is ejected from the channel more rapidly. Consequently, the assumption that the film exists throughout the evaporation is quite reasonable.

The calculations neglect the energy expended on warming the film. At pressures up to 50 bars the energy expended on warming the film to the corresponding T_s is appreciable. The constancy of the ejected mass assumed for the calculations leads to some increase in the pressure in the bubble and to some increase in the ejection time.

The results of these calculations show that the superheating of the liquid is important only at the initial stage of the boiling. The ejection velocity and the pressure rise in the bubble depend on the presence of the film and on the heat flux from the wall in the film zone.

These calculated results agree qualitatively with the data of [7], obtained on the basis of a more complete description of the boiling process, but with other parameter values. This calculation model will be refined in accordance with the results of planned experiments.

NOTATION

q_v is the specific volume heat evolution;
q_s is the heat flux from surface;
Π is the perimeter;
F is the cross-sectional area;
T is the temperature;
T_s is the saturation temperature;
T_0 is the initial temperature;
T_{in} is the temperature at the liquid–bubble interface;
h is the height of ejected liquid column;
l is the bubble length;
γ is the density;
τ is the time;
P_v is the pressure in bubble;
P_0 is the pressure in the free volume;
g is the acceleration of gravity;

d	is the pipe diameter;
ξ	is the resistance coefficient, $\xi = 0.316/\mathrm{Re}^{0.25}$;
Re	is the Reynolds number, $\mathrm{Re} = d(dl/d\tau)/\nu$;
k	is the thermal diffusivity;
c_p	is the specific heat;
λ	is the thermal conductivity;
r	is the specific heat of vaporization;
x	is the coordinate in the liquid, reckoned from the bubble boundary;
prime	is the liquid;
double prime	is the vapor.

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